

POWER PLANT ENGINEERING

R. K. HEGDE

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Power Plant ENGINEERING

R. K. Hegde
Professor
Department of Mechanical Engineering
Srinivas Institute of Technology
Mangalore, Karnataka, India

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Fax: 080-30461003, Phone: 080-30461060
www.pearson.co.in, Email: companysecretary.india@pearson.com

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Dedicated to my father, N. R. Hegde, a humble teacher, who during his life time strived very hard to cherish the dreams of budding young kids and helped them to grow as responsible citizens.

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Preface

Energy, or power as it is often referred to, is critical, directly or indirectly, in the entire process of evolution, growth and survival of all living beings. It plays a vital role in the socio-economic development and human welfare of a country.

Power is a 'strategic commodity'. Any uncertainty about its supply can threaten the functioning of an economy, particularly in developing countries. Since the demand for power is growing consistently, an emphasis is given to generate power by utilizing various sources of environment-friendly energy. In this context, acquiring the fundamental aspects of power generation is a must for everyone.

The book discusses about the intricacies of power generation using both renewable and non-renewable resources throughout its nineteen chapters. Conventional method of power generation and the system details have been covered in the first fourteen chapters while the non-conventional methods of power generation have been explained in chapters 15 and 16.

The chapters cover issues like thermal power plant, steam power plant, diesel engine power plant, gas turbine plant, hydro-electric plant, and nuclear power plant, power from non-conventional sources and power plant economics and Instrumentation. All chapters are supplemented by neat sketches and illustrations while SI units have also been used throughout the book which makes it a good reference for engineering students and teachers alike.

Key Features

Salient features of the book are:

1. Thermal, hydro and nuclear power generations
2. Solar and wind energy conversion systems
3. Ocean, geothermal and biomass energy systems
4. Fluidized bed combustion technology
5. Power from non-conventional sources
6. Illustrations, sketches and supplementary photographs
7. Environmental aspect of power generation
8. Power plant economics

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About the Author



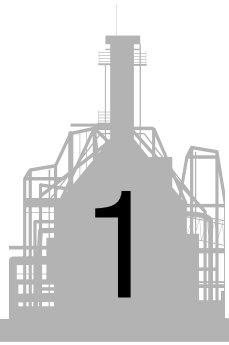
R. K. Hegde obtained his Ph.D. in nanofluid heat transfer from National Institute of Technology, Karnataka. The author has more than 20 years of rich industrial and academic experience. Earlier he was involved in power plant operation and maintenance, handling high pressure FBC boilers, Babcock–Wilcox boilers, turbines and pumps. He worked in a power plant in maintenance and is also an authorized boiler operation engineer. He is also a member of I.S.T.E. His area of interest includes nanofluids, compact heat exchangers, and CFD. He has more than 30 papers published in reputed international journals and also presented 15 papers in international/national conferences. He is also an active reviewer for reputed international journals.

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Introduction to Power Plants



1

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1.1 INTRODUCTION TO THE SOURCES OF ENERGY: CONVENTIONAL AND NON-CONVENTIONAL PRINCIPLE OF POWER GENERATION

A variety of energy sources are available to supply to expanding needs in each country around the world. These sources are broadly classified as commercial or conventional energy sources and noncommercial or nonconventional energy sources. Most of the developed countries are highly dependent on conventional form of energy, whereas developing countries such as India is dependent on both the forms of energy sources.

Energy sources may be mainly classified into two categories: renewable and non-renewable energy sources.

1. *Renewable energy sources*

These are produced by nature and are inexhaustible. Renewable energy sources include both direct solar radiation utilized by solar collectors and cells and indirect solar energy in the form of wind, hydropower, ocean energy and sustainable biomass resources.

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2. *Non-renewable energy sources*

These are either available in nature or produced by man artificially. They are exhaustible and non-renewable. Conventional energy sources such as nuclear power and fossil fuels are nonrenewable.

Advantages and disadvantages of renewable energy sources are as follows:
Advantages

(i) They are produced by nature and considered as inexhaustible.
(ii) They are pollution free, and hence eco-friendly.
(iii) If utilized properly in developing countries, they can save a lot of foreign exchange and generate employment opportunities.
(iv) Deployment is easy and rapid due to flexibility in their utilization.
(v) They are economical when considered over a longer period of time.

Disadvantages

(i) Their availability is intermittent (e.g. solar, wind, tidal, hydro, etc.) and hence need the assistance of nonrenewable energy.
(ii) Complete commercialization is difficult on a larger scale.
(iii) Initial cost is high due to the newer technologies used, which are still at preliminary stages.
(iv) Sources are not evenly spread across the globe.

In sections that follow, both conventional and nonconventional energy sources are discussed in detail.

1.1.1 Conventional Energy Sources

These are commercial forms of energy available. They include the following:

(i) Fossil fuel that may be in solid, liquid or gaseous forms
(ii) Water power or energy stored in water
(iii) Nuclear energy

Worldwide consumption of total energy is as shown in the Table 1.1.

Table 1.1 *Worldwide Consumption of Total Energy*

Source of energy	Contribution (%)	Overall contribution (%)
Coal	32.5	–
Oil	38.3	–
Gas	19.0	92
Uranium	0.13	–
Hydro	2.00	–
Wood	6.6	8

(*Continued*)

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Table 1.1 (*Continued*)

Source of energy	Contribution (%)	Overall contribution (%)
Dung	1.2	–
Waste	0.3	–

From Table 1.1, it is evident that 92 per cent of total energy comes from coal, oil, gas, and uranium, and hence these are the most commonly used commercial energy sources.

1. *Coal*

It is the most common source of energy that is being used since industrialization. Modern steam boilers burn coal mainly as primary fuel in any of its available forms. Different ranks of coal available are peat, lignite, bituminous and anthracite.

Figure 1.1 shows a thermal power plant using steam as a working fluid. It consists of a steam generator, a condensing turbine coupled to a generator, a condenser with a condensate extraction pump and a feed water tank with a feed pump.

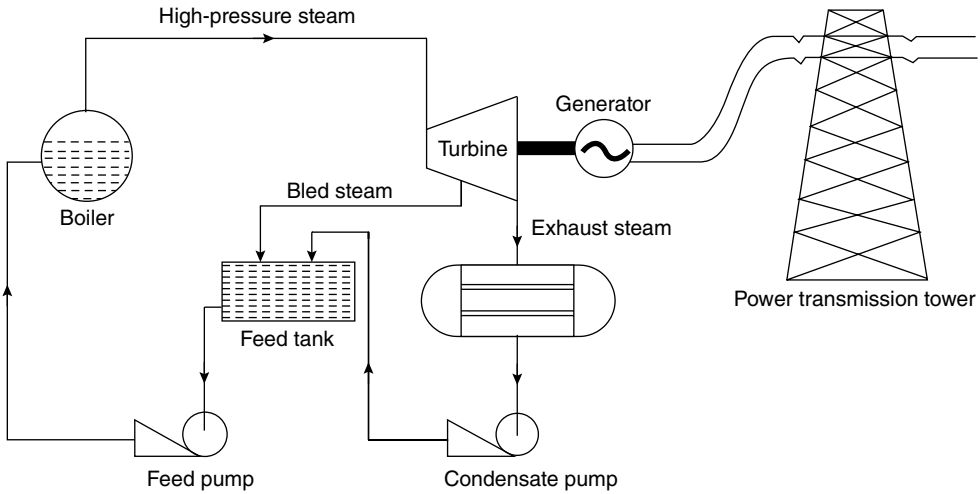


Fig. 1.1 *A Typical Thermal Power Plant*

In a coal-fired thermal power plant, coal is burnt in a boiler furnace. The heat generated is utilized to convert water, which is the working fluid, into superheated steam in the boiler or steam generator. The high-pressure steam is used to run the prime mover (steam turbine). The prime mover rotates along with an electrical generator coupled to it. Thus, mechanical energy is converted into electrical energy that is supplied to various points using power feeders. The steam that expands in the turbine is condensed into water in a surface condenser. The condensate water is pumped back to the feed tank. The feed water in the feed tank is heated by bleeding some amount of steam from the turbine. The hot feed water is pumped to the boiler using a feed pump.

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According to estimates, coal reserves are sufficient enough to last for 200 years. However, coal reserves have lower calorific value, and their transportation is uneconomical. When burnt, they produce pollutants such as CO and CO_2, and hence they are responsible for ecological imbalance.

2. *Oil*

Almost 40 per cent of energy needs is met by oil alone. With present consumption and a resource of 250,000 million tons of oil, it is estimated to last for only 100 years, unless more oil is discovered. Major chunk of oil comes from petroleum.

3. *Gas*

Due to the non-availability of ready market gas is not completely and effectively utilized and is burnt in huge quantities. Its transportation cost is much higher than oil. Large reserves are estimated to be located in inaccessible areas. Gaseous fuels are classified as follows:

(i) Gases of fixed composition such as acetylene, ethylene, methane, etc.

(ii) Industrial gases such as producer gas, coke oven gas, blast furnace gas, water gas, etc.

4. *Agricultural and organic wastes*

These include saw dust, bagasse, garbage, animal dung, paddy husk, corn stem, etc., accounting for a major energy consumption.

5. *Water*

It is one of the potential sources of energy meant exclusively for hydro-electric power generation. Potential energy of water is utilized to convert it into mechanical energy by using prime movers known as hydraulic turbines. The operating cost of the plant is cheaper as compared to other types of power plants. It is the only renewable non-depleting source of energy that does not contribute to pollution.

Figure 1.2 shows a hydraulic power plant designed for high head. In a hydro-electric power plant, water is stored behind a dam that forms a reservoir. Water is taken from the reservoir through tunnels from where it is distributed to *penstocks*. A penstock is a large diameter pipe that carries water to the turbine. Trash racks are fitted at the inlet of tunnels to prevent any foreign matter from entering into the tunnels. A surge tank built before the valve house prevents sudden pressure rise in the penstock when the load on the turbine decreases or when the inlet valves to the turbine suddenly get closed. The flow of water in the penstocks is controlled in the valve house that is electrically driven. Thus, potential energy of water is utilized to run the prime mover (hydraulic turbine) coupled to an electric generator in the power house. After doing work, the water is discharged to the tail race.

6. *Nuclear power*

Any matter consists of atoms held together by means of binding energy. Controlled fission of heavier unstable atoms such as U_{235}, Th_{232} and Pu_{239} liberates enormous amount of energy. This is possible only by utilizing small amount of nuclear fuels. It may be noted that the energy released by fission of one kilogram of U_{235} is equivalent to the heat generated by burning 4,500 kg of coal. This factor makes the nuclear energy more attractive. The energy generated during nuclear fission reaction is used to produce steam in heat exchangers, which is utilized to run the turbo-generators.

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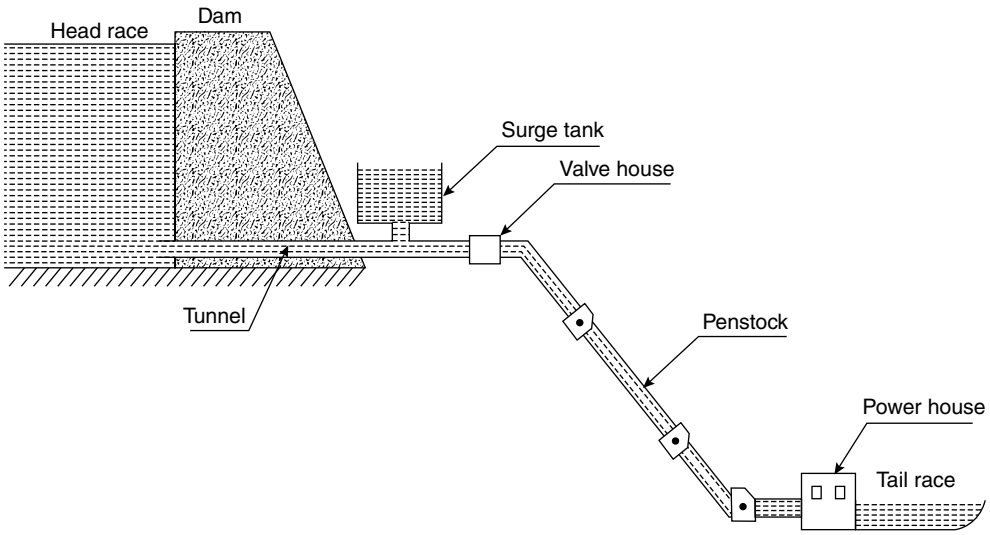


Fig. 1.2 *A Hydraulic Power Plant*

For nuclear power generation, three systems are considered. The first is based on natural uranium yielding power and plutonium. The second is by using plutonium and depleted uranium in a fast breeder reactor. The third system is by using thorium and converting it into uranium in a fast breeder reactor. India has uranium reserves only enough to produce 6×10^6 kW of energy that is a meager 1 per cent of its current energy requirements.

1.1.2 Non-Conventional Energy Sources

1. *Solar energy*

Solar energy has the greatest potential of all the sources of renewable energy that comes to the earth from sun. This energy keeps the temperature of the earth above that in colder space, causes wind currents in the ocean and the atmosphere, causes water cycle and generates photosynthesis in plants. The solar energy reaching the surface of the earth is about 10^{16} W, whereas the world-wide power demand is about 10^{13} W. This means solar energy gives us 1,000 times more energy than our requirement. Even if we use 5 per cent of this energy, it is more than 50 times out requirement. The total solar radiation absorbed by the earth and its atmosphere is 3.8×10^{24} J/yr. Figure 1.3 shows a concentrating type solar collector used to trap solar energy.

2. *Wind energy*

Wind energy can be economically used for the generation of electrical energy. Winds are caused by two main factors:

(i) Heating and cooling of the atmosphere which generates convection currents. Heating is caused by the absorption of solar energy on the earth's surface and in the atmosphere.

(ii) The rotation of the earth with respect to atmosphere and its motion around sun.

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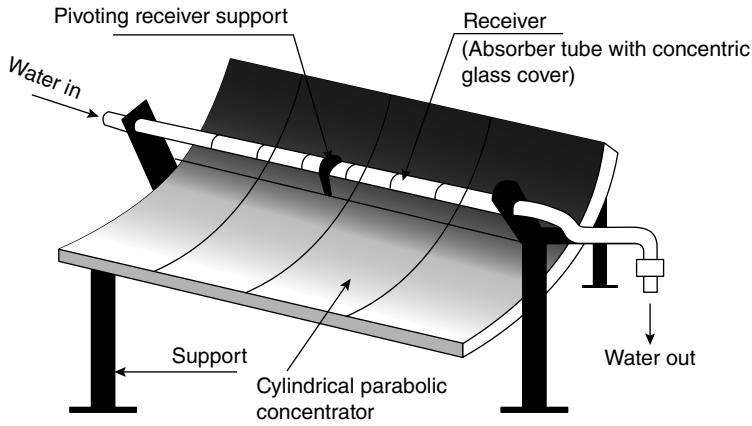


Fig. 1.3 *A Concentrating-Type Solar Collector*

The energy available in the winds over the earth's surface is estimated to be 1.6×10^7 MW, which is almost the same as the present-day energy consumption. Wind energy can be utilized to run windmill that in turn is used to drive to generators. India has a potential of 20,000 MW of wind power.

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Due to pressure differential existing between any two places on earth, air moves at high speed. This pressure differential is caused due to earth's rotation and by uneven heating of the earth by sun. The kinetic energy of air can be utilized to generate electric power. The kinetic energy per unit volume of moving air is given by the following equation:

$$E = \frac{1}{2}\rho V^2$$

where ρ is the density of air and V = average linear speed.

Figure 1.4 shows a typical windmill where water can be pumped out for irrigation and drinking purpose. Here the rotational motion of the wheel can be either translated into rotary motion (to generate electricity) or reciprocating motion (to drive the pump).

3. *Energy from biomass and biogas*

Bio-mass means organic matter that is produced in nature through photosynthesis. In the presence of solar radiation, water and carbon dioxide are converted into organic material, CH_2O.

CH_2O is stable at low temperature, it but breaks at higher temperature releasing heat equal to 469 kJ/mole.

$$CH_2O + O_2 \rightarrow CO_2 + H_2O + 469 \text{ kJ/mole.}$$

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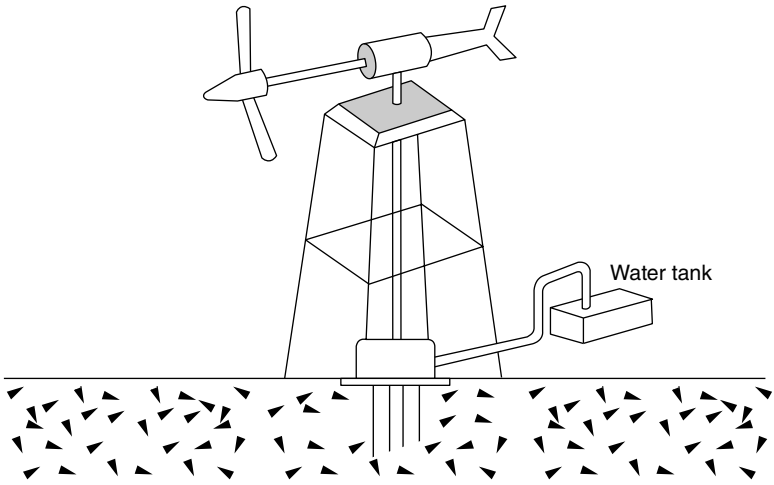


Fig. 1.4 *A Typical Windmill*

It is possible to produce large amount of carbohydrate by growing plants such as algae in plastic tubes or ponds. The algae could be harvested, dried and burned for production of heat that could be converted into electricity by conventional methods. The biomass can be either used directly by burning or can be processed further to produce more convenient liquid or gaseous fuels.

Three different categories of bio-mass resources are as follows:

(i) **Bio-mass in its traditional solid mass** such as wood and agricultural wastes that are burnt directly to get energy.

(ii) **Bio-mass in its non-traditional form** in which bio-mass is converted into ethanol and methanol that are used as liquid fuels in engines.

(iii) **Bio-mass in fermented form** in which biomass is fermented anaerobically to obtain a gaseous fuel called biogas that contains 55 – 65 per cent methane, 30 – 40 per cent CO_2 and rest as impurities such as H_2, H_2S and some N_2.

Various resources of bio-mass are as follows:

(i) Concentrated waste – municipal solids, sewage wood products, industrial waste and manure of large lots.

(ii) Dispersed waster residue – crop residue, legging residue and disposed manure.

(iii) Harvested biomass, stand by bio-mass and bio-mass energy plantation.

Total solar radiation absorbed by plants is about 1.3×10^{21} J/yr and world's standing bio-mass has an energy content of about 1.5×10^{22} J.

4. *Energy from oceans*

A large amount of solar energy is collected and stored in oceans. The surface of water acts as a collector for solar heat, while the upper layer of the sea constitutes infinite heat storage

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reservoir. The heat contained in the oceans could be converted into electricity due to the temperature difference (20–25 K) between the warm surface water of the tropical oceans and the colder waters in the depths. This is the basic idea of OTEC systems. The surface water that is at higher temperature could be used to heat some low-boiling-point organic fluid, the vapours of which would run a heat engine. The amount of energy available from OTEC is enormous, and is replenished continuously.

Figure 1.5 shows an OTEC plant. In this plant, sea water acts as a heat source, working fluid, coolant and heat sink. The warm surface water at 27°C is used to boil a low-boiling-point liquid, the vapours of which run a turbine. The working fluid may be ammonia, propane or freon operating at a pressure of about 10 bar.

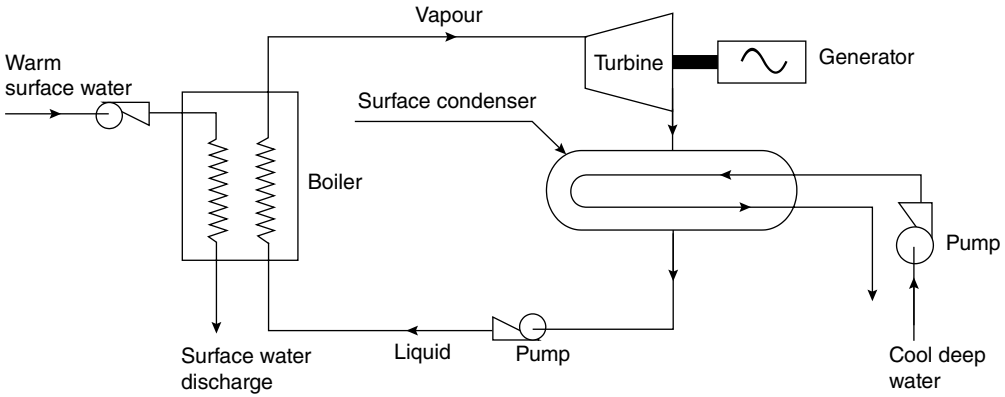


Fig. 1.5 *An OTEC Plant*

Disadvantages

(i) Efficiency is extremely low, and hence the system needs extremely large power plant heat exchangers and components.

(ii) Even though there is no fuel cost, the capital cost is very high, and hence the unit power cost is higher.

(iii) Involves developmental problems and uncertainties of market penetration.

5. *Tidal energy*

Due to universal gravitational effects of sun and moon on the earth tides in the sea are generated. Due to fluidity of water mass, there results periodic rise and fall of water level during the rising and setting of sun and moon. This periodic rise and fall of water level results in tidal power. When the water is above the mean sea level, it is called flood tide (high tide); and when the water is below the mean sea level, it is known as ebb tide (low tide). The energy dissipated with slowing down the rotation of the earth as a result of tidal action is about 10^{26} J/yr.

The tides are rhythmic but not constant. Their occurrence is due to a balance of forces, mainly gravitational force of the moon and sun to some extent, balancing the centrifugal force on water due to earth rotation. This results in rhythmic rise and fall of water. The moon rotates

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around the earth every 24 hr 50 min. During this period, tide rises and falls twice, resulting in a tidal cycle lasting for 12 hr 25 min. Thus, the tidal range R is given by the following relation;

R = water elevation at high tide − water elevation at low tide.

This range is maximum during new and full moons and is known as spring tide and neap tide.

To harness tides, a dam is built across the mouth of the bay with large gates and low head hydraulic reversible turbines (Figure 1.6). A tidal basin formed thus gets separated from the sea, by dam. There always exists a difference between the water levels on either side of the dam during low tide and high tide. Thus, the reversible water turbine runs continuously producing power by using the generator connected to it.

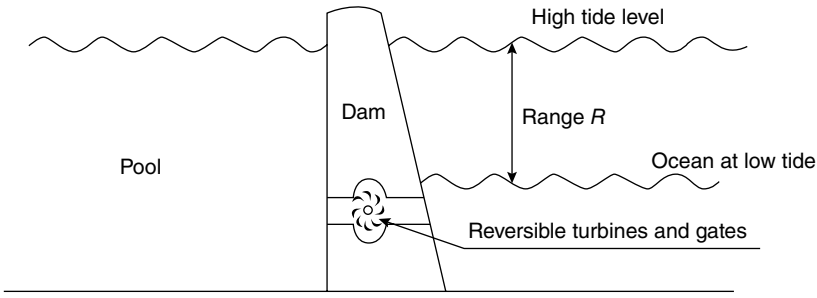


Fig. 1.6 *A Tidal Energy Harnessing Plant*

6. *Wave energy conversion*

Waves are caused by wind that in turn is caused by the uneven solar heating and subsequent cooling of the earth's crust and the rotation of the earth. When at its most active stage, wave energy produces more power than incident solar energy at its peak.

Total energy of waves is the sum of potential and kinetic energies. The potential energy arises from the elevation of the water above the mean sea level. The kinetic energy of the wave is that of the liquid between two vertical planes perpendicular to the direction of wave propagation x and placed one wavelength apart.

Figure 1.7 shows a typical wave energy conversion system. It consists of a square float that moves up and down with the water guided by four vertical manifolds. The platform is stabilized within the water by four large underwater floatation tanks supported by buoyancy forces that restrict the vertical or horizontal displacement of the platform due to wave action. Damping fins may be used so that the platform is stationary in space even in heavy seas.

A piston attached to the float moves up and down the cylinder. The cylinder is attached to the platform, and hence is stationary. The piston–cylinder arrangement acts as a reciprocating air compressor. The downward motion of the piston draws air into the cylinder via an inlet check valve. The upward motion compresses the air and sends it through an outlet check valve to the four underwater floatation tanks via the four manifolds. The four floatation tanks serve the dual purpose of buoyancy and air storage, and the four vertical manifolds serve the dual purpose of manifolds and float guides. The compressed air in the buoyancy-storage tanks is in turn used to drive an air turbine that drives an electrical generator. The electric current is transmitted to the shore via an underground cable.

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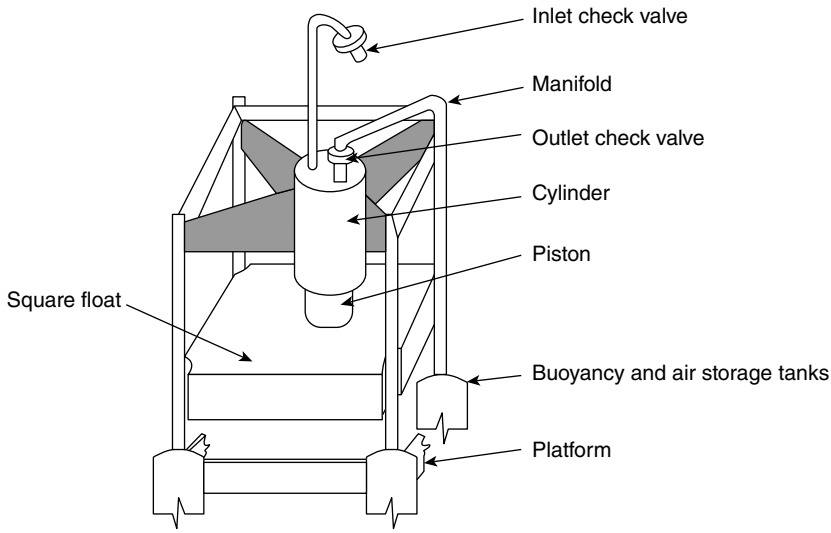


Fig. 1.7 *A Typical Wave Energy Conversion System*

7. *Geothermal energy*

This is the energy that comes from within the earth's crust. In some locations of the earth, the steam and hot water comes naturally to the surface. For large-scale use, bore holes are normally sunk with depth up to 1,000 m releasing steam and water at temperatures 200–300°C and pressure up to 3,000 kN/m^2. Two methods are generally used to generate power using geothermal energy.

Method I

In this method, the heat energy is transferred to a working fluid that operates the power cycle. It is found that molten interior mass of earth vents to the surface through fissures at temperatures ranging between 450 and 550°C.

Method II

In this method, the hot geothermal water and/or steam is used to operate the turbines directly. From the well head, steam is transmitted by using pipes of 1 m diameter over distances up to 3,000 m to the power plant. In this system, water separators are used to separate moisture and solid particles from steam.

The heat flux from the earth's interior through the surface is 9.5×10^{20} J/yr. The total amount of heat stored in water or steam to a depth of 10 km is estimated to be 4×10^{21} J and that stored in the first 10 km of dry rock is around 10^{27} J. Figure 1.8 shows a typical geothermal field.

8. *Hydrogen energy*

Hydrogen as an energy is another alternative for conventional fuels. It can be easily produced from water that is available abundantly in nature. It has the highest energy content per unit of mass of any chemical fuel and is a better substitute for hydrocarbons, with increased combustion efficiency. It is non-polluting and can be used in fuel cells to produce both electricity and useful heat. However, it has technical problems such as production, storage and transportation.

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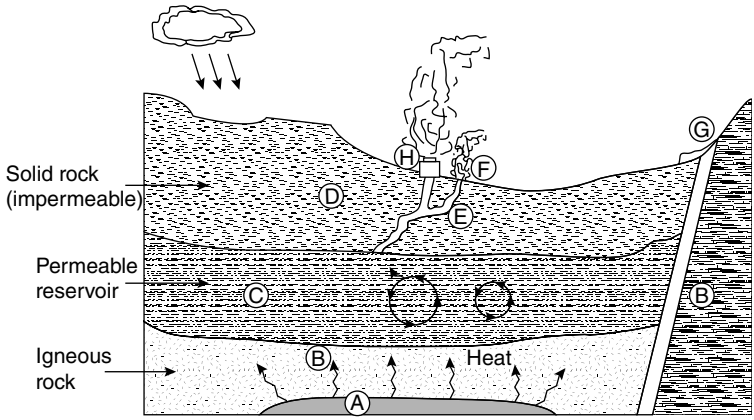


Fig. 1.8 *A Typical Geothermal Field: F-fumaroles; E-fissures; H-Well; G-Hot springs*

9. *Fuel cells*

These are electrochemical devices that are used for the continuous conversion of the portion of the free energy change in a chemical reaction to electrical energy. It operates with continuous replenishment of the fuel, and the oxidant at active electrode area and does not require recharging.

Main components of a cell are (i) a fuel electrode, (ii) an oxidant or an air electrode, and (iii) an electrolyte. Some of the fuel cells used are hydrogen, oxygen (H_2, O_2), hydrazine (N_2H_4, O_2), carbon/coal (C, O_2) methane (CH_4, O_2), etc.

Figure 1.9 shows a typical hydrogen–oxygen cell popularly known as a *hydrox cell*. It consists of two porous or permeable electrodes made up of either carbon or nickel immersed

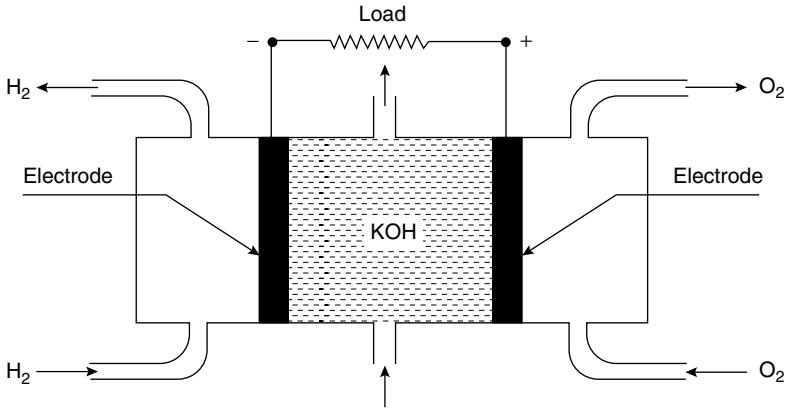


Fig. 1.9 *A Hydrox Cell*

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in an electrolyte of KOH solution. Since the electrochemical reaction at the porous electrode where gas, electrolyte and electrode in contact are slow, a catalyst (finely divided platinum or platinum such as material) is embedded in the electrodes. The concentration of KOH solution is maintained at about 30–40 per cent, since it has higher thermal conductivity and less corrosive compared to acids.

Working principle:
At the negative electrode, hydrogen gas is converted into hydrogen ions releasing free electrons.

$$H_2 \text{ Catalyst} \rightarrow 2H^+ + 2e^- \quad (1)$$

Under the influence of the catalyst in the electrode, the hydrogen ions react with the hydroxyl ions in the electrolyte to form water:

$$2H^+ + OH^- \rightarrow 2\,H_2O \quad (2)$$

When the cell is operating, the free electrons from negative electrode flow through the external load towards the positive electrode. The electrons interact with oxygen and water in the electrolyte to form negatively charged hydroxyl ions:

$$O_2 + H_2O + 2e^- \rightarrow 2OH^- \quad (3)$$

The hydrogen and hydroxyl ions then combine to form water:

$$H^+ + OH^- \rightarrow H_2O \quad (4)$$

A single hydrogen–oxygen cell can produce an electromagnetic force (emf) of 1.23 V at atmospheric pressure and at a temperature of 298 K. By combining the cells in series, it is possible to generate power ranging between a few kilo-watts to mega-watts.

Advantages

(i) Since power conversion is a direct process, the conversion efficiency is as high as 70 per cent.
(ii) It is pollution free when operated using hydrogen and operates with minimum noise.
(iii) It is compact in size and lighter in weight.
(iv) Maintenance cost is less due to lesser mechanical components.

Disadvantages

(i) It involves higher initial cost.
(ii) It has lower voltage.
(iii) It has lower service life.

10. *MHD generators*

MHD generators are used for direct conversion of thermal energy into electrical energy (Figure 1.10). They work on Faraday principle. When an electric conductor moves across a magnetic field, a voltage is induced in it, which produces an electric current. In MHD generators, the solid conductors are replaced by a fluid that is electrically conducting. The working fluid may be either an ionized gas or liquid metal. The hot, partially ionized and compressed gas is expanded in a duct, and forced through a strong magnetic field, electrical potential is

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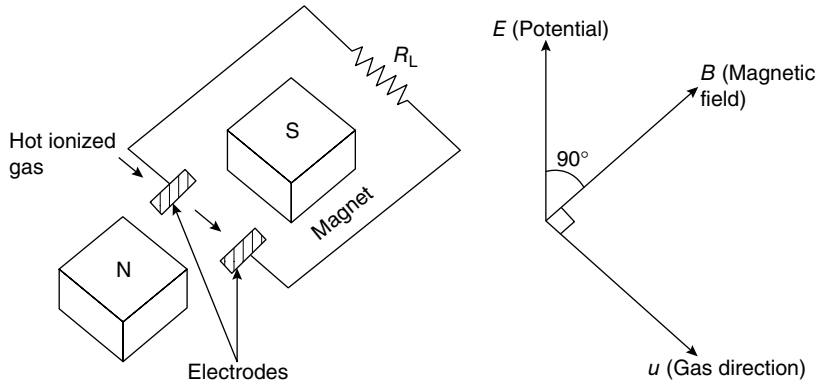


Fig. 1.10 *Principle of MHD Power Generation*

generated in the gas. Electrodes placed on the side of the duct pick-up potential generated in the gas. The direct current thus obtained can be converted into AC using an inverter.

The system is simple with large power and temperature-handling capacity without any moving parts. It is highly reliable and can be brought to full load within 45 sec. Power output can be changed from no load to full load in fraction of a second.

11. *Thermionic converter*

Figure 1.11 shows a thermionic convertor. It consists of two electrodes held in a container filled with ionized cesium vapour. Heating one electrode 'boils out' electrons that travel to the opposite colder side electrode. The positive ions in the gas neutralize the space-charge effect of the electrons that normally prevent the flow of electrons. Ionized gas offsets space-charge effect that tends to repel migration of electrons.

Electrons that are emitted by heating cathode are migrated to cooler anode collector and flow through outer circuit to develop electric power. Low-work function materials such as barium and strontium oxides are used for anodes, whereas high-work function materials such as Tungsten impregnated with barium is used for cathodes.

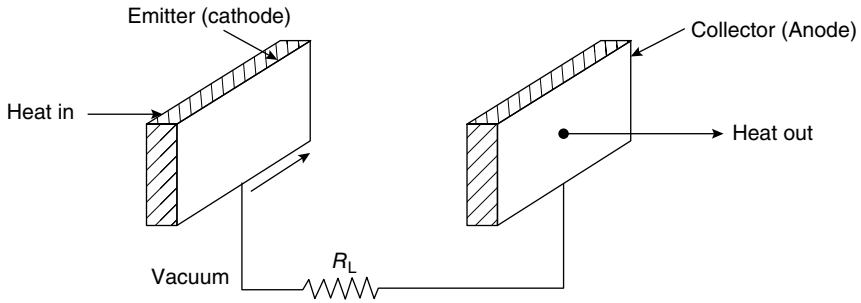


Fig. 1.11 *Thermionic Converter*

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12. *Thermoelectric power*

It is a device that converts heat directly into electric power. It eliminates the conversion of heat into kinetic energy of gas or steam flow. Its principle is based on Seebeck effect: If two dissimilar materials are joined to form a loop and the two junctions are maintained at different temperatures, an emf will be developed around the loop. The magnitude of the emf (E) developed by the above process is proportional to the temperature difference between the two junctions. Figure 1.12 shows a thermoelectric generator.

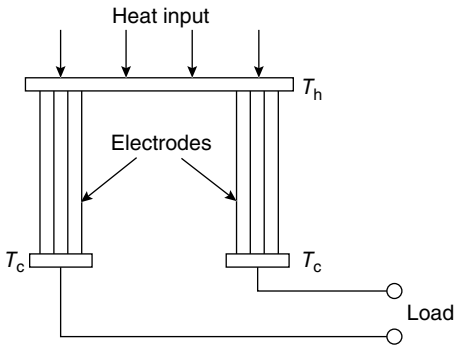


Fig. 1.12 *Thermoelectric Generator*

$$E \alpha (T_2 - T_1) = \alpha T_h - T_c)$$

where T_h = Temperature of hot junction; T_c = Temperature of cold junction; α = Seebeck coefficient.

The hot junction is maintained at a temperature T_h by the applied heat source that may be small oil or gas burner, a nuclear reactor, or direct solar radiation by paraboloidal concentrator, and the cold junction is maintained at T_c by either water cooling or radiative heat transfer.

The amount of power generated in a country depends on the utilization of natural resources available in that country apart from its geographical location. Total power generated is usually contributed by power generated by hydel plant, thermal power plant and nuclear power plant. The amount of power available by a hydel plant entirely depends on natural sites available and on the hydrological cycle in the area concerned. Since the amount of the rainfall varies considerably over years, it is difficult to run a hydel plant without sufficient water source. Thus for variable load conditions, a hydraulic power plant is not reliable. This difficulty is overcome by installing a thermal power plant. The advantage of steam power plant is that human beings can create the site near a water resource and fuel resource. The steam power plant can be used as a base load plant along with a hydel or nuclear power plant.

1.2 FACTORS AFFECTING SELECTION OF SITE

Selecting a proper site for a thermal power plant is vital for its long-term efficiency. Many factors have to be considered, deciding suitable location of a plant. As it may not be possible to satisfy all requirements in one place, an ideal location should contain an optimum mix of the feasible requirements, for long-term economic justification of the plant. In this section, some major factors that need to be considered while selecting the site for major power plants are listed below.

1.2.1 The Factors to be Considered for Site Selection of Steam Power Plants

The factors to be considered for site selection of steam power plants are but not limited to, nearness to load centre, supply of water, availability of coal, availability of land at a reasonable price, transportation facilities, availability of labors and distance from populated area.

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1. *Availability of coal*

As the power plant consumes large amount of coal, enough quantity must be available either in the vicinity, or it should be easily and economically viable to transport the coal to the power plant site. Proximity to sea route and rail transport are the major criteria.

2. *Transportation facilities*

A typical power plant with 1,000 MW capacity approximately consumes more than ten thousand tons of coal per day. Hence, the necessity for continuous supply and storage capability of coal in the power station by rail, by sea and by road must be ensured. It is also essential to have easy and enough access to transportation network during power plant construction and operation periods.

3. *Availability of land*

Ash is the main byproduct of combustion. Since the amount of coal used is large, proportionately, thousands of tons of ash is generated per day. Ash as it comes out hot from the boiler is hot and is very corrosive in nature. Disposing of such huge quantities of ash requires a large amount of empty space where it can be safely dumped.

In addition, enough space must be available for the storage of coal, disposal of ash, building of the power plant, township or residential colony of workers, markets, etc. Normally, for every MW of power generated, at least 3–4 acres of land available for this purpose.

4. *Supply of water*

Large amount of water is required for cooling and ash-handling systems in the power plant. Hence, it is essential to have water resources in the form of rivers, lake, sea, etc., are available nearby. During construction and operation stages also, different quantities of water are required. However, care should be taken to ensure that the site is at a reasonable distance from permanent and seasonal rivers and floodways.

5. *Availability of labours*

Apart from these, major requirements availability of skilled labour is also essential in the plant locality, as the semi- and unskilled labour requirements are very high. Proper transportation facility is very vital for the arrangement of labours.

6. *Geology and soil type*

The power plant should be built on a land that is not prone to vibrations generated by rotating equipments. Proper inspection and testing of soil and rock layers is essential. In addition, the site should be away enough from the faults and earthquake prone areas as weak and small earthquakes can damage many parts of a power plant intensively.

7. *Distance from populated area*

Location of the plant must be chosen in such a way that it has minimum or no ill impact on the surrounding environment. Plant location should be away from national parks, wildlife, protected areas, etc. As nearby forests, ecosystem, etc., are very sensitive to the pollutions caused by a power plant, ideal location should be away from them. Similarly, for the same reasons, the site should be away from population centres.

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8. *Nearness to load centre*

In general, the site should be near to the load centres to avoid transmission and distribution losses, capacity to decrease the amount of power loss and transmission expenses. It should be near to the local electric grid.

9. *Climatic conditions*

As climatic conditions such as temperature, humidity, wind direction and speed affect the productivity of a power plant, proper study must be done before installing the major components such as chimneys and cooling towers.

1.2.2 Factors Affecting Selection of Site for Hydro-Electric Power Plant

In a hydraulic power plant, several structures such as dam, conduits, intakes, surge tank and power house are involved. To select a site several alternative layouts are to be studied and the most economical one is adopted. The scope for the site investigation depends partly on the availability of existing published data, partly on the nature and size of the proposed plants and partly on the difficulties preset by the existing topographical and geological features, which in turn affects the number of alternatives that have to be studied. Site investigations are carried out is two main stages.

1. *Preliminary investigation* – to provide sufficient information to find out the practicability of the proposed scheme and to choose between alternatives.
2. *Final investigation* – detailed exploration of recommended site to establish the suitability and to enable the final design.

Investigations include the following:

(i) Hydrological
(ii) Topographical
(iii) Geological

(i) *Hydrological investigations*

(a) *Water availability*

Run-off data at the proposed site must be available beforehand. If not available calculation are based on the rainfall data.

(b) *Water storage*

Necessity to store the water for continuous power generation as there is a wide variation in rainfall during the year. Storage capacity and minimum quantity of water available are calculated with the help of mass curve. Two types of storage in use are as follows:

(1) Storage for one year only with no carryover for the next season full at the beginning empty at the end of the year.
(2) Enough storage sufficient even during worst dry periods.

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(c) *Water head*

It depends on topography. Increase in head reduces water to be stored and therefore the capital cost.

(d) *Ground water data*

Stability of ground slopes, amount and type of grounding required to prevent leakage.

(ii) *Topographical Investigation*

Topographical features of the whole catchment area should be mapped either by ground surveying or aerial surveying methods (for larger area and difficult terrain).

(iii) *Geological Investigation*

To provide most accurate picture of the ground character of the site. Selection depends in the following factors:

(a) Light basin of ample size
(b) Narrow outlet requiring a dam of less volume
(c) Strong foundation to support dam structure
(d) Safe and ample spill way to dispose surplus water
(e) Water should not submerge valuable mineral and agricultural land
(f) Availability of material for constructing a dam
(g) Distance from load centre to reduce the cost of creation of transmission lines and their maintenance
(h) Access to site by rail and road

(iv) *Consideration of water pollution effects*

Poor-quality water results in foul-smelling gases, corrosion of metallic structure and electrical equipment, overheating of machine parts. Water gets polluted by submerged vegetation mineral deposits, presence of algae. Leaves, twigs, logs under water decay and generate bad smelling gases. Thermal stratification and depletion of oxygen in lower strata generate harmful gases, such as CO_2, H_2S, CH_4, pollute the water.

(v) *Effects of water pollution*

(a) *Damage to dam structure and fitting*

Water containing H_2S, CO_2 and CH_2 is highly corrosive to dam structure. H_2SO_4 formed by oxidation of H_2S attack the cement and disintegrate the concrete. These gases increase the solubility of soil and rocks. Structures, metallic piping and penstock are also affected by the gases.

(b) *Damage of electrical machinery*

Presence of corrosive gases such as H_2S and SO_2 affect the voltage drop at brush contacts in generator leading to poor commutation. Silver contacts of relays get blackened and lead to high contact resistance and failure of relay occurs. Copper bus bars corrode.

(c) *Health hazard to operating personnel*

Headache, dizziness, dryness, nose, throat, chest pain are the effects of H_2S. High concentration of H_2S may even cause paralysis.

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(vi) *Sedimentation effect*

It reduces the capacity of reservoir and causes rapid erosion of turbine blades.

(vii) *Environment aspects of site selection*

These are compatible with the local environment site selection is to be made.

1.2.3 Factors Affecting Selection of Site for a Nuclear Power Plant

To locate a nuclear power plant geological, meteorological, hydrological topographical special transport, radiological investigations, are essential. Some of the major factors to be considered in locating a nuclear station are as follows:

(i) Availability of cooling water
(ii) Transportation facilities
(iii) Distance from load centre
(iv) Safety
(v) Radioactive waste and disposal facility
(vi) Foundation requirement

1.3 PRINCIPAL TYPES OF POWER PLANTS

The amount of power generated in a country depends on the utilization of natural resources available in that country apart from its geographical location. Total power generated is usually contributed by power generated by hydel plant, thermal power plant and nuclear power plant. The amount of power available by a hydel plant entirely depends on natural sites available and the hydrological cycle in the area concerned. Since the amount of the rainfall varies considerably over years, it is difficult to run a hydel plant without sufficient water source. Thus for variable load conditions, a hydraulic power plant is not reliable. This difficulty is overcome by installing a thermal power plant. The advantage of steam power plant is that human beings can create the site near a water resource and fuel resource. The steam power plant can be used as a base load plant along with a hydel or nuclear power plant. Figure 1.13 gives state-wise installed generating capacity of electricity in India.

Based on the data available, principal types of power plants in India are thermal, hydro and nuclear. The detailed working principle of these plants is discussed separately in the subsequent chapters.

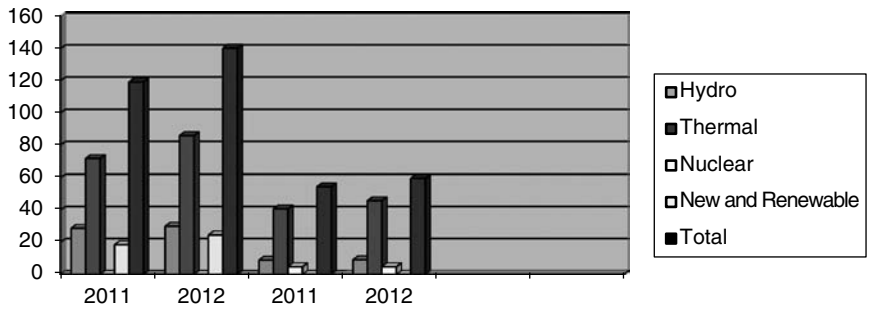


Fig. 1.13 *State Wise Installed Generating Capacity of Electricity in India*

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1.4 PRESENT STATUS AND FUTURE TRENDS

India has the fifth largest generation capacity in the world with an installed capacity of 236.38 GW as on year 2012[1], which is about 4 percent of global power generation. Captive power plants contribute to an additional power of 36.5 GW. Out of this, thermal power plants constitute 66 per cent of the installed capacity, hydroelectric about 19 per cent and rest being a combination of wind, small hydro-plants, biomass, waste-to-electricity plants and nuclear energy. India generated about 855 BU [1]electricity during 2011–2012 fiscal. As of March 2012, the per capita total consumption in India was estimated to be 879 kWh, which is However, this is fairly low when compared to that of some of the developed and emerging nations such the United States (~15,000 kWh) and China (~1,800 kWh). Taking into account the renewable energy utilization, especially wind energy, India's electricity sector is one amongst the world's most active countries. As on March 2012[1], India had an installed capacity of about 24.9 GW of new and renewable technologies-based electricity. During the Eleventh Five-Year Plan, nearly 55,000 MW of new generation capacity was created, but with an overall energy deficit of 8.7 per cent and peak shortage of 9.0 per cent. Resources currently allocated to energy supply are not sufficient for narrowing the gap between energy needs and energy availability.

India's energy contribution comes from all the resources available including renewable. The dominance of coal in the energy mix is likely to continue further in future. At present, about 54 per cent of the total installed electricity generation capacity is coal based, and 67 per cent of the capacity planned to be added during the Eleventh Five-Year Plan period 2007–2012 is coal based. Out of this, over 70 per cent of the electricity generated is from coal-based power plants. Other renewable energy sources such as wind, geothermal, solar and hydroelectricity contribute only 2 per cent share of the Indian fuel mix, whereas 1 percent share is contributed by Nuclear power.

The future trend indicates that by 2021–2022[1], share of coal and petroleum is expected to be about 66.8 per cent in total commercial energy produced and about 56.9 per cent in total commercial energy supply. The demand for coal is projected to reach 980 MT during the Twelfth-Plan period, whereas domestic production is expected to touch 795 MT in the terminal year (2016–2017)[1]. Even though the demand gap can be bridged through imports, domestic coal production will also need to grow at an average rate of 8 per cent compared to about 4.6 per cent in the Eleventh Five-Year Plan. The share of crude oil in production and consumption is expected to be 6.7 per cent and 23 per cent, respectively, by 2021–2022[1].

In 2011–2012[1], India was the fourth largest consumer in the world of crude oil and natural gas, after the United States, China and Russia. India's energy demand continued to rise despite slowing global economy. Owing to increased vehicle ownership, petroleum demand in the transport sector is expected to grow rapidly in the coming years for which India must rely on imports for a considerable amount of its energy use, particularly for crude petroleum.

Combustible renewable energy sources and waste constitute about one-fourth[1] of Indian energy use, which includes traditional biomass sources such as firewood and dung, used by more than 800 million Indian households for cooking.

To meet the future demand, the country needs quality or efficiency of the technology used in power generation. Energy exploration and exploitation, capacity additions[1], clean energy alternatives, conservation, and energy sector reforms will be crucial for energy security. Energy conservation has also emerged as one of the major issues in recent years. Conservation and

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efficient utilization of energy resources could dramatically bridge the gap between demand and supply of energy. Hence, thrust on improving energy efficiency is one of the most desirable options for bridging the gap during the short run.

1.5 LAYOUT OF STEAM, HYDEL, DIESEL, NUCLEAR AND GAS TURBINE POWER PLANTS

In this section, layouts of all the major types of power-generating plants are briefly explained.

1.5.1 Layout of Steam Turbine Plant

Figure 1.14 shows a general layout of coal-fired thermal power station. The layout consists of four important systems. These are as follows:

(i) Coal- and ash-handling system
(ii) Air and flue gas system
(iii) Cooling water system
(iv) Feed water and steam generation system

A steam power plant works on *Rankine cycle*. Steam generated in the boiler expands in the *steam turbine*. Low-pressure exhaust steam from the turbine is condensed in the *condenser*. The condensate is fed back to the boiler along with feed water.

A thermal power plant mainly uses hot products of combustion known as *flue gas* to generate steam in a steam generator. Heat is generated by burning coal, oil or gas. For this purpose, boiler furnace is designed for firing coal, air or gas.

High-pressure steam generated in the steam boiler is passed through steam turbine where steam expands in the nozzles. Heat energy of steam is converted into mechanical energy. Mechanical energy is utilized to run the generator, and hence to generate electric power. The low-pressure steam after expanding in the turbine enters the condenser. Here, steam condenses by rejecting heat to the cooling water. The condensed steam is supplied back to the boiler using condensate extraction pump. The overall efficiency of the plant is increased by using economizer, air preheater, super heater and feed water heaters.

A brief explanation of different flow paths of a power plant is given below.

1. *Coal- and ash-handling system*

In this system, coal from the coal yard is crushed to the required size in coal crusher. The *crushed* coal is supplied to the coal bunker in the boiler house. This *coal* is fed to the furnace through coal *hoppers* where it is burnt. The products of combustion, namely ash and un-burnt coal, are disposed to the ash storage site.

2. *Air and flue gas system*

In this system, air from the air heater is supplied to the boiler furnace using *forced draught* fan. The products of combustion in the form of flue gases are circulated in the furnace to heat the water in the tubes. The heat in the exhaust flue gases is utilized by passing it first over

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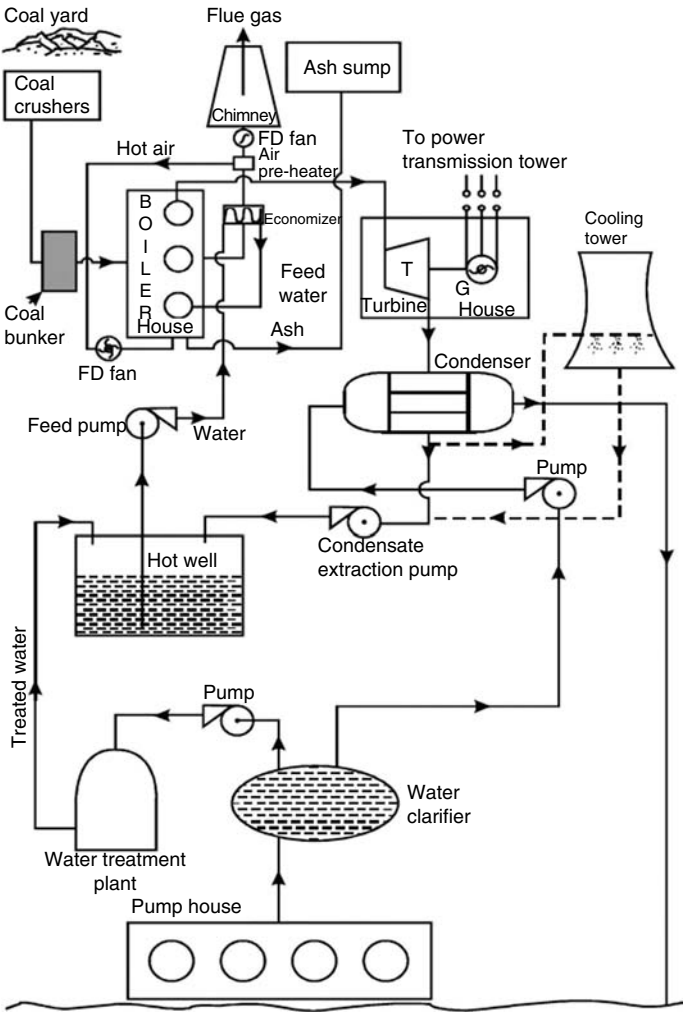


Fig. 1.14 *Coal-Fired Thermal Power Station Showing All Working Systems*

economizer tubes and then over air *preheater* tubes. After extracting maximum possible heat, the flue gases are discharged through the chimney using an *induced draught* fan.

3. *Cooling water system*

This system mainly consists of a *condenser, condensate extraction pump, cooling tower* and *cooling water circulation pump*. When plenty of water is not available, a cooling tower is preferred. Water pumped from the water clarifier plant is circulated through the condenser tubes in order to condense the exhaust steam coming from the turbine. Water leaving the condenser is discharged to the downward side of the river. Such a system is known as *open system*

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that is very useful when abundant water supply is available. When there is scarcity of water, a cooling tower is installed. This system is also known as a *closed system*. In this system, the condensate water is cooled by passing it through the cooling tower. The cold water is pumped to the hot well. The loss of water due to evaporation is compensated by supplying water directly from the river.

4. *Feed water and steam generation system*

This system mainly consists of *feed water pump, steam generator, condenser* and *hot well*. Feed water from the hot well is supplied to the steam generator by using the feed pump. Steam generated in the boiler is superheated in super heater tubes. Superheated steam is supplied to the prime mover to develop power. The steam exhausted in the prime mover is condensed in the condenser. Any water that is lost in the system is compensated by supplying feed water from an external source. Such feed water should be treated in a feed water treatment plant before sending it to the hot well.

1.5.2 Layout of Hydro-Electric Plant

Water is a great source of energy as it is available in abundance and is quite cheap. The potential energy of water at high levels is utilized for the generation of electrical energy. The generating stations that obtain energy from water are called hydro-electric power stations.

The general layout of the plant is decided by the type of the hydraulic plant. In the case of vertical turbines, the most convenient and economical layout will be placing turbines in a line parallel to the length of the turbine house. The spacing between the machines is determined by the size of the scroll case, width of the fume or by the overall diameter of the alternators. If the machines are horizontal, the most suitable layout will be arranging the turbines at right angles to the length of the turbine house or even can replaced parallel to the longitudinal axis of the turbine house. A repair bay should be provided at one end of the turbine house close to the workshop and sufficient space should also be provided for the dismantling and re-erection of machines. Figure 1.15 shows a typical layout of a water plant with vertical machines.

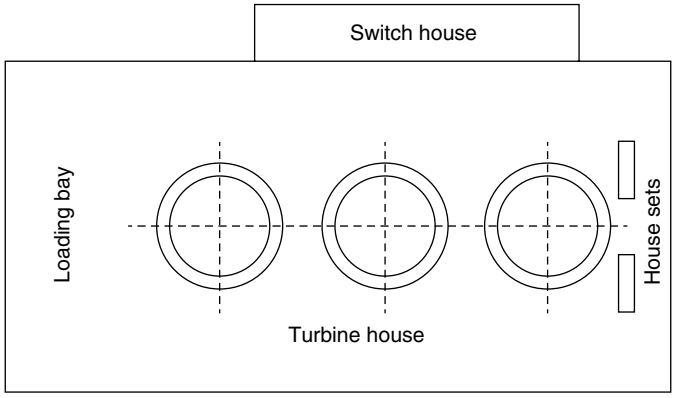


Fig. 1.15 *Layout of Hydro-Electric Plant*

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1.5.3 Plant Layout of Diesel Engine Plant

The layouts of diesel power plants for high capacity (50 MW and above) is shown in Figure 1.16. Generally, in all three cases, the generating units are installed parallel with parallel centre lines. Sufficient space must be provided around various units for dismantling and repairing purposes. Fuel storage tanks may be located outside the building to avoid fire hazards. Some space should be left for future expansion. The engine room should be provided with sufficient ventilation. The plant must include a number of instruments for the guidance of operators to ensure reliability, economy and safety.

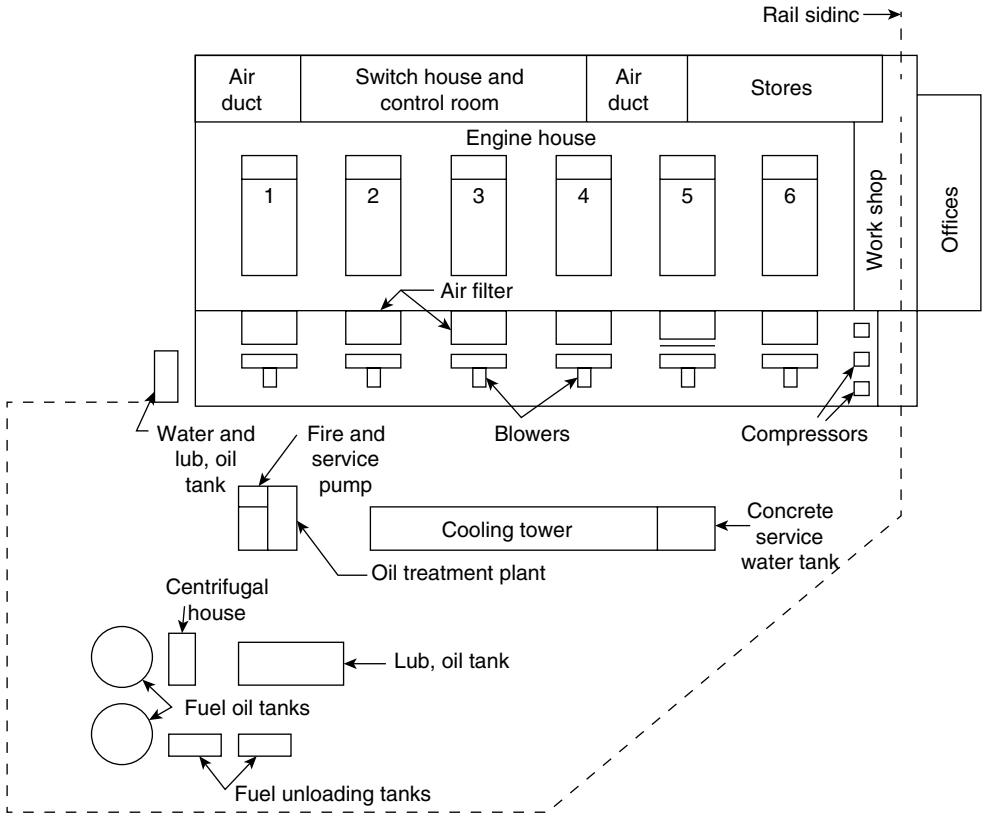


Fig. 1.16 *Layout of Diesel Power Plant for High Capacity (50 MW and Above)*

1.5.4 Layout of a Nuclear Plant

Figure 1.17 shows the layout of a typical nuclear power plant, where 1 indicates the reactor building, 2 the group 1 service building, 3 the group 2 service building, 4 the reactor auxiliary building, 5 the maintenance building, 6 the turbine building, 7 the administration building, 8 the pump house and 9 the group 2 pump house.

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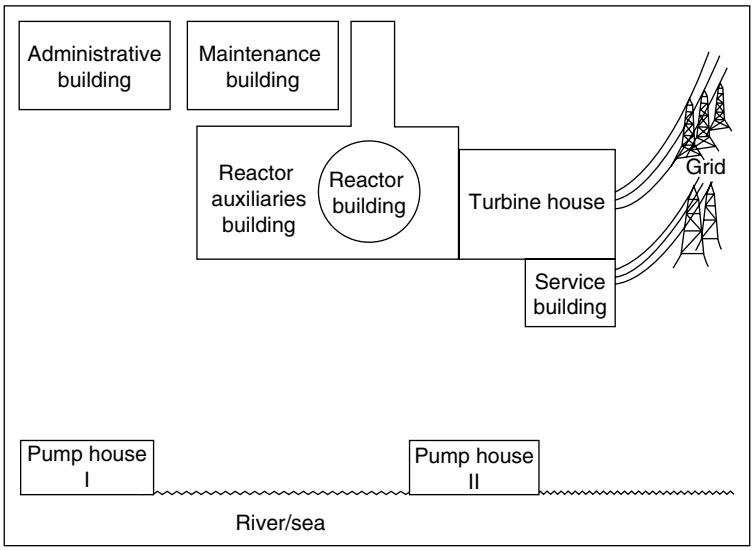


Fig. 1.17 *Layout of a Nuclear Plant*

A typical layout of a nuclear power station – is shown in Figure 1.18.

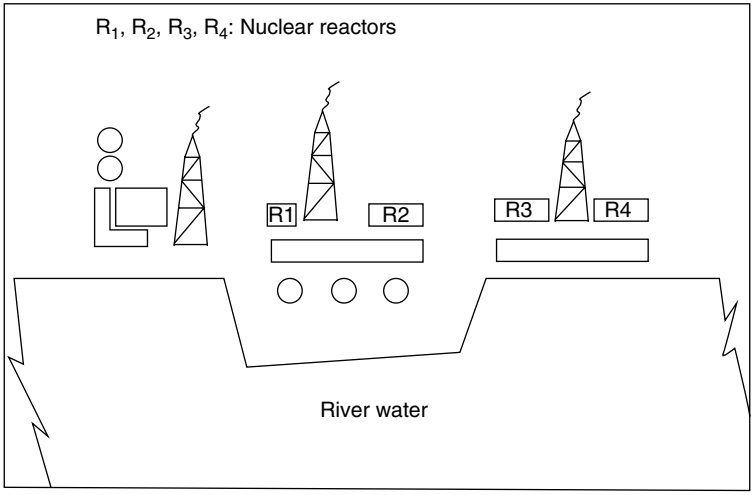


Fig. 1.18 *A Typical Layout of a Nuclear Power Station*

1.5.5 Layout of Gas Turbine Plant

Figure 1.19 shows typical layout of a gas turbine plant. The air filler is used to clean the air. The air leaving the air filler flows to the LP compressor. The compressed air then flows through the inter-cooler and then it enters the high-pressure compressor. The air leaving the air compressor

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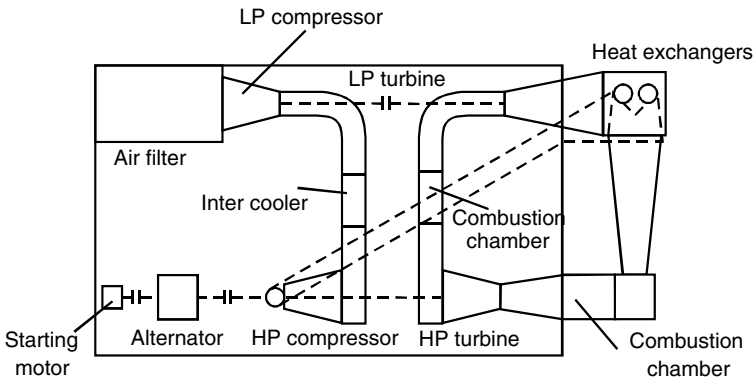


Fig. 1.19 *Layout of a Gas Turbine Plant*

flows through the heat exchanger and the hot air flows to the combustion chamber. Products of combustion then enter in to low-pressure turbine, where it expands doing useful work.

1.6 COMBINED POWER CYCLES – COMPARISON AND SELECTION

Simple cycle gas turbine (GTs) plants using natural gas and fuel oil operate at around 33 per cent and 25 per cent efficiency, respectively, major part of the heat being wasted as thermal energy in the hot exhaust gases. Combining of multiple thermodynamic cycles to generate power, overall plant efficiency can be increased up to 60 per cent by using a heat recovery steam generator (HRSG). The HSRG captures heat from high-temperature exhaust gases to generate steam, which is then supplied to a steam turbine to generate additional electric power. The cycle working under this principle is known as combined cycle. In most of the instance, a combined cycle power plant utilizes gas turbines in conjunction with a steam turbine and is called a combined cycle gas turbine (CCGT) plant. Different configurations of CCGT power plants are in use where each GT has its own associated HRSG, and multiple HRSGs supplying steam to one or more steam turbines. For example, in a plant with a 3×1 configuration, three GT/HRSG trains supply to one steam turbine. Similarly there can be 1×1, 2×1, or 4×1 arrangements also. The steam turbine capacity is decided to match the number and capacity of supplying GTs/HRSGs. This type of power plants is being installed in increasing numbers round the world where there is access to substantial quantities of natural gas.

Combined cycle principles of operation

The HRSG is basically a heat exchanger popularly known as a boiler, or comprises a series of heat exchangers. It generates steam for the steam turbine exchanging heat from the hot exhaust gas flow from a gas turbine through heat exchanger tube banks. The tubes are arranged in sections, or modules also known as economizers, evaporators, superheater/reheater and preheaters. The fluid circulation in HRSG could be either natural or forced circulation using pumps. Saturated steam from the steam drums or once-through system is passed through superheater or reheater tubes to superheat the steam. The superheated steam produced by the HRSG is

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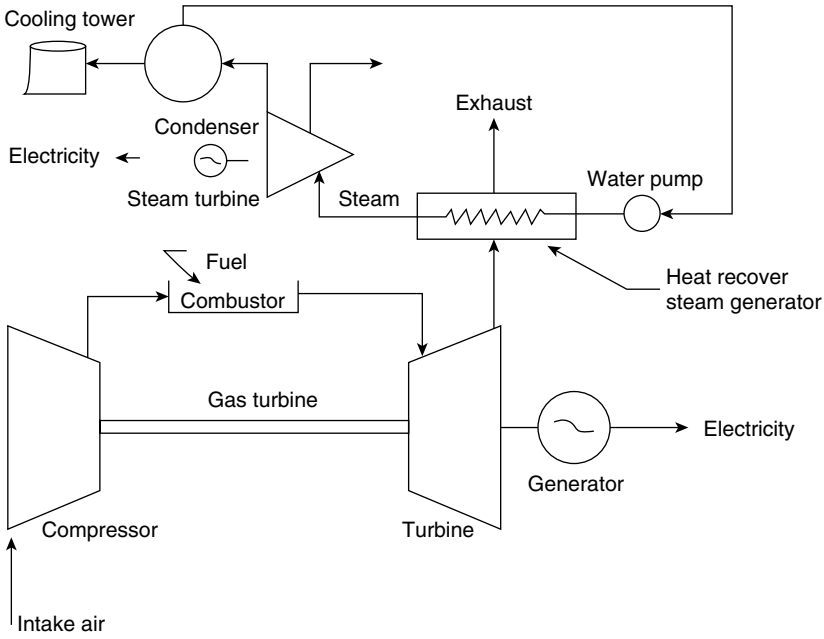


Fig. 1.20 *Combined Cycle Power Plant*

supplied to the steam turbine where it expands through the turbine blades, imparting rotation to the turbine shaft. The energy delivered to the generator coupled to the drive shaft is converted into electricity. After exiting the steam turbine, the steam condenses in a condenser and fed back to the HRSG. Figure 1.20 shows a schematic diagram of a combined cycle gas turbine plant.

1.7 MERITS OF STEAM, GAS, DIESEL, HYDRO AND NUCLEAR POWER PLANTS

In this section, merits, de-merits and relative merits of steam, gas, diesel, hydro and nuclear power plants are briefly discussed.

1.7.1 Advantages and Disadvantages of the Gas Turbine Plant

Advantages over diesel plants

(i) Due to perfect balancing of gas turbine unit, the gas turbine plants are subjected to fewer vibrations.

(ii) The mechanical efficiency of gas turbine plant is as high as 95 per cent, whereas the mechanical efficiency of diesel plant is less.

(iii) The torque characteristics of turbine plants are far better than diesel plants as it is a continuous power developing system.

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(iv) The work developed per kilogram of air is large compared with diesel plant as the expansion of gases to atmospheric pressure is possible.

(v) The weight of gas turbine plant is hardly 0.15 kg/HP, whereas the weight of diesel plant is 2.5 kg/HP. Therefore, the space required and capital cost of the gas turbine plant is considerably less than diesel plant.

(vi) All the parts of gas turbine (compressor, combustion chamber and turbine) can be designed and tested individually, and they can be arranged as per requirements as the compression, combustion and expansion take place in different units.

(vii) The running speed of the turbine (4,000–100,000 rpm) is considerably large compared with diesel engine (1,000–2,000 rpm).

(viii) The lubrication and ignition systems are more simplified compared with diesel plant.

(ix) The specific fuel consumption does not increase with time in gas turbine plant as rapidly as in diesel plants.

(x) The installation and maintenance costs are less than diesel plants.

(xi) The exhaust of gas turbine is free from smoke as the quantity of air supplied is four times greater than theoretically required for complete combustion.
Smokeless combustion has been achieved by the gas turbine with proper control of primary zone mixture ratios and methods of secondary air introduction and recirculation could produce reduced pollution conditions. It is claimed that the exhaust gases coming out are smokeless without increasing the production of nitric oxide emissions.

(xii) Any poor-quality fuels can be used in gas turbine plant, whereas special-grade fuels are required for diesel engines to avoid knocking. This is a great advantage of gas turbine plant.

The disadvantages of gas turbine plants over diesel plants can be summarized as easier maintenance, improved reliability, lower initial cost, smaller plant dimensions and less space requirements for same output, less vibrations, absence of cyclic variations and greater starting torque.

The disadvantages of gas turbine plants over diesel power plants include poor part load efficiency, requirement of special metals and alloys for different components of the plants, special cooling methods for cooling the turbine blades, low starting torque and less life.

Advantages over steam power plant

(i) The handling of ash is major problem in steam plants that are completely eliminated in open-cycle gas turbine plants using gas or liquid as fuel.

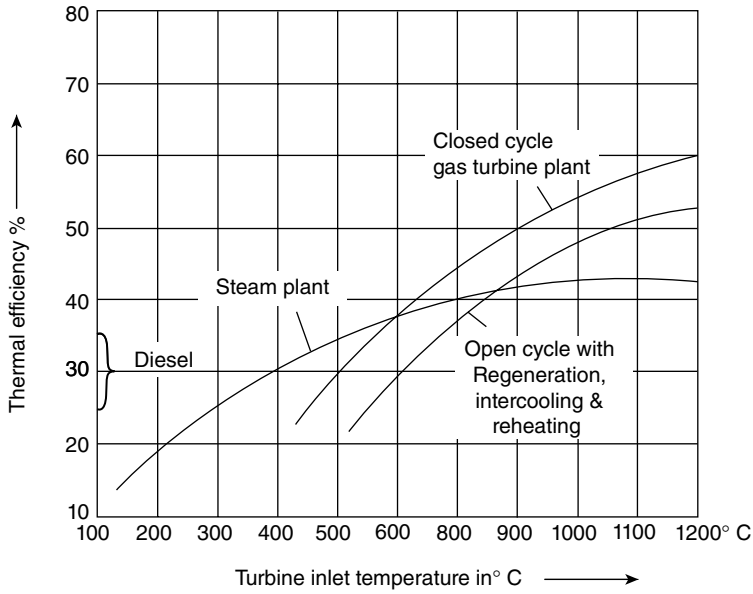
(ii) The cubic capacity of the buildings required for gas turbine plant is about 50 per cent of steam plant. The total weight of the materials required for gas turbine plant is also 50–60 per cent of steam plant. Therefore, considerable saving in capital cost is possible having same efficiency as steam plant. The gas turbine plants can be installed at selected load centers as space requirement is considerably less where steam plant could not be accommodated.

(iii) The site of the steam power plant is dictated by the availability of large cooling water, whereas an open-cycle gas turbine plant can be located near load centers as no cooling

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water is required. The cooling water required for closed cycle gas turbine plant is hardly 10 per cent of the steam plant.

(iv) The comparison of thermal efficiency is shown in figure below. It is obvious from the figure that the gas turbine plant should be operated above 550°C to compete with the steam plants. Above 375°C, the gap between the steam cycle and gas turbine cycle efficiency widens. Above 550°C, the efficiency of the gas turbine plant increases three times as fast as the steam cycle efficiency for a given temperature increase.



(v) The ratio of exhaust to inlet volume for the same pressure and initial temperature conditions would be only 3.95 in case of gas turbine plant as against 250 for steam plant.

(vi) Steam plant developing power at 17 per cent efficiency costs equal to the gas turbine plant developing power at 34 per cent. This being the case, the gas turbine plant as a base load plant has little future. This is economical as base load plant only where the fuel oil is available at a considerably cheaper rate.

(vii) The gas turbine plant uses fewer auxiliaries than the steam turbine plant; therefore, small size of the gas turbine components enables complete work-tested units to be transported to the site.

(viii) The ease and rapidity with which the output of gas turbine plants can be made available from a cold start permit a reduction in the capacity of steam plant held in reserve in a hot condition and needed to cover for short-load prediction errors and random load fluctuations. The gas turbine plants peak-up the load hardly within 15 min. This loading response can be put to good effect in the event of an unscheduled disconnection of a large generator.

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(ix) The gas turbine plants are always desirable as peak-load plants, irrespective of the cost due to their good response, but its adoption as base load is justified on the economic ground. It can be used as a base load plant only where the gas turbine fuel costs are considerably less.

(x) In general, gas turbines can be built relatively quicker and require much less space and civil engineering works and water supplies.

(xi) The components and circuits of a gas turbine plant can be arranged to give the most economic results in any given circumstances that are not possible in the case of steam power plants.

(xii) The gas turbine becomes more economical for operating conditions below a given load factor as saving on the capital charges outweighs the additional cost of fuel.

(xiii) The gas turbine plant as peak-load plant is more preferable as it can be brought on load quickly and surely. The fuel consumption is of secondary importance, since the time of operation is limited.

Advantage and disadvantages of gas turbine plant
Advantages

(i) It is smaller in size and weight compared to an equivalent steam power plant. For large capacities, it is smaller in size than a comparable diesel engine plant. Hence, it is most suitable where size and weight are the main considerations, namely in ships, aircraft engines, locomotives, etc.

(ii) It has low initial cost and low operating costs compared to steam power plant of same capacity.

(iii) It is available at relatively short delivery times and is quick to install and put to use.

(iv) It is quick starting and smooth running.

(v) It offers flexibility in supplying process needs such as compressed air in addition to electric power and in using a range of liquid and gaseous fuels and even low-calorific-value fuels.

(vi) Water requirement is less compared to a condensing steam power plant.

(vii) No stand by losses as in a steam power plant in which boilers are operated even when turbine in not loaded.

(viii) Easy to lubricate the compressor, turbine bearings and maintenance cost in less.

(ix) Plant does not require heavy foundations and building and subject to fewer environmental restrictions.

(x) Due to higher operating temperature, thermal efficiency is more.

Disadvantages

(i) Overall cycle efficiency is less as a portion of power generated by the turbine is consumed by the compressors.

(ii) The plant is incompatible with solid fuels.

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(iii) It is costlier due to the use of high heat-resistant materials and sophisticated manufacturing processes for blade manufacture.

(iv) It runs at comparatively higher speeds, and hence requires a reduction gear for normal industrial applications.

1.7.2 Advantages and Disadvantages of the Nuclear Plant

Advantages of nuclear power plant

(i) Less space requirement

(ii) Consumes very small quantity of fuel

(iii) Involves less fuel transportation cost

(iv) Does not require large fuel-storage facility

(v) Reliable in operation

(vi) Not effected by weather conditions

Disadvantages

(i) High initial cost, maintenance cost

(ii) Not suited for varying load condition

(iii) Radioactive wastes should be disposed carefully

(iv) Requires trained operators

1.7.3 Advantages and Disadvantages of Diesel Plants

Advantages and disadvantages of diesel plants are as follows:

(i) It has no standby losses and can burn fairly wide range of fuels.

(ii) Manufacturing periods are short; therefore, plant expansion can be done just by adding the generating units of required size.

(iii) Diesel plants can be fully automated with reasonably low capital costs, namely for starting, synchronizing and shutting down diesel sets, to overcome scarcity of skilled labour.

(iv) Maintenance can be simplified by the provision of easily replaceable assemblies of parts, thus enabling reconditioning to be undertaken away from generating plants.

(v) Can be installed in remote, cramped or difficult to access areas as the unit is compact, light weight, available in smaller capacity.

(vi) The diesel plants can be located very near to the load centers, many times in the heart of the town.

(vii) They can be used in all climatic zones, and they are very adaptable and can be easily extended to the given power requirements.

(viii) Machine sets are readily available as standard sets in the range of 500–40 MW.

(ix) Due to its relatively low cost, it is possible to install a spare unit which can be kept as permanent standby.

(x) Operational reliability with high availability is ensured.

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Disadvantages

(i) The diesel plants are not economical where fuel has to be imported.

(ii) The noise is a serious problem in diesel plant.

(iii) Selected types of fuels are required in diesel engines, whereas there is more mobility in case of thermal plant.

(iv) The lubrication cost is high.

Diesel plants vs. Steam power plants

(i) The diesel power plants are more efficient than steam power plants in the range of 150 MW capacity. The operating efficiency is high in the load range of 50–100 per cent of full load.

(ii) The first cost diesel plants are lesser than steam plants for plant capacity up to about 7 kW. Above this capacity, the diesel cost rises rapidly while that of steam plants continues to fall.

(iii) It can be quickly started up and brought into service (within 1 min).

(iv) The cooling water required for the same capacity is considerably less in diesel plants than the thermal plant. The site of the thermal plant is dictated by the availability of cooling water in many cases, whereas diesel plant can be located without giving much importance to the availability of cooling water. The water ratio for thermal to diesel is approximately 50:1.

(v) The storage required for the fuel is considerably less than the thermal and it can be handled more easily.

(vi) There is no problem of ash handling as there is practically no refuse.

(vii) The space required for diesel plant is considerably less than thermal plant, and therefore the cost of foundation and buildings is less.

(viii) A diesel engine lubrication system is more economical compared to a steam plant and permits the use of specially compounded cylinder lubricants that provide dramatic improvement in wear rates.

(ix) A Diesel engine plant layout is very simple compared with a thermal plant. They are easy to operate and control also.

(x) The unit capacity of diesel engine is considerably smaller than the thermal unit. The cost of unit increases with an increase in unit capacity for diesel plant. However, the cost of the unit goes on decreasing in case of thermal plant with an increase in unit capacity.

(xi) The repair and maintenance costs of diesel plants are generally much higher than for steam plants. These costs are more or less fixed in case of steam plants and more or less are proportional to output in the case of diesel plants.

(xii) Life of 25–30 years is normal for thermal plant, whereas the life of diesel plant is hardly 2–5 years or less.

(xiii) The diesel plants are not suitable for continuous operation under overloads, whereas steam plants can work under 25 per cent overload continuously.

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1.7.4 Advantages of Hydro-Electric Power Plants

Advantages

The hydroelectric power plants are the best from many viewpoints. These are simple in design, easy for maintenance, pollution free with zero fuelling cost. Hydro-generation has a unique and significant role to play particularly in the operation of interconnected power system. In fact, thermal or nuclear power can seldom produce electrical energy economically without an adequate proportion of hydro power generation. Some of the advantages of hydroelectric power plants are mentioned below:

(i) The operating cost of the hydro-electric plant including auxiliaries is considerably low when compared with thermal plants. The annual operating and maintenance cost of a thermal plant is approximately five to six times that of hydro-plant of equal capacity.

(ii) Cost of power generation is less. The cost of each unit of electrical energy delivered from a thermal power plant is very adversely affected by the reduction of load factor at which the plant is operated. The cost per kilo-watt generated by a thermal plant designed for 80 per cent load factor would go up by 20 per cent if operated at 60 per cent load factor, and by 65 per cent if operated at 40 per cent load factor. The cost of generation by hydro-electric plant is more or less unaffected by the load factor.

(iii) Life expectancy of a hydro-electric power plant is more. The useful life of a thermal plant is 20–25 years as against 100–125 years for hydro-plant.

(iv) There is no problem of handling the fuel and ash and no nuisance of smoke exhaust gases and soots, and there are no health hazards due to air pollution.

(v) The fuel needed for the thermal plants has to be purchased; whereas in hydro-plant, the fuel cost is totally absent. The security of thermal power plant is mostly dependent on transport facilities, and its security is less when the fuel is imported from outside countries.

(vi) Hydro-electric plants are quick to respond to the change of load compared with thermal or nuclear plant. Full output can be reached in about 10–15 sec. Therefore, they are ideally suited for peaking purposes. The rapid fluctuating loads are served most economically by hydro-plants.

(vii) The machines used in hydel plants are more robust and generally run at low speeds at 300–400 rpm, whereas the machines used in thermal plants run at a speed 3,000–4,000 rpm. Therefore, there are no specialized mechanical problems or special alloys required for construction. There are no complications due to high temperature and pressures. Therefore, the outages of the machines are very few.

(viii) Flexible and economical in operation. Hydel power plays a major role not only providing an economic source of energy but also in improving the operation of the combined power system by meeting the system load at various times and providing regulating functions.

(ix) The efficiency of hydro-plant does not change with age, but there is considerable reduction in efficiency of thermal as well as nuclear power plant with age.

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(x) In a hydroelectric plant, there is no stand-by losses, whereas these are unavoidable for thermal plants, and the number of operations required is considerably small compared with thermal power plants.

(xi) It does not contribute to air and water pollution to greenhouse effect.

(xii) Usually, the hydro-stations are situated away from the developed areas, therefore the cost of land is not a major problem.

Disadvantages

(i) The capital cost (cost per kilo-watt capacity installed) of the hydro-plant is considerably more than thermal plant.

(ii) It takes considerably long time for its erection compared with thermal plants.

(iii) Power generation by the hydro-plant is only dependent on the quantity of water available, which in turn depends on the natural phenomenon of rain. The dry year is more serious for the hydroelectric project.

(iv) The site of hydro-electric station is selected on the criterion of water availability at economical head. Such sites are usually away from load centers. The transmission of power from power station to the load centre requires long transmission lines. Therefore, investment required for long transmission lines and loss of power during transmission is unfavorable factors for the economical selection of hydro-plants.

1.8 RESOURCES AND DEVELOPMENT OF POWER IN INDIA

1.8.1 Coal and Lignite

In India, coal deposits are mainly confined to eastern and south central parts of the country. The states of Jharkhand, Odisha, Chhattisgarh, West Bengal, Andhra Pradesh, Maharashtra and Madhya Pradesh account for more than 99 per cent of the total coal reserves in the country. As on 31 March 12, the estimated reserves of coal was around 293.5 billion tons, as against the total estimated reserve of coal of around 285.86 billion tons as on 31 March 11 (Figure 1.21; Figure 1.22).

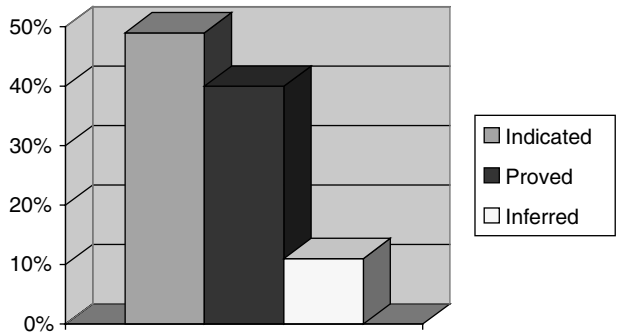


Fig. 1.21 *Estimated Reserve of Coal in India as on 31 March 2012*

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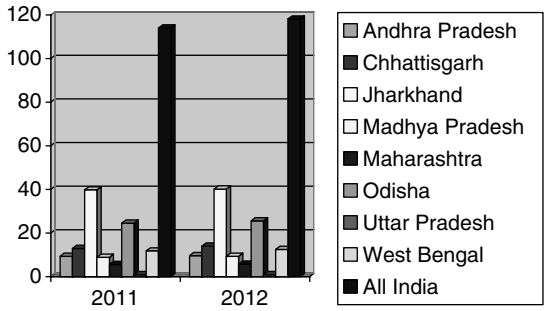


Fig. 1.22 *State-Wise Estimated Reserve of Coal (in Billion Tonnes) in India as on 31 March 2011 and 31 March 2012*

The above figure shows the state wise estimated reserve of coal in billion tonnes (proved) in India as on 31 March 2011 and 31 March 2012 as per the information available from Office of Coal Controller, Ministry of Coal, Government of India from major coal mines across different states of India.

The estimated reserve of lignite as on 31 March 2012 was 41.96 billion tonnes against 40.91 billion tonnes as on 31 March 2011.

According to another estimate from Ministry of Coal, Government of India, proved estimate of lignite coal amounting to 14.73 billion tonnes (as on 31 March 2012) and an indicated estimate of another 61.4 billion tonnes was available. The following bar chart provides the proved lignite reserve in India from some of the major mines across different states (Reserve in Billion tonnes).

The estimated reserve of lignite as on 31 March 12 was 41.96 billion tonnes against 40.91 billion tonnes as on 31 March 11 (Figure 1.23).

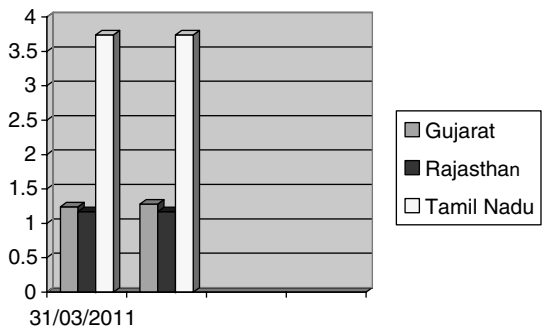


Fig. 1.23 *State-Wise Estimated Reserve of Lignite Coal (in Billion Tonnes) in India as on 31 March 2011 and 31 March 2012*

The increase in the estimated reserve of lignite during the year 2011–2012 was 1.22 per cent, Tamil Nadu accounting for the maximum increase of 2.99 per cent.

1.9 PETROLEUM AND NATURAL GAS

The estimated reserves of crude oil in India as on 31 March 2012 stood at 759.59 million tonnes (MT) (see Table 1.2; Figure 1.24).

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Table 1.2 *State-Wise Estimated Reserve of Crude Oil and Natural Gas in India as on 31 March 2011 and 31 March 2012*

	Crude petroleum (million tonnes)				Natural gas (billion cubic metres)			
	31.03 2011		31.03 2012		31.03 2011		31.03 2012	
States/UTs	Estimated reserves	Distribution (%)	Estimated reserves	Distribution (%)	Estimated reserves	Distribution (%)	Estimated reserves	Distribution (%)
Andhra Pradesh	5.22	0.69	5.59	0.74	40.58	3.18	42.30	3.18
Arunachal Pradesh	3.39	0.45	3.41	0.45	2.21	0.17	2.15	0.16
Assam	170.33	22.49	172.54	22.71	139.61	10.93	139.82	10.51
CBM	0.00	0.00	0.00	0.00	87.63	6.86	97.54	7.33
Eastern Offshore[1]	26.18	3.46	24.12	3.18	438.03	34.27	462.03	34.73
Gujarat	137.43	18.14	135.72	17.87	78.96	6.18	78.19	5.88
Nagaland	2.69	0.36	2.69	0.35	0.12	0.01	0.12	0.01
Rajasthan	75.33	9.95	68.37	9.07	12.04	0.94	12.13	0.91
Tamil Nadu	8.43	1.12	8.86	1.17	36.88	2.89	39.30	2.95
Tripura	0.08	0.01	0.07	0.01	33.09	2.59	36.05	2.71
Western Offshore[2]	328.27	43.34	337.72	44.46	408.88	31.99	420.63	31.62
Total	757.40	100.00	759.59	100.00	1278.06	100.00	1330.26	100.00

[1] Includes JVC/Pvt parties for crude oil includes West Bengal for natural gas.
[2] Includes Bombay High offshore. Rajasthan and JVC for crude oil and Bombay High offshore. Rajasthan and Madhya Pradesh (coal bed methane or CBM) for natural gas.
Notes: @ Proved and indicated balance recoverable reserves.
Source: Ministry of Petroleum and Natural Gas.

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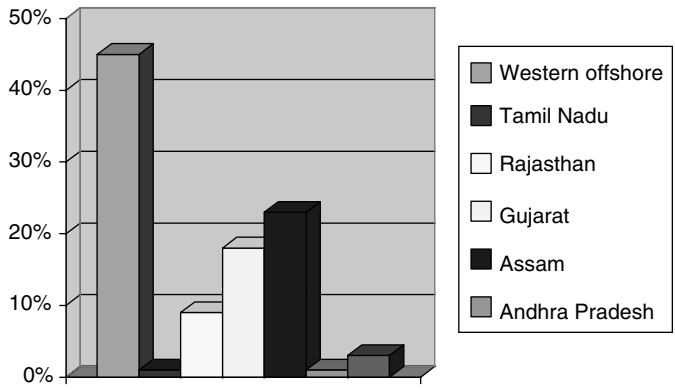


Fig. 1.24 *Estimated Reserve of Crude Oil in India as on 31 March 2012*

Geographical distribution of crude oil indicates that the maximum reserves are in the Western offshore (44.46%) followed by Assam (22.71%), whereas the maximum reserves of natural gas are in the Eastern offshore (34.73%) followed by Western offshore (31.62%). There was an increase of 0.29 per cent in the estimated reserve of crude oil for the country as a whole during 2011–2012. There was an increase of estimated crude oil reserves by 7.09 per cent in Andhra Pradesh followed by Tamil Nadu (4.48%). The estimated reserves of natural gas in India as on 31 March 2012 reached at 1330.26 billion cubic metres (BCM) (see Figure 1.25; Table 1.3), as per the open source information available on Ministry of Petroleum and Natural Gas website, the state wise estimated reserve of natural gas in India as on 31 March 2012, is depicted in the following bar chart. The figures indicate billion cubic meters of gas.

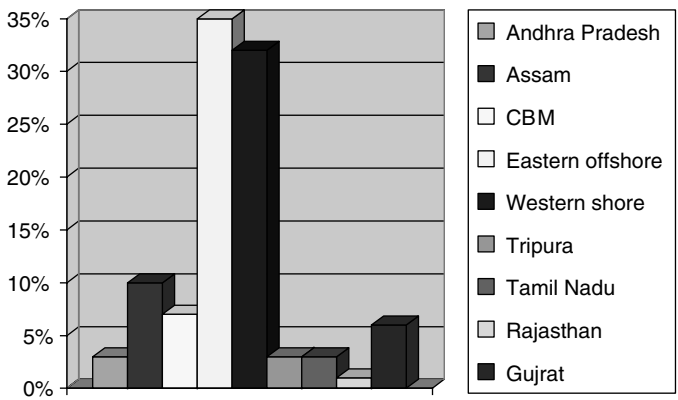


Fig. 1.25 *Estimated Reserve of Natural Gas in India as on 31 March 2012*

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Table 1.3 *Source-Wise and State-Wise Estimated Potential of Renewable Power in India as Reserve of Crude Oil and Natural Gas in India as on 31 March 2012*

States/UTs	Wind potter	Small hydro power	Bio-mass potter	Cogenera-tion bagasse	Waste to energy	Total	
						Estimated reserves	Distribu-tion (%)
1	2	3	4	5	6	7	8
Andhra Pradesh	5394	560	578	300	123	6955	7.75
Arunachal Pradesh	201	1334	8	0	0	1543	1.72
Assam	53	239	212	0	8	512	0.57
Bihar	0	213	619	300	73	1205	1.34
Chhattisgarh	23	993	236	0	24	1276	1.42
Goa	0	7	26	0	0	33	0.04
Gujarat	10609	197	1221	350	112	12489	13.91
Haryana	0	110	1333	350	24	1817	2.02
Himachal Pradesh	20	2268	142	0	2	2432	2.71
Jammu & Kashmir	5311	1418	43	0	0	6772	7.54
Jharkhand	0	209	90	0	10	309	0.34
Karnataka	8591	748	1131	450	151	11071	1133
Kerala	790	704	1044	0	36	2574	2.87
Madhya Pradesh	920	804	1364	0	78	3166	3.53
Maharashtra	5439	733	1887	1250	287	9596	10.69
Manipur	7	109	13	0	2	131	0.15
Meghalaya	44	230	11	0	2	287	0.32
Mizoram	0	167	1	0	2	170	0.19
Nagaland	3	197	10	0	0	210	0.23
Odisha	910	295	246	0	22	1473	1.64
Punjab	0	393	3172	300	45	3910	4.36
Rajasthan	5005	57	1039	0	62	6163	6.87
Sikkim	98	266	2	0	0	366	0.41
Tamil Nadu	5374	660	1070	450	151	7705	8.58
Tirpura	0	47	3	0	2	52	0.06
Uttar Pradesh	137	461	1617	1250	176	3641	4.06

(*Continued*)

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Table 1.3 (*Continued*)

States/UTs	Wind potter	Small hydro power	Bio-mass potter	Cogenera-tion bagasse	Waste to energy	Total	
						Estimated reserves	Distribu-tion (%)
1	2	3	4	5	6	7	8
Uttaranchal	161	1577	24	0	5	1767	1.97
West Bengal	22	396	396	0	148	962	1.07
Andaman & Nicobar	2	7	0	0	0	9	0.01
Chandigarh	0	0	0	0	6	6	0.01
Dadar & Nagar Have	0	0	0	0	0	0	0.00
Daman & Diu	0	0	0	0	0	0	0.00
Delhi	0	0	0	0	131	131	0.15
Lakshadweep	16	0	0	0	0	16	0.02
Puducherry	0	0	0	0	3	3	0.00
Other's*	0	0	0	0	1022	1022	1.14
All India total	49130	15399	17533	5000	2707	89774	100.00
Distribution (%)	54.73	17.15	19.54	5.57	3.02	100.00	

Notes: *Industrial waste.
Source: Ministry of New and Renewable Energy, Government of India.

1.10 PRESENT STATUS OF POWER GENERATION IN INDIA

According to the available information from central electricity authority (see Figure 1.26), all India gross electricity generation from utilities, excluding that from the captive generating plants, was 55,828 gigawatt-hours (GWh) during 1970–1971. It rose to 9, 23,203 GWh during the year 2011–2012. The production of electricity from utilities has increased from 8, 44,846 GWh during 2010–2011 to 9, 23,203 GWh during 2011–2012, registering an annual growth rate of about 9.27 per cent. From the table abstract taken by CEA, it is clear that out of the total electricity generation in the country, from utilities and non-utilities taken together (10, 51,375 GWh for the year 2011–2012), 7, 59,407 GWh was generated from thermal (72.2%), 1, 30,510 GWh (12.4%) was from hydro and 33,286 GWh (3.16%) was generated from nuclear sources.

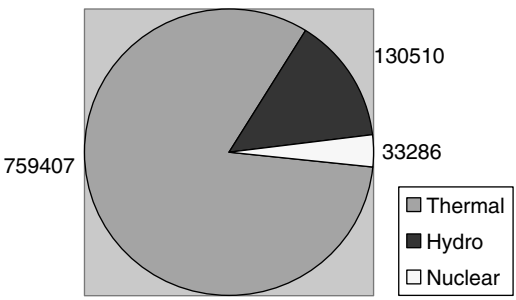


Fig. 1.26 *All India Gross Electricity Generation from Utilities (Numbers Indicate Giga Watt Hour)*

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1.11 ROLE OF PRIVATE AND GOVERNMENT ORGANIZATION

Just after independence, India had a power-generating capacity of 1,362 MW, and generation and distribution of electrical power was carried out by private utility companies. Power availability was only in a few urban centers, whereas rural areas and villages did not have electricity.

Subsequently post independence, all new power generation, transmission and distribution in the rural sector and the urban centers, not coming under the purview of private utilities came under state and central government agencies. State electricity boards (SEBs) were formed in all the states, thereafter.

National Thermal Power Corporation (NTPC), National Hydro-electric Power Corporation (NHPC) and Power Grid Corporation Limited (PGCL) were constituted by the government to assist in meeting the increasing demand for electricity throughout the country. In India, both state and central governments participate in the power sector's development. The Ministry of Power in the central government formulates the policies for the power sector and the Central Electricity Authority (CEA) also functions as a regulatory authority. As per government guidelines, all power projects above a certain capacity have to obtain techno-economic clearance from CEA before they can be implemented. Apart from this, Ministry of Non-Conventional Energy Sources focuses on renewable energy sources to augment the generation capacity of electrical power.

The liberalized government policy announced in 1991 and consequent amendments in Electricity (Supply) Act have opened doors to private sectors to investment in electricity industry. Considerable emphasis has been given to attract private investment and the major policy changes have been announced by the government in this regard[1]

(i) The Electricity (Supply) Act, 1948, amended in 1991, provides for creation of private generating companies to set up power generating facilities and selling the power in bulk to the grid or other persons.

(ii) Financial environment for private sector units was modified to allow liberal capital structuring and an attractive return on investment. Up to hundred percent (100%) foreign equity participation can be permitted for projects set up by foreign private investors in the Indian Electricity Sector[1].

(iii) Administrative and legal environment simplifies the procedures for clearances of the projects.

(iv) Policy guidelines for private sector participation in the renovation and modernization of power plants issued in 1995.

Present scenario

At present in power sector nearly half that is, 49.15 per cent is under SEBs, which is followed by public sector units, such as NTPC, NHPC, and NPCIL, etc. The role of private sector is about 19.85 per cent, the least among the three sectors. However, of late, private sector is growing at a greater pace than the other two sectors.

The major companies in these three sectors are listed below:

Public sector units in power sector

Major players of energy in public sector are listed below:

(i) Power Finance Corporation

(ii) Central Electricity Regulatory Commission

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(iii) Damodar Valley Corporation
(iv) Neyveli Lignite Corporation Limited
(v) Nuclear Power Corporation of India Limited
(vi) NHPC Limited
(vii) NTPC Limited

Private company in power sector

The following companies play a major role in private sector:

(i) Adani Power
(ii) Tata Power Company Ltd.
(iii) Suzlon
(iv) Reliance Power Limited
(v) JSW Energy

State electricity board in power sector

In addition to the above, the following list provides the names of the companies in state sector:

(i) Assam Electricity Board
(ii) Arunachal Pradesh Electricity Board
(iii) Andhra Pradesh Electricity Board
(iv) Chhattisgarh Electricity Board
(v) Delhi Electricity Board
(vi) Goa Electricity Board
(vii) Gujarat Electricity Board
(viii) Haryana Electricity Board
(ix) Himachal Pradesh Electricity Board
(x) Jammu & Kashmir Electricity Board
(xi) Jharkhand State Electricity Board
(xii) Karnataka State Electricity Board
(xiii) Kerala State Electricity Board
(xiv) Madhya Pradesh Electricity Board
(xv) Orissa Electricity Board
(xvi) Punjab State Electricity Board
(xvii) Rajasthan Electricity Board
(xviii) Tamil Nadu Electricity Board
(xix) UP Electricity Board
(xx) Uttarakhand Electricity Board
(xxi) West Bengal State Electricity Board

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1.12 STATE-LEVEL SCENARIO, LOAD SHEDDING

The following chart (Figure 12.7) shows the state wise renewable energy production in MW.

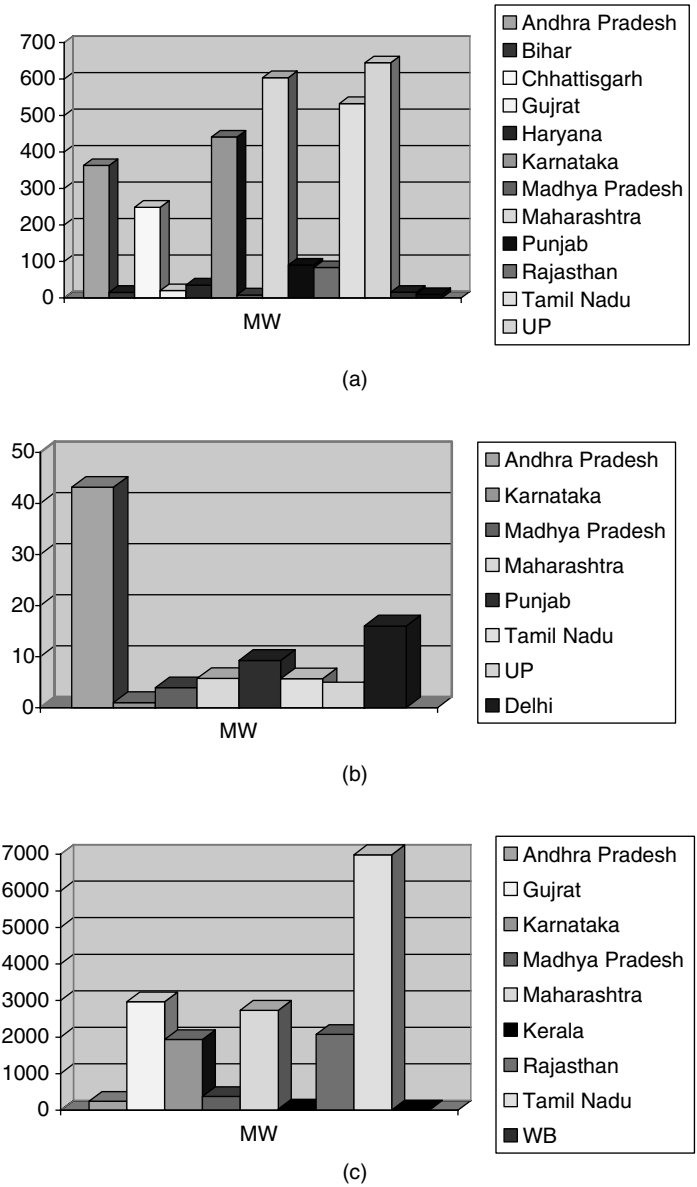


Fig. 1.27 *(a) Bio-mass, (b) Waste to Energy, (c) Wind Power*

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1.13 CARBON CREDITS

In 1992, several countries came together to create United Nations Framework Convention on Climate Change (UNFCCC). This compelled the countries to work together to limit emissions of greenhouse gases that cause an increase in global temperature and climate change, and to deal with the resulting impacts.

In 1995, UNFCCC adopted the Kyoto Protocol held in Kyoto, Japan. The protocol has a legally binding emission reduction mechanism for a specific period that was to be respected by the developed countries. The period of first commitment ran from 2008 to 2012. The duration of second period began on 1 January 2013, which will end in 2020. A total of 195 nations across the globe (parties) participated in the UNFCCC and 191 nations (parties) ratified the Kyoto Protocol. In 2010, nations (parties) agreed to limit the global temperature increase to below 2°C. To reduce current or future emissions of greenhouse gases, organizations in these countries are free to engage in various environment-friendly activities or 'methodologies' (voluntary or non-voluntary).

Based on the Kyoto Protocol, an authority such as government department decides the cap or quota of pollution that a group or organization can cause. Groups and organizations that pollute below this quota are eligible to sell their balance quota as carbon credits certificates (or just credit) to another organization that has already gone or could go beyond the assigned cap. Under the Kyoto Protocol, the caps are issued by National registry as 'assigned amounts' or 'Kyoto units'. Companies can choose to donate their allowance to a non-profit group, while availing tax deductions. Transfers and acquisitions of these units are tracked and recorded through the 'registry systems'. An 'international transaction log' based in Bonn, Germany, verifies and ensures secure transfer of emission reduction units between countries keeping the rules in mind.

United Nation's Kyoto Protocol created by the United Nations set limits in the form of caps/quota for parties. With it, the United Nations also introduced a 'carbon market' governed by *three mechanisms* to assist organizations reach the emission reduction targets.

As per the United Nations, 'The Kyoto mechanisms:

(i) Stimulate sustainable development through technology transfer and investment.

(ii) Help countries with Kyoto commitments to meet their targets by reducing emissions or removing carbon from the atmosphere in other countries in a cost-effective way.

(iii) Encourage the private sector and developing countries to contribute to emission reduction efforts.'

The three mechanisms by UN's Kyoto protocol:

(i) IET or International Emissions Trading

Under the Kyoto Protocol, Article 17, countries have accepted to reduce the emissions to meet the targets committed to UN. These targets refer to allowed emissions or assigned emissions. The allowed emissions are further broken into 'assigned amounts units (AAUs).' Countries those are able to save units assigned to them can sell them to countries that are over their assigned targets. IET created a new trading platform called, carbon market.

Another unit that may be transferred under the scheme, each equal to one ton of CO_2 is a Removal Unit (RMU), Article 3.3, 3.4, which is based on the land use, land-use change and forestry (LULUCF) activities such as reforestation. RMUs are issued by national registry.

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(ii) JI or Joint Implementation
Developed countries working with developed countries are allowed to earn Emission Reduction Units (ERU). Under the Kyoto Protocol, Article 6, a developed country that is committed to reduce or limit its greenhouse gases' emissions can implement or run or sponsor projects or activities that are environment friendly, in a developed country. This allows for technology transfers and foreign investments. It also enables both the parties to earn carbon credits.

(iii) CDM or clean development mechanism – Developed countries working with developing countries can earn certified emission reductions (CER) equaling a reduction of one ton of carbon dioxide equivalent. CERs are issued by CDM registry.

Under the Kyoto Protocol, Article 12, a *developed* country that has committed to reduce or limit its greenhouse gases' emissions can implement or run or sponsor projects or activities that are environment friendly, in a *developing* country, enabling for technology transfers and foreign investments. Thus, the organizations in developed countries to earn CER credits and meet the caps or quota set under Kyoto Protocol. If an organization is able to earn more CER credits than it requires, it can even sell them to other groups.

Developing countries do not have restrictions to emit gases because such countries go through the same phase of development as developed countries went through years ago. But by participating in the CDM project, they assist in controlling the damage already done by the developed nations.

Reserves of ERUs, CERs, AAUs and/or RMUs have to be maintained in the national registries of respective countries to ensure that parties do not 'oversell' units.

Eligibility requirements for participating in carbon credit trading

UN offers accreditation to environment-friendly projects that deliver more than what is expected of them under normal working conditions. There is a mechanism to calculate the difference between emission reductions that would have occurred under normal circumstances and additional reduction in emissions due to special efforts.

According to the United Nations:

'To participate in the mechanisms, Annex I Parties must meet, among others, the following eligibility requirements:

(i) They must have ratified the Kyoto Protocol.

(ii) They must have calculated their assigned amount in terms of tons of CO_2-equivalent emissions.

(iii) They must have in place a national system for estimating emissions and removals of greenhouse gases within their territory.

(iv) They must have in place a national registry to record and track the creation and movement of ERUs, CERs, AAUs and RMUs and must annually report such information to the secretariat.

(v) They must annually report information on emissions and removals to the secretariat.'

The exception to the eligibility is nuclear-powered energy and cancelling or stopping a deforestation project.

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India being a developing country has no emission targets to be followed but can enter into CDM projects. India has generated approximately 30 million carbon credits and approximately 140 million in run, the second highest transacted volumes in the world. India's carbon market is growing faster than even information technology, bio technology and BPO sectors as 850 projects with a huge investment of Rs 650,000 million are in pipeline. As per the Prime Minister's Council on Climate Change, the revenue from 200 projects is estimated at Rs. 97 billion till 2012. India has been able to register approximately 350 projects spread across various sectors with major dominance of renewable energy, energy efficiency and biomass energy projects. Carbon is being traded on India's multi-commodity exchange and has become first exchange in Asia to trade carbon credits.

1.14 QUESTIONS

1.14.1 Objective Questions

1. The commercial sources of energy are
   (a) solar, wind and biomass
   (b) fossil fuels, hydropower and nuclear energy
   (c) wood, animal wastes and agriculture wastes
   (d) None of the above
2. The main source of production of biogas is
   (a) human waste (b) wet cow dung
   (c) wet livestock waste (d) All of the above
3. India's first nuclear power plant was installed at
   (a) Tarapore (b) Kota
   (c) Kalpakkam (d) None of the above
4. In fuel cell, the following energy is converted into electrical energy.
   (a) Mechanical (b) Chemical
   (c) Heat (d) Sound
5. Solar thermal power generation can be achieved by
   (a) using focusing collector or heliostats
   (b) using flat plate collectors
   (c) using a solar pond
   (d) any of the above system
6. The energy radiated by sun on a bright sunny day is approximately
   (a) 700 W/m^2 (b) 800 W/m^2
   (c) 1 kW/m^2 (d) 2 kW/m^2
7. The overall efficiency of thermal power plant is equal to
   (a) Rankine cycle efficiency
   (b) Carnot cycle efficiency
   (c) Regenerative cycle efficiency
   (d) boiler efficiency × turbine efficiency × generator efficiency

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8. Rankine cycle efficiency of a good steam power plant may be in the range of
   (a) 15 – 20 percent (b) 35 – 45 percent
   (c) 70 – 80 percent (d) 90 – 95 percent
9. A steam power station requires space
   (a) equal to diesel power station
   (b) more than diesel power station
   (c) less than diesel power station

1.14.2 Review Questions

1. Draw a general layout of a steam power plant showing the different circuits and systems and explain them.
2. Discuss the relative advantages and disadvantages of a gas turbine plant versus steam and diesel engine plants.
3. With the help of block diagram, draw the layout of the principal power plants.
4. Discuss the various factors to be considered for the site selection of
   (i) steam power plant
   (ii) gas turbine plant
   (iii) hydro-electric plant
   (iv) nuclear Plant
5. What are the different sources of energy? Classify them.
6. Discuss the possibilities of various renewable energy sources as potential contenders for power generation.
7. Discuss principal types of power plants, their special features and applications.
8. Write a short note on present status and future trends of power generation under Indian context.
9. How cycle efficiency can be increased in combined power cycles?
10. Discuss briefly on resources and development of power in India.
11. Elaborate on present status of power generation in India.
12. What is the role played by private and government organizations in power sectors?
13. What do you understand by carbon credits? Discuss.

Answers

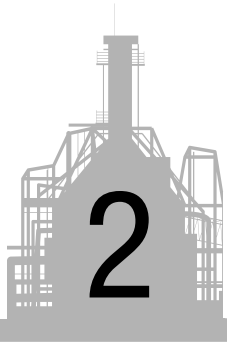
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Fuels and Combustion



2

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2.1 Introduction
2.2 Classification of fuels and different types of fuels used for steam generation
2.3 Calorific values of fuels
2.4 Combustion of fuels
2.5 Properties of coal, indian coals
2.6 Selection of coal in thermal power station
2.7 Questions

2.1 INTRODUCTION

A fuel is a combustible substance that generates heat when burnt in the presence of oxygen. Carbon and hydrogen are the main constituents of a fuel. In addition to these, fuel also contains sulphur, oxygen and nitrogen. Based on the fuel, the percentage of carbon (C) ranges from 50 to 95 per cent, hydrogen (H_2) 2 to 6 per cent, oxygen (O_2) 2 to 4 per cent, sulphur (S) 0.5 to 3 per cent and nitrogen (N_2) 5 to 7 per cent. A solid fuel may also contain 2 to 30 per cent ash. In a fuel, *carbon, hydrogen and sulphur* are *combustible elements*, whereas *nitrogen* and *ash* are *incombustible elements*. Since the major combustible elements in the fuel are carbon and hydrogen, a fuel is also known as a *hydrocarbon fuel*.

When a fuel is burnt in the presence of oxygen, it generates energy in the form of heat. This heat energy is converted into mechanical energy, which can be used for electrical power generation in thermal power plants and for propelling ships, automobiles and locomotives, etc. This chapter deals with fuels and combustion.

2.2 CLASSIFICATION OF FUELS AND DIFFERENT TYPES OF FUELS USED FOR STEAM GENERATION

Based on the occurrence and the state, fuels are classified as follows:

(i) *Based on the occurrence*, fuels are classified as *primary* and *secondary fuels*. *A primary fuel*, also called as a *natural fuel*, is available in nature, for example, wood, peat, lignite, natural gas, petroleum, etc. *A secondary fuel*, also known as an *artificial fuel*, is not

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available in nature, but it is prepared from a primary fuel by artificial means, for example, gasoline, diesel, coke gas, blast furnace gas, kerosene, etc.

(ii) *Based on the state*, fuels are classified as follows: solid, liquid and gaseous.

2.2.1 Solid Fuels

These are either *natural solid fuels* or *artificial solid fuels*. Natural solid fuels are wood, peat, lignite, bituminous coal and anthracite coal. Artificial solid fuels are charcoal, coke, briquetted coal and pulverized coal.

1. *Natural solid fuels*

(i) *Wood*

It is mainly used as a domestic fuel. It has a low calorific value (CV) of about 10,500 kJ/kg. It consists of about 48.5 per cent C, 6 per cent H_2, 1.5 per cent ash, 43.5 per cent O_2 and 0.5 per cent N_2.

(ii) *Peat*

It is partially carbonized and decomposed material formed mainly due the transformation of buried vegetation. It is considered as the primary stage in the formation of coal. It is a spongy substance with high moisture content (up to 80 per cent). Hence, it has to be dried before use. In its dry form, peat has a CV of about 14,650 kJ/kg. It consists of 58 per cent C, 6.3 per cent H_2, 30.8 per cent O_2, 0.9 per cent N_2 and 4 per cent ash.

(iii) *Lignite*

It is considered as the secondary stage in the formation of coal. It is brown in colour and therefore known as *brown coal*. It is used as a low-grade fuel, particularly for domestic use and also as a boiler fuel. This brown coal is an intermediate stage between bituminous coal and peat. In India, it is chiefly available in Neyveli, Tamil Nadu. It has a CV of about 12,500 kJ/kg. It consists of 66 per cent C, 5 per cent H_2, 20 per cent O_2, 1 per cent N_2, 1 per cent S and 3.5 per cent ash and other incombustibles.

(iv) *Bituminous coal*

It is most commonly used form of coal. It is available both in *caking* and *non-caking* forms. It burns with a yellow, smoky flame. It has a CV of about 32,000 kJ/kg. It consists of 81 per cent C, 5 per cent H_2, 8 per cent O_2, 1.5 per cent N_2, 1 per cent S and 3.5 per cent ash. It is mainly found in Bihar, Madhya Pradesh, Orissa and West Bengal.

Caking bituminous coal softens and forms a pasty mass when heated at higher temperature. This coal is used to manufacture coke.

Non-caking bituminous coal does not get softened on heating, and therefore it is used in steam power plants.

(v) *Anthracite coal*

It is considered as the final form of transformation of buried vegetation into coal. It is of the non-caking form and has the highest CV of about 35,500 kJ/kg. It burns at high temperature

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and is smokeless. It consists of 91 per cent C, 3 per cent H_2, 2.5 per cent O_2, 0.5 per cent N_2, 0.5 per cent S and 2.5 per cent ash. It is also used as a boiler fuel. In India, it is mainly found in Jammu and Kashmir.

2. *Artificial solid fuels*

(i) *Wood charcoal*

It is prepared by the strong heating of wood in a limited supply of air, below 2800°C. It contains 80–90 per cent of carbon. It is used for domestic purpose, namely cooking and heating and also in some metallurgical processes.

(ii) *Coke*

It is prepared by heating bituminous coal at 5000–7000°C continuously for 15–18 hours in a closed chamber in the absence of air. During this process, the volatile matter is removed from the coal, leaving it dull black in colour. Coke is a hard, porous structure and is smokeless. It contains 85–95 per cent of carbon. When the bituminous coal is heated between 9000 and 10,000°C, *hard coke* is produced. It is used in gas-producing plants and blast furnaces.

(iii) *Briquetted coal*

It is prepared by moulding finely ground coal under pressure with or without the addition of any binding agent. The most commonly used binding materials are coal tar, crude oil and clay. The loss of fuel through boiler grates can be avoided by using briquetted coal.

(iv) *Pulverized coal*

It is prepared by powdering low-grade coal with high ash content. Dry coal is pulverized in pulverizing mills. It burns most efficiently when mixed with air, and hence it is used in most of the thermal power generation plants.

Advantages and disadvantages of solid fuels
Advantages

(i) They can be stored in a convenient place without any risk of spontaneous explosion.
(ii) They can be easily transported.
(iii) They have a low ignition temperature.
(iv) They involve low production cost.

Disadvantages

(i) They have large percentage of ash content.
(ii) The rate of combustion of solid fuels cannot be easily controlled.
(iii) They produce large amount of smoke.
(iv) The necessity of various fuel-handling equipments (conveyers, crushers, hoppers, bunkers, etc.) increases the cost of fuel handling.
(v) They have a lower CV when compared with liquid fuels.
(vi) The amount of heat wasted is more due to unburnt and partially burnt fuel.
(vii) They have the tendency to form *clinkers* at higher temperatures.
(viii) They require excess amount of air for complete combustion.

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2.2.2 Liquid Fuels

Liquid fuels are available in nature in the form of crude petroleum. Petroleum is a mixture of hydrocarbons that can be grouped into the following four categories:

1. Parffins (C_nH_{2n+2})
2. Olefins (C_nH_{2n})
3. Naphthalenes (C_nH_{3n})
4. Aromatics (C_nH_{4n-2}).

All commercial liquid fuels are derived from crude petroleum or crude oil. The crude oil, as it comes out from oil wells, contains 83–87 per cent of carbon, 10–14 per cent of hydrogen and small percentages of sulphur, nitrogen, oxygen and gases containing mainly ethane and methane.

The impurities present in the crude oil are eliminated by the process of refining oil by *distillation*. It works on the principle that the boiling points of various hydrocarbons increase with an increase in molecular weight; and when distilled, the crude oil is separated into gasoline, kerosene, fuel oil or reduced crude oil.

Following liquid fuels are obtained during the process of distillation.

1. *Gasoline*

It is also known as petrol. It is the lightest liquid petroleum fraction and is highly volatile in nature. Gasoline has a specific gravity ranging from 0.70 to 0.78. It is obtained between 65 and 200°C, and it is a mixture of a number of hydrocarbons. It is light yellow in colour and is used as a fuel in automobiles and aeroplanes.

2. *Kerosene*

It is heavier than gasoline but less volatile when compared to it. It has a specific gravity ranging from 0.78 to 0.85. It is obtained between 150 and 300°C. It is mainly used in aviation gas turbines as jet fuel and also in lamps and stoves.

3. *Distillate*

It is slightly heavier than kerosene and is used in tractors and as domestic fuel.

4. *Diesel*

It is heavier than kerosene. It has a specific gravity ranging from 0.86 to 0.95. It is obtained between 200 and 370°C. It is used in diesel engines.

5. *Fuel oil*

It is similar to diesel but has wider composition than diesel. It is used in furnaces of oil-fired boilers.

In addition to the natural fuels discussed earlier, some artificial liquid fuels are also used.

Alcohol is an artificial fuel obtained from vegetable matter by the process of fermentation. It is used in many chemical processes.

Advantages and disadvantages of liquid fuels
Advantages

(i) They have a higher CV when compared to solid fuels.

(ii) They require very less excess air for complete combustion due to uniform mixing and atomization of fuels.

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(iii) They can be stored and handled easily when compared to solid fuels.
(iv) They posses higher combustion efficiency due to uniform mixing of fuel and air.
(v) Combustion process and combustion rate can be easily controlled by varying the quantity of fuel.
(vi) They require less storage space when compared to solid fuels.
(vii) They burn without forming ash, thus eliminating ash disposal problem.
(viii) Oil-fired combustion systems are much cleaner than coal-fired combustion systems.
(ix) They involve lower handling cost, i.e. lower operation and maintenance cost.

Disadvantages

(i) Leakage loss during transportation and handling.
(ii) Highly volatile and easily vapourized.

2.2.3 Gaseous Fuels

Gaseous fuels are either *natural fuels* or *artificial fuels*. Natural gaseous fuels occur in the form of natural gas near oil-bearing areas under the earth's surface. Artificial gaseous fuels prepared are coal gas, coke oven gas, blast furnace gas, producer gas, water gas and Mond gas.

1. *Natural gas*

It is found dissolved in petroleum or in large quantity under the earth's surface in oil- and gas-bearing areas. It mainly contains 85 per cent of methane (CH_4) and hydrogen (H_2), along with small percentages of ethane and ethylene. Liquefied petroleum gas (LPG) is a mixture of propane, butane and other hydrocarbons separated from natural gas. It is stored under pressure in liquid form. It is used as a domestic fuel and as fuel in automobiles.

2. *Coal gas*

It is a by-product obtained during destructive distillation of coal. It mainly contains H_2, CO, CO_2, CH_4 and N_2. It has a CV ranging from 21,000 to 25,000 kJ/m^3. It is also known as *town gas*.

3. *Coke oven gas*

It is obtained by the carbonization of bituminous coal. It is used for industrial heating.

4. *Blast furnace gas*

It is obtained as a by-product during blast furnace operation. Due to the high percentage of dust content, it should be purified before it is used. It has a CV ranging between 3800 and 4200 kJ/m^3. It is used as a fuel in gas engines, steam generation and in steel plants.

5. *Producer gas*

It is obtained by partial combustion of coke or coal in the presence of air and steam mixture. It has a CV ranging between 5000 and 6800 kJ/m^3. It is used for industrial gas melting and also for power generation.

6. *Water gas*

It is obtained by passing steam over candescent coke. It burns with a blue flame, and hence it is known as blue water gas. It has a CV ranging between 10,500 and 2300 kJ/m^3. It is used in welding and in furnaces.

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7. *Mond gas*

It is produced by passing air and a large amount of steam over waste coal at about 650°C. It has a CV of 5800 kJ/m^3. It is used for heating and steam generation.

Advantages and disadvantages of gaseous fuels
Advantages

(i) The rate of combustion and temperature can be easily controlled by regulating the flow rate of gas.
(ii) Mixing of fuel and air is uniform, and hence combustion is complete.
(iii) They are free from solid and liquid impurities and burn completely.
(iv) They burn without leaving traces of smoke and ash.
(v) They are eco-friendly and do not pollute atmosphere.
(vi) They can be conveyed easily through pipelines.
(vii) Combustion efficiency is more due to the necessity of minimum amount of excess air.
(viii) They can be directly used as fuels in internal combustion engines.

Disadvantages

(i) They are highly inflammable.
(ii) They require larger storage volume capacity.

2.2.4 Nuclear Fuels

Nuclear reactions, namely *fission* and *fusion*, result in the generation of enormous amount of energy. *In fission reactions*, a heavy atom is split by neutrons into two lighter fragments. *In fusion reactions*, two lighter atomic nuclei are fused together to form a single, heavy nucleus. In order to carry out nuclear reactions, fissionable materials are used as fuels.

1. *Natural fuels*

Uranium is the only fissionable nuclear fuel occurring in nature. It consists of 99.3 per cent $_{92}U^{238}$, 0.7 per cent $_{92}U^{235}$ and small traces of $_{92}U^{234}$. Out of these isotopes, only $_{92}U^{235}$ is used in fission reaction.

2. *Artificial fuels*

Plutonium ($92Pu^{239}$) and Uranium ($_{92}U^{238}$) are the fissionable materials produced artificially from $_{92}U^{238}$ and Thorium ($_{90}Th^{232}$), respectively. Uranium-238 and Thorium-232 are available in nature. They are known as fertile materials. These *fertile materials* when placed close to $_{92}U^{235}$ in a reactor absorb the emitted neutrons to produce fissionable materials as follows:

1. $_{92}U^{238} + {}_0n^1 \rightarrow {}_{92}U^{239} + \gamma$ (radiation)
   $_{92}U^{239} \rightarrow {}_{93}Np^{239}$ (Neptunium) $+ {}_{-1}e^0$ (electron)
   $_{92}Np^{239} \rightarrow {}_{94}Pu^{239} + {}_{-1}e^0$
2. $_{90}Th^{232} + {}_0n^1 \rightarrow {}_{90}Th^{233} + \gamma$
   $_{90}Th^{233} \rightarrow {}_{91}Pa^{233}$ (Protactinium) $+ {}_{-1}e^0$
   $_{91}Pa^{233} \rightarrow {}_{92}U^{233} + {}_{-1}e^0$

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Advantages and disadvantages over solid fuels
Advantages

(i) Under favourable conditions, the cost of generation of nuclear power is competitive with coal-based power generation.
(ii) Isotopes produced as by-products can be used in agriculture, medicine, scientific research and industrial processing.
(iii) They are used in transportation systems, especially in nuclear-powered submarines.
(iv) Complete fission of 1 kg of fissile uranium produces about 23×10^6 kWh of energy that is equivalent to the energy generated by burning 3300 t of bituminous coal.

Disadvantages

(i) The radiation emitted during the reaction is highly hazardous.
(ii) Nuclear waste disposal is a major problem; and if it not disposed properly, it is hazardous to all living things on the earth.

2.3 CALORIFIC VALUES OF FUELS

The calorific value (CV) of fuel is defined as the amount of heat liberated by the complete combustion of unit quantity of fuel.

The CV of a *solid* or *liquid* fuel is defined as the amount of heat liberated by the complete combustion of unit mass of fuel. In SI units, it is expressed in kJ/kg.

The CV of a gaseous fuel is defined as the amount of heat liberated by the complete combustion of unit volume of fuel. In SI units, it is expressed in kJ/m^3.

2.3.1 Higher Calorific Value and Lower Calorific Value of Fuels

1. *Higher calorific value*

As all fuels contain hydrogen, they produce water vapour during combustion. When the products of combustion containing water vapour are cooled back to initial (room) temperature, then all water vapours formed condense and evolve latent heat. This adds up to the heat liberated by burning the fuel, producing maximum amount of heat per kilogram of fuel. This heat is known as *higher calorific value of fuel*, and it is denoted by HCV.

Higher calorific value of the given fuel can be determined by using *Dulong's formula*.

Let C, H, O and S represent the percentage by weight of carbon, hydrogen, oxygen and sulphur, respectively. Then,

$$\text{HCV} = \frac{1}{100}\left[33{,}800\text{C} + 1{,}44{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 9270\text{S}\right] \text{kJ/kg}$$

2. *Lower calorific value*

In most of the combustion processes, the products of combustion (e.g. gas coming out of boiler chimney) cannot be cooled to its initial temperature. Thus, water vapours do not condense, and hence the latent heat of water vapours is lost to the atmosphere. The resultant heat liberated by

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the fuel that excludes the latent heat of evaporation of water vapours is known as *lower calorific value of fuel*, and it is denoted by LCV.

The latent heat lost to the atmosphere depends on evaporation pressure and the amount of water vapours formed. Due to the difficulty in measuring the evaporation pressure, it is assumed that evaporation takes place at a saturation temperature of 15°C. The latent heat corresponding to this saturation temperature is 2466 kJ/kg. Thus, LCV is given by the following expression:

$$\text{LCV} = \text{HCV} - m \times 2466$$

where m is the mass of water vapour formed per kilogram of fuel burnt.

As one part of hydrogen by mass produces nine parts of water vapour by mass, the mass of water vapour produced is 9H. Thus,

$$\text{LCV} = \text{HCV} - 9\text{H} \times 2466$$

where H is the percentage of hydrogen by weight.

The CVs of solid, liquid and gaseous fuels are listed below (see Tables 2.1–2.3).

Table 2.1 *Calorific Values of Solid Fuels*

S. no.	Fuel	HCV (kJ/kg)
1.	Wood (dry)	10,465
2.	Peat (dry)	14,650
3.	Lignite	21,500
4.	Bituminous coal	31,400
5.	Anthracite coal	35,600
6.	Coke	31,600

Table 2.2 *Calorific Values of Liquid Fuels*

S. no.	Fuel	Specific gravity	HCV (kJ/kg)
1.	Gasoline	0.70–0.78	47,000
2.	Kerosene	0.78–0.85	–
3.	Diesel	0.86–0.95	45,500
4.	Heavy fuel oil	0.95	44,500

Table 2.3 *Calorific Values of Gaseous Fuels*

S. no.	Fuel	HCV (kJ/m^3)	LCV (kJ/m^3)
1.	Natural gas	37,000	–
2.	Coal gas	32,000	29,000
3.	Town gas	19,500	17,500

(*Continued*)

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Table 2.3 (*Continued*)

S. no.	Fuel	HCV (kJ/m^3)	LCV (kJ/m^3)
4.	Coke oven gas	21,400	19,200
5.	Blast furnace gas	4000	400
6.	Producer gas	5000	4800

2.3.2 Experimental Procedure for Determining CV of Fuels

Different types of calorimeters are used to determine the CV of solid, liquid and gaseous fuels. *Bomb calorimeter* is used to determine the CV of *solid and liquid fuels*. *Lewis Thompson calorimeter* is used to determine the CV of *solid fuels*. *Junker's gas calorimeter* is used to determine the CV of *gaseous fuels*. The experimental procedure for determining the CV of fuels is discussed further in detail.

1. *Bomb calorimeter*

This device is used to determine the higher CV of solid fuels and liquid fuels.

Figure 2.1 shows a schematic sketch of a bomb calorimeter. It consists of a strong steel shell known as *bomb* that withstands high heat and pressure (200 atm). The bottom of the shell is screwed to a cover plate to form a gas-tight joint. Corrosion of the inner surface of the bomb due

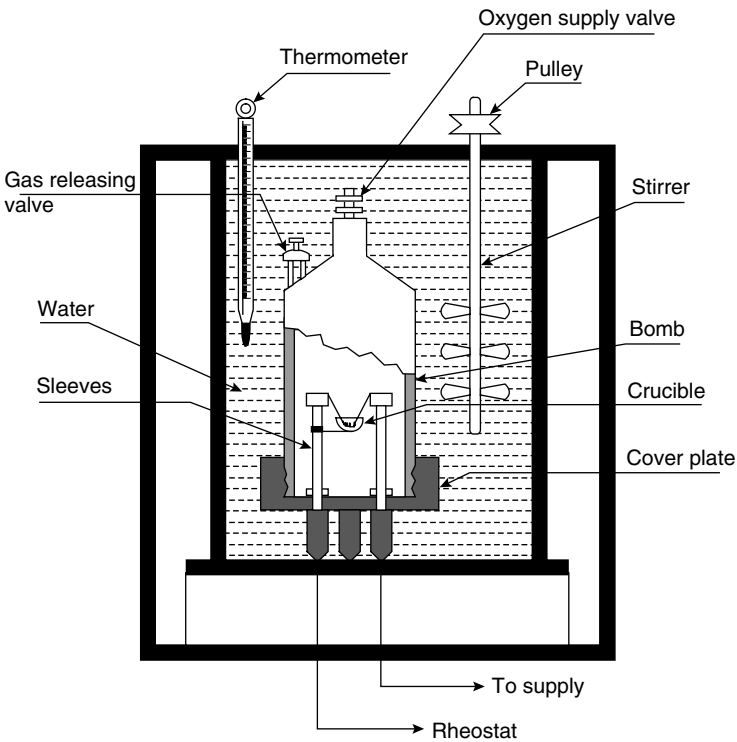


Fig. 2.1 *Bomb Calorimeter*

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to combustion of sulphur- and nitrogen-based fuels is prevented by applying an anti-corrosive enamel coating. Water is filled in the bomb to a specific level to form a water seal. The cover head of the shell carries an *oxygen supply valve* and an exhaust *gas-releasing valve*.

The bottom cover of the bomb contains two pillars; one of the pillars carries a ring for supporting a *crucible*, which is made of silica and quartz. Two insulated firing plugs connected to the bottom portion of the two pillars supply power from the mains through a *rheostat*. The bomb is placed in a copper vessel or *calorimeter* carrying water, and the calorimeter in turn is placed inside another container as shown in Figure 2.1. The air space between the two containers acts as an insulator, reducing radiation loss. A Beckmann thermometer having a range of 5°C and accuracy of 0.01°C is used to record the rise in the temperature of water. The water in the calorimeter is agitated by using a stirrer run by a motor. The capacity of the bomb is 0.65 *l*, whereas the capacity of the calorimeter is 2.5 *l*.

(i) *Procedure*

A known quantity of given fuel (solid or liquid) is taken in the crucible. If the fuel is a solid, a known quantity of fuel is powdered and a briquette is prepared from it. This briquette is placed in the crucible with a fuse wire coiled around it. If the fuel is a liquid, then the fuse wire is immersed in it. The crucible is supported over the ring. The oxygen supply valve is opened slowly and a pressure of 25–30 atm is maintained. The water in the calorimeter is stirred using a stirrer and the rise in the temperature is noted down by using a thermometer. When the temperature becomes steady, the power is switched on. The fuse wire ignites the fuel in the presence of oxygen. With the stirrer set in motion, the temperature is noted down for every 15 s till the maximum temperature is recorded. Later, the temperature is noted down for every 30 s till the drop in temperature for three successive readings are uniform. The bomb is now removed from the calorimeter, and the products of combustion are released by slowly opening the *release valve*. The products are dried and unburnt fuse wire if any is weighed, and a temperature time curve is plotted.

(ii) *Calculations*

The radiation correction is calculated using Newton's Law of cooling. According to the law, the heat loss is proportional to the temperature difference between the hot and cold bodies:

$$Q \propto (T_2 - T_1)\, t$$

where t is the time required to reach the maximum temperature.

The rate of temperature drop is determined from the temperature–time graph for the bomb calorimeter.

$$\therefore \quad \text{True temperature rise} = \frac{r}{2} \times t + \text{Recorded temperature rise}$$

$$= \frac{r}{2} \times t + (T_2 - T_1)$$

where r = Cooling rate, T_1 = Initial steady temperature before combustion, °C and T_2 = Maximum temperature after combustion, °C.

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(iii) *Water equivalent of calorimeter*

In order to determine the water equivalent of a calorimeter, a known quantity of fuel, namely benzoic acid (CV = 26,500 kJ/kg) or naphthalene (CV = 40,540 kJ/kg), is burnt.

Let
HCV = Higher calorific value of fuel, kJ/kg (unknown)
CV_w = Calorific value of fuse wire, kJ/kg (known)
m_f = Mass of fuel burnt, kg (known)
m_w = Mass of fuse wire burnt, kg (known)
M = Mass of water in calorimeter, kg (known)
w = Water equivalent of calorimeter, bomb, etc. (known)
C_{PW} = Specific heat of water,
= 4.186 kJ/kg K

Making heat balance,
Heat liberated by fuel and wire = Heat absorbed in the calorimeter

That is,
$$\text{HCV} \times m_f + m_w \times CV_w = C_{PW}\,(M+m)\left[\frac{r}{2}\times t + (T_2 - T_1)\right]$$

$\therefore$
$$\text{HCV} = \frac{C_{pw}\,(M+m)\left[\frac{r}{2}\times t + (T_2 - T_1)\right] - m_w \times CV_w}{m_f}$$

2. *Lewis thompson calorimeter*

Lewis Thompson calorimeter is used to determine the CV of a solid fuel.
Figure 2.2 shows a schematic diagram of a Lewis Thompson calorimeter. It consists of a copper *diving bell*, which is a cylindrical tube of 50 mm diameter with spring clips attached to it.

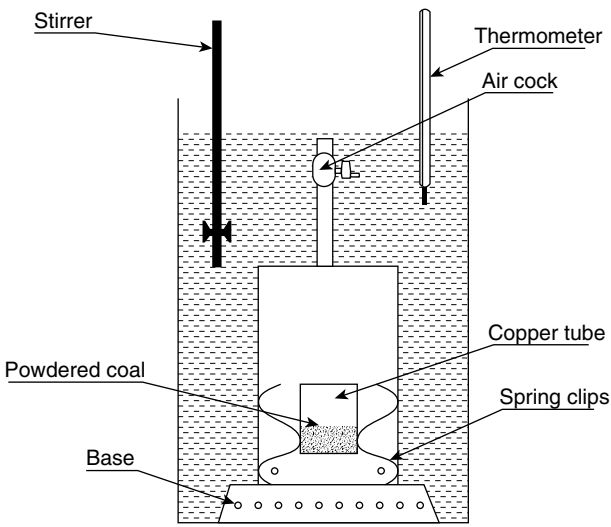


Fig. 2.2 *Lewis Thompson Calorimeter*

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A few holes are provided at the bottom of the diving bell around its circumference. A small tube of 10 mm diameter is attached to the closed end of the diving bell. An air cock is fitted to the top end of the tube. A brass cylindrical cup holds the sample fuel in its powdered form, and the cup is held in position by spring clips. The brass cup is kept on the copper base that supports the diving bell.

(i) *Procedure*

A sample of fuel (which is a mixture of powdered dry coal, potassium chlorate and potassium nitrate in the ratio 1:8.25:1.75) is taken in the cylindrical cup. The cup is held in position by the spring clips on the copper base. In the meantime, the known mass of water m_w is taken in the glass jar along with a stirrer and a thermometer. Initial temperature of water t_{W1} in the glass jar is noted. The air cock on the diving bell is closed and the diving bell is kept ready. Now, the fuel mixture containing the test sample is ignited and the diving bell is gently slipped over the test sample by releasing the spring clips. The whole unit is now immersed in the cylindrical glass jar. As combustion continues, burnt gases escape through the holes provided at the bottom of the bell and finally through the water in the glass jar. When combustion is complete, the air cock is opened so that the water rushes into the diving bell. The water in the glass jar is now stirred continuously using a stirrer and the rise in temperature of water is noted down for every 15 s till the maximum temperature is reached. The time taken for reaching this maximum temperature is noted down using a stopwatch. When the temperature starts dropping down, keep noting it down for every 30 s till the drop in temperature for successive three readings is uniform.

(ii) *Calculations*

Let
T_1 = Initial temperature of water, °C
T_2 = Final (maximum) temperature of water, °C
CV_f = Calorific value of fuel, kJ/kg
m_w = Mass of water, kg
w = Water equivalent of calorimeter, kg
m_f = Mass of fuel, kg

Assuming that 10 per cent of heat is lost due to radiation, heat balance is calculated as follows:

Heat lost by fuel = Heat gained by calorimeter and water.

$$m_f \times CV_f = (w + m_w)\, C_{PW}\, (T_2 - T_1)$$

$$\therefore \quad CV_f = (w + m_w)\, C_{PW}\, (T_2 - T_1)/m_f$$

(iii) *Note*

In this experiment, the sample of fuel is burnt under water. The oxygen that is necessary for combustion is supplied by potassium chlorate and potassium nitrate. In the presence of oxygen, heat is liberated by potassium chlorate, whereas it is absorbed by potassium nitrate. Hence, they are taken in the ratio of 8.25:1.75 in order to see that the heat liberated and the heat absorbed are balanced.

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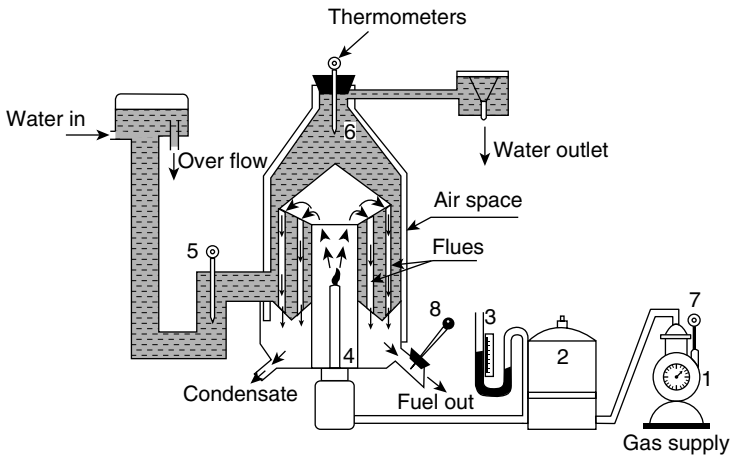


Fig. 2.3 *Junker's Gas Calorimeter*

3. *Junker's gas calorimeter*

Junker's gas calorimeter is used to determine the CV of a gaseous fuel.

Figure 2.3 shows a schematic sketch of a continuous flow gas calorimeter. It consists of a *gas meter* (1), which is used to supply sample gas. The pressure of the gas is regulated using a gas *pressure regulator* (2) and the pressure is measured using a *manometer* (3) that is attached to it. The gas passes through the pressure regulator to a *burner* (4). The inlet gas temperature is recorded by the *thermometer* (7) and the temperature of products of combustion leaving the calorimeter is recorded by the *thermometer* (8). The gaseous products of combustion from the burner flame move upwards in the combustion chamber and are deflected downwards. They enter the flues on either side of the combustion chamber before entering the outlet. A water jacket that covers the central combustion chamber absorbs the heat produced by the burning gas. An air space is maintained between the outer cover and the calorimeter, which acts as an insulation, thereby reducing radiation losses. An overhead water tank supplies water to the calorimeter continuously. Water flows through the calorimeter against the direction of flow of flue gases (products of combustion), cooling the hot gases to atmospheric temperature. The water vapour condensed due to the combustion of hydrogen is drained from the base and is collected in a measuring jar.

(i) *Procedure*

Before starting the experiment, continuous flow of water through the water jacket and flow of a measured volume of gas through the gas meter are ensured. The pressure regulator is adjusted to ensure uniform flow of water, and the manometer reading is recorded. The cooling water *inlet temperature* (5) and the inlet gas temperature are recorded. The sample gas is ignited and the time taken for collecting 2000 cc of water from the calorimeter is noted. The cooling water *outlet temperature* (6) and the temperature of products of combustion are also noted. The condensate formed due to the burning of hydrogen is collected and weighed.

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(ii) *Calculations*

Let

V_g = Volume of the gaseous fuel burnt, m^3

T_g = Temperature of the gaseous fuel leaving the gas meter, K

p_g = Absolute pressure of the gaseous fuel corresponding to the difference in water column h_w in the manometer

$$= \left(\frac{h_w}{13.6} + h_b\right) \text{cm of Hg}$$

h_b = Barometer reading in cm of Hg

m_w = Mass of cooling water circulated during the test, kg

m_c = Mass of the condensate collected during the test, kg

t_{W1} = Initial temperature of water, °C

t_{W2} = Final temperature of water, °C

V_0 = Volume of gas reduced to standard temperature and pressure (STP), m^3

p_0 = Pressure at STP (76 cm of Hg)

T_0 = Temperature of the gas at STP (288 K)

The CV of gaseous fuels is normally expressed in heat units per cubic metre of gas at STP.

Applying ideal gas equation,

$$\frac{p_0 V_0}{T_0} = \frac{p_g V_g}{T_g}$$

$\therefore$

$$V_0 = \frac{p_g V_g}{T_g} \times \frac{T_0}{p_0}$$

Making heat balance,

Heat lost by fuel due to combustion = Heat gained by water

That is,

$$V_0 \times \text{HCV} = m_w C_{PW} (t_{W2} - t_{W1})$$

HCV of the gaseous fuel is given by the following expression:

$$\text{HCV} = \frac{m_W C_{PW} (t_{w2} - t_{w1})}{V_0} \text{kJ/m}^3 \text{ at STP}$$

where C_{PW} = Specific heat of water = 4.186 kJ/kgK

The LCV of the fuel is obtained by subtracting the heat in the condensate from HCV.

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Heat lost in the condensate,

$$Q_C = \frac{m_c \times h_{fg}}{V_0} = \frac{m_c \times 2466}{V_0} \text{ kJ/m}^3$$

∴ Lower calorific value is,

$$\text{LCV} = \text{HCV} - Q_C \text{ kJ/m}^3$$

where h_{fg} = Latent heat of evaporation of water vapour

= 2466 kJ/kg

2.4 COMBUSTION OF FUELS

Due to the presence of combustible elements such as carbon (C), hydrogen (H_2) and sulphur (S), fuels burn in the presence of oxygen (O_2) and generate heat. For the complete combustion of fuels, sufficient amount of air must be supplied. If the air supplied is not sufficient, then combustion will be incomplete. This means that the carbon in the fuel burns to produce carbon monoxide (CO) instead of carbon dioxide (CO_2), thereby reducing the amount of heat generated.

The following combustion equations are used to calculate the amount of oxygen required and the amount of gases produced.

1. $C + O_2 \rightarrow CO_2$

Based on molecular weight, 12 kg of carbon is combined with 32 kg of oxygen to form 44 kg of CO_2.

i. e., 12 kg C + 32 kg O_2 → 44 kg CO_2

1 kg C + 8/3 kg O_2 → 11/3 kg CO_2 (1)

2. $2C + O_2 \rightarrow 2CO$

Based on molecular weight, 24 kg of carbon is combined with 32 kg of oxygen to form 56 kg of carbon monoxide.

i. e., 24 kg C + 32 kg O_2 → 56 kg CO

1 kg C + 4/3 kg O_2 → 7/3 kg CO (2)

3. $2CO + O_2 \rightarrow 2CO_2$

Based on molecular weight, 56 kg of carbon monoxide is combined with 32 kg of oxygen to form 88 kg of CO_2.

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i. e., 56 kg CO + 32 kg O_2 → 88 kg CO_2

1 kg CO + 4/7 kg O_2 → 11/7 kg CO_2 (3)

4. $2H_2 + O_2 \rightarrow 2H_2O$

Based on molecular weight, 4 kg of H_2 is combined with 32 kg of oxygen to form 36 kg of water.

i. e., 4 kg H_2 + 32 kg O_2 → 36 kg H_2O

1 kg H_2 + 8 kg O_2 → 9 kg H_2O (4)

5. $S + O_2 \rightarrow SO_2$

Based on molecular weight, 32 kg of sulphur is combined with 32 kg of oxygen to form 64 kg of SO_2.

i.e., 32 kg S + 32 kg O_2 → 64 kg SO_2

1 kg of S + 1 kg of O_2 → 2 kg of SO_2 (5)

2.4.1 Minimum Air Required per Kilogram of Liquid Fuel for Complete Combustion

Let C, H, O and S represent percentage by mass of C, H_2, O_2 and S, respectively.

The mass of oxygen required for *complete combustion* of fuel can be calculated using the above Equations (1), (4) and (5) as follows:

Oxygen per kilogram of fuel for complete combustion:

$$= \frac{1}{100}\left[\frac{8}{3}C + 8H - O + S\right]$$

$$= \frac{1}{100}\left[\frac{8}{3}C + 8\left(H - \frac{O}{8}\right) - O + S\right] \quad (1)$$

As air contains 23 per cent of oxygen by mass, the minimum air required for burning 1 kg of liquid fuel completely is given by the following equation:

$$= \frac{1}{100}\left[\frac{8}{3}C + 8\left(H - \frac{O}{8}\right) + S\right] \times \frac{100}{23}$$

$$= \frac{1}{23}\left[\frac{8}{3}C + 8\left(H - \frac{O}{8}\right) + S\right] \quad (2)$$

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2.4.2 Minimum Air Required per Cubic Metre of Gaseous Fuel for Complete Combustion

For determining the minimum air required for completely burning 1 m^3 of a gaseous fuel, the volumetric analysis of the gaseous fuel is done. A gaseous fuel has a mixture of combustible and incombustible gases. Volumetric analysis of the fuel gives the percentage by volume of the following gases:

1. Hydrogen (H_2)

$$2H_2 + O_2 \rightarrow 2\,H_2O$$

i.e., 2 volume H_2 + 1 volume $O_2 \rightarrow$ 2 volume H_2O

$$1H_2 + ½O_2 \rightarrow H_2O \quad (1)$$

1 volume of H_2 + ½ volume of $O_2 \rightarrow$ 1 volume H_2O

2. Carbon monoxide (CO)

$$2CO + O_2 \rightarrow 2CO_2$$

$$CO + ½O_2 \rightarrow CO_2 \quad (2)$$

1 volume CO + ½ volumeof $O_2 \rightarrow$ 1 volume CO_2

3. Methane (CH_4)

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \quad (3)$$

1 volume CH_4 + 2 volume of $O_2 \rightarrow$ 1 volume CO_2 + 2 volume H_2O

4. Ethane (C_2H_4)

$$C_2H_4 + 3O_2 \rightarrow 2CO_2 + 2H_2O \quad (4)$$

1 volume C_2H_4 + 3 volume of $O_2 \rightarrow$ 2 volume CO_2 + 2 volume H_2O

5. Carbon dioxide (CO_2)
   It is not combustible, and hence it does not require oxygen.
6. Nitrogen (N_2)
   It is not combustible, and hence it does not require oxygen.

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From the above four equations, we can determine the amount of oxygen required to burn 1 m^3 of a gaseous fuel as follows:

Oxygen required per cubic metre of fuel for complete combustion:

$$= \frac{1}{100}\left[\frac{H_2}{2} + \frac{CO}{2} + 2CH_4 + 3C_2H_4\right]$$

As atmospheric air contains 21 per cent of oxygen by volume, the minimum air required to burn 1 m^3 of a gaseous fuel is given by the following equation:

Minimum volume of air required (m^3/m^3 of fuel)

$$= \frac{1}{100}\left[\frac{H_2}{2} + \frac{CO}{2} + 2CH_4 + 3C_2H_4\right] \times \frac{100}{21}$$

$$\therefore \quad \text{Minimum air required} = \frac{1}{21}\left[\frac{H_2}{2} + \frac{CO}{2} + 2CH_4 + 3C_2H_4\right] \text{ m}^3/\text{m}^3 \text{ of fuel}$$

where H_2, CO, CH_4 and C_2H_4 are the percentages by volume of respective fuels.

2.4.3 Combustion Equation for a Hydrocarbon Fuel

Combustion equation for any hydrocarbon fuel can be generated as follows.

Consider a hydrocarbon fuel say aromatics, C_nH_{4n-2}. When this fuel is burnt, the products of combustion are carbon dioxide (CO_2) and water (H_2O). Writing the combustion equation, we have

$$xC_nH_{4n-2} + yO_2 \rightarrow aCO_2 + bH_2O \tag{1}$$

where x, y, a and b are constants to be determined.

Making mass balance on both sides of the equation for C, H and O separately.

Mass balance for C: $x \times n = a \Rightarrow a = x.n$ (2)

Mass balance for H: $x \times (4n - 2) = 2b$

$\therefore \quad b = (4n - 2) = x\,(2n - 1)$ (3)

Mass balance for O: $2 \times y = 2 \times a + b$

$\therefore \quad y = a + \frac{b}{2}$

$$= x.n + \frac{x}{2}\,(2n - 1)$$

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$$= \frac{x}{2}[n+2n-1]$$

$$y = \frac{x}{2}[n-1] \tag{4}$$

Substituting Equations (2), (3) and (4) in Equation (1), we get:

$$xC_nH_{4n-2} + \frac{x}{2}[n-1]O_2 \rightarrow x.n\ CO_2 + x\ (2n-1)\ H_2O \tag{5}$$

2.4.4 Flue Gas Analysis

The percentage by volume of carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2) and nitrogen (N_2) in a flue gas is determined by using an *Orsat* apparatus. (Fig. 2.4)

The flue gas analysis carried out by the Orsat apparatus gives fairly accurate results upto ± 0.5 per cent of CO_2 content. In its simplest form, the apparatus is arranged to determine the composition of CO_2, O_2, CO and N_2 by difference. It contains a *eudiometer* (A) graduated upto 100 cc and surrounded by water jacket. The base of the eudiometer is connected to an *aspirator bottle* (B) by means of a flexible *rubber tube*. By lowering or raising the aspirator bottle, the eudiometer can be either charged or discharged. The flasks 1, 2 and 3 with duplicate flasks behind them contain KOH solution (caustic potash with KOH:H_2O 1:2 by weight) for absorbing CO_2; alkaline solution of pyrogallic acid (5 gm of pyrogallic acid in 15 cc of water is mixed with 120 gm of KOH in 80 cc of water) for absorbing O_2 and cuprous chloride (CuO dissolved in 20 times its weight in concentrated HCl with copper wire immersed till it becomes colourless) for absorbing CO. Small glass tubes packed in the flasks 1, 2 and 3 accelerate the absorption of gases.

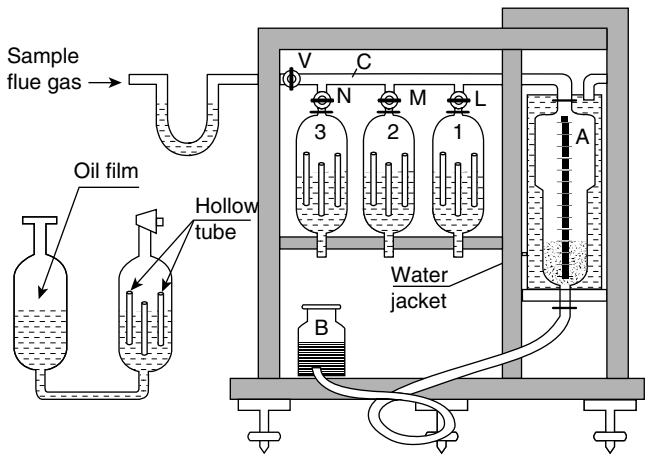


Fig. 2.4 *Orsat Apparatus*

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(i) *Procedure*

The chemical reagents in the flasks 1, 2 and 3 are freshly prepared and filled. The eudiometer jackets are filled with water. The aspirator bottle is filled with salt water so that the water does not absorb the gases. The three-way cock 'V' is opened to the atmosphere, and the aspirator bottle is raised till the salt water level in the eudiometer tube reaches 100 cc mark and then the cock is closed. Now, cock 'L' is opened and the aspirator bottle is lowered. The reagent in the flask 1 raises and touches the mark on the capillary tube. Cock 'L' is immediately closed. Due to this action, all gases including air are expelled from the flask. The same procedure is repeated with flasks 2 and 3. The aspirator bottle is now raised to expel all the gases removed from the flasks. It can be noted that the amount of gases present in the capillary tube 'C' is negligible.

The sample gas tube is now connected to the three-way cock, and the cock is opened to the flue gases. The aspirator bottle is lowered to aspire exactly 100 cc of sample gas (till the water level touches 0 cc mark). Cock 'V' is then closed. In order to ensure that there is no leakage, the aspirator bottle is raised again. If there is a leakage, then the water level in the eudiometer keeps rising. The sample of gas is aspirated through a U-tube containing wool to ensure only filtered gas is entering the setup. The water jacket of the eudiometer ensures that all the moisture condenses, and a dry sample of 100 cc gas is obtained. The aspirator bottle is now brought near to the eudiometer so that the water level in the eudiometer and that inside the bottle remains the same. This ensures that the gas sample collected is at atmospheric pressure.

The cock 'L' is opened and the aspirator bottle is raised to pass the sample of gas into flask 1 that contains KOH solution. During this period, the reagent passes to the duplicate flask to accommodate the sample gas and the wet surface of the glass tube absorbs CO_2 gas. The aspirator bottle is now lowered so that the reagent rises, thereby washing the glass tubes. The aspirator bottle is raised and lowered a few times so that all CO_2 is absorbed. The cock 'L' is then closed. The aspirator bottle is brought near to the eudiometer again such that the level of the water in the bottle and the eudiometer remains the same. The reading on the eudiometer tube is noted, and this gives the percentage by volume of CO_2.

The same procedure is repeated by opening cocks, 'M' and 'N' one by one. The corresponding readings on the eudiometer tube give the percentage by volume of oxygen (O_2) and carbon monoxide (CO), respectively.

Percentage by volume of nitrogen (N_2) is determined as follows:

$$\text{Per cent by volume of } N_2 = 100 - \text{per cent by volume of } (CO_2 + O_2 + CO).$$

2.4.5 Conversion of Gravimetric Analysis to Volumetric Analysis and Vice Versa

If the volumetric analysis of a gas is known, then it can be converted into gravimetric analysis (mass analysis), by multiplying the volume of each constituent gas by its molecular weight.

If the gravimetric analysis of the gas is known, then it can be converted into volumetric analysis using the following relation:

$$\text{Per cent by volume} = \frac{\text{Per cent by mass } (a)}{\text{Molecular weight } (b)}$$

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Table 2.4 *Volumetric Analysis of Exhaust Gases*

S. no.	Constituent gas	Percentage by volume (*a*)	Molecular weight (*b*)	Proportional volume (*c*) = (*a*)/(*b*)	Percentage by volume (*d*) = (*c*/Σ*c*) 100
1.	CO_2				
2.	CO				
3.	O_2				
4.	N_2				
				$\Sigma c =$	

If the exhaust gas contains CO, CO_2, O_2 and N_2 percentage by volume as obtained from the analysis of dry exhaust gas, then the volumetric analysis of exhaust gases can be done as given in Table 2.4.

The volumetric analysis of exhaust gas can be converted into gravimetric (mass) analysis as shown in Table 2.5.

Table 2.5 *Gravimetric (Mass) Analysis from Volumetric Analysis*

S. no.	Constituent gas	Percentage by volume (*a*)	Molecular weight (*b*)	Proportional mass (*c*) = (*a*) × (*b*)	Percentage by mass (*d*) = (*c*/Σ*c*) 100
1.	CO_2				
2.	CO				
3.	O_2				
4.	N_2				
				$\Sigma c =$	

2.4.6 Mass of Air Supplied per Kilogram of Fuel

When the volumetric analysis of dry flue gases and the percentage of carbon by mass are known, we can determine the mass of air supplied (ma) per kilogram of fuel burnt as follows:

Let C = Percentage of carbon by mass in the fuel burnt
CO_2 = Percentage of carbon dioxide by volume in the dry flue gas
CO = Percentage of carbon monoxide by volume in the dry flue gas
N_2 = Percentage of nitrogen by volume in the dry flue gas

Considering 1 m^3 of gas, the mass of gas per kilogram of dry flue gases being formed can be determined as follows:

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S. no.	Constituent gas	Percentage by volume (*a*)	Molecular weight (*b*)	Proportional mass (*c*) = (*a*) × (*c*)	Mass of gas/kg of dry flue gas formed (*d*)= (*c*/Σ*c*)
1.	CO	CO/100	28	$\frac{28CO}{100}$	$\frac{28CO}{100\Sigma c}$
2.	CO_2	CO_2/100	44	$\frac{44CO_2}{100}$	$\frac{44CO_2}{100\Sigma c}$
3.	N_2	N_2/100	28	$\frac{28N}{100}$	$\frac{28N}{100\Sigma c}$
4.	O_2	O_2/100	32	$\frac{32O}{100}$	$\frac{32O}{100\Sigma c}$
				Σc =	

where $\Sigma c = 1/100\ [28CO + 44CO_2 + 28N + 32O]$

Consider the reactions producing CO_2

$$C + O_2 \rightarrow CO_2$$

$$12 \text{ kg C} + 32 \text{ kg } O_2 \rightarrow 44 \text{ kg } CO_2$$

$$\therefore \quad \frac{12}{44} \text{ kg C} + \frac{32}{44} \text{ kg } O_2 \rightarrow 1 \text{ kg } CO_2$$

or, 1 kg CO_2 is produced by $\mathbf{\frac{12}{44}}$ kg C (1)

Consider reaction producing CO

$$2C + O_2 \rightarrow 2CO$$

$$2 \times 12 \text{ kg C} + 32 \text{ kg } O_2 \rightarrow 56 \text{ kgCO}$$

$$\therefore \quad \frac{12}{28} \text{ kg C} + \frac{32}{56} \text{ kg } O_2 \rightarrow 1 \text{ kg CO}$$

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or, 1 kg CO is produced by $\frac{12}{28}$ kg C (2)

From the above table, it is evident that 1 kg of flue gas contains

$$\frac{28CO}{100\Sigma c} \text{ kg CO} \tag{3}$$

and $$\frac{44CO_2}{100\Sigma c} \text{ kg } CO_2 \tag{4}$$

∴ The amount of carbon present in $\frac{28CO}{100\Sigma c}$ kg CO

$$= \frac{28CO}{100\Sigma c} \times \frac{12}{8} = \frac{12CO}{100\Sigma c}$$

The amount of carbon present in $\frac{44CO_2}{100\Sigma c}$ kg CO_2

$$= \frac{44CO_2}{100\Sigma c} \times \frac{12}{44} = \frac{12CO_2}{100\Sigma c}$$

∴ The total mass of C in 1 kg of flue gas,

$$= \frac{12}{100}\left[\frac{CO + CO_2}{\Sigma c}\right] \tag{5}$$

Note: The above analysis holds good with the assumption that the fuel does not contain any nitrogen, and all the carbon in CO and CO_2 come from the carbon present in the fuel.

If m_g is the mass of gases formed per kilogram of fuel burnt, then

Mass of carbon in fuel = mass of carbon in flue gases

i. e. $$= \frac{12}{100}\left[\frac{CO + CO_2}{\Sigma c}\right] \times \text{mg}$$

∴ $$m_g = \frac{C.\Sigma c}{12(CO + CO_2)} \text{ per kg of fuel} \tag{6}$$

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Nitrogen in air is carried away by the flue gases. Since air contains 77 per cent of nitrogen by mass,

$$\text{Mass of nitrogen gas per kilogram of dry flue gas formed} = \frac{28N_2}{100\Sigma c}$$

$$\therefore \quad \text{Air supplied per kilogram of flue gas formed} = \frac{28N_2}{100\Sigma c} \times \frac{100}{77} \tag{7}$$

Mass of air supplied per kilogram of fuel = mass of air/kg of flue gas + mass of flue gases per kilogram of fuel

Substituting into Equations (6) and (7), we get:

$$m_a = \frac{28N_2}{100\Sigma c} \times \frac{100}{77} \times \frac{C.\Sigma c}{12(CO + CO_2)}$$

$$\therefore \quad m_a = \frac{28N_2}{77} \times \frac{C}{12(CO + CO_2)}$$

or

$$m_a = \frac{1}{33}\left[\frac{CN_2}{CO + CO_2}\right] \text{ per kg of fuel} \tag{8}$$

Note: If combustion is complete, the CO = 0 in the above equation.

2.4.7 Carbon Burnt to CO_2 and CO

If the volumetric composition of dry flue gases is known, we can determine the carbon burnt to CO_2 and CO as follows:

$$\text{Carbon burnt to } CO_2 = \frac{\text{Mass of carbon in } CO_2 \text{/kg flue gas}}{\text{Mass of carbon in 1 kg of flue gas}}$$

$$= \frac{44CO_2}{100\Sigma c}$$

$$= \frac{44CO_2}{100\Sigma c} \times \frac{12}{44} + \frac{28CO}{100\Sigma c} \times \frac{12}{28}$$

$$= \frac{CO_2}{CO + CO_2} \tag{1}$$

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Carbon burnt to CO $$= \frac{CO \times C}{CO + CO_2} \quad (2)$$

where C = actual proportion of carbon burnt per kilogram of fuel

2.4.8 Excess Air Supplied

Theoretically, we calculated the minimum air required for complete combustion of a fuel. However in actual practice, air may not come in contact with the fuel completely resulting in incomplete combustion in order to overcome this difficulty, preheated air is supplied in excess ranging from 25 to 50 per cent to ensure complete combustion of fuel.

Let V = volume of flue gases in m^3/m^3 of fuel with minimum air supplied for complete combustion

V_e = volume of excess air supplied in m^3/m^3 of fuel.

then, volume of excess air in flue gases in m^3/m^3 of flue gas

$$= \frac{V_e}{V + V_e} \ m^3 \quad (1)$$

If air contains O_2 m^3 of oxygen per cubic metre of flue gas, then air in the flue gas per cubic metre of flue gas:

$$= \frac{O_2}{21} \quad (2)$$

Equating (1) and (2) $$\frac{V_e}{V + V_e} = \frac{O_2}{21}$$

Excess air, $$V_e = \frac{O_2 . V}{21 - O_2} \ m^3/m^3 \text{ of fuel} \quad (3)$$

Example 2.1

The volumetric composition of a gaseous fuel is given by H_2 = 27%, CO = 7%, CH_4 = 48%, C_2H_4 = 13%, CO_2 = 3% and N_2 = 2%. Determine the minimum quantity of air required for burning one m^3 of fuel.

Solution: The minimum volume of air required:

$$= \frac{1}{21}\left[\frac{H_2}{2} + \frac{CO}{2} + 2CH_4 + 3C_2H_4\right]$$

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$$= \frac{1}{21}\left[\frac{27}{2}+\frac{7}{2}+2\times 48+3\times 13\right]$$

$$= 7.24\ \text{m}^3/\text{m}^3 \text{ of fuel.}$$

Example 2.2

Analysis of a sample of boiler coal has the following compositions by weight. Find the HCV and LCV of the fuel: C = 81%, H_2 = 5%, O_2 = 5%, S = 2%, ash = 7%.

Solution: By using Dulong's formula,

$$\text{HCV} = \frac{1}{100}\left[33{,}800\text{C}+1{,}44{,}000\left(\text{H}-\frac{\text{O}}{8}\right)+9270\text{S}\right]$$

$$= \frac{1}{100}\left[33{,}800\times 81+1{,}44{,}000\left(5-\frac{5}{8}\right)+9270\times 2\right]$$

$$\text{HCV} = 33863.4 \text{ kJ/kg}$$

The LCV is given by

$$\text{LCV} = \text{HCV}-\frac{9\text{H}}{100}\times 2466$$

$$= 33863.4-\frac{9\times 5}{100}\times 2466$$

$$\text{LCV} = 32753.7 \text{ kJ/kg}$$

Example 2.3

Find the HCV of the fuel whose composition by mass is as follows: C = 91%, H = 3%, S = 0.8% and the remainder being ash.

Solution: By using Dulong's formula, we get

$$\text{HCV} = \frac{1}{100}\left[33{,}800\text{C}+1{,}44{,}000\left(\text{H}-\frac{\text{O}}{8}\right)+9270\text{S}\right]$$

$$= 33{,}800\times 0.91 + 1{,}44{,}000\ (0.03-0) + 9270\times 0.008$$

$$\text{HCV} = 35152.16 \text{ kJ/kg}$$

Example 2.4

Find the LCV of a fuel whose composition by mass is as follows: C = 75%, H = 5%, O = 8%, N = 4%, S = 2% and ash = 6%.

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Solution: Nitrogen and ash are incombustibles.

By using Dulong's formula, we get

$$\text{HCV} = \frac{1}{100}\left[33{,}800\text{C} + 1{,}44{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 9270\text{S}\right]$$

$$= \frac{1}{100}\left[33{,}800 \times 75 + 1{,}44{,}000\left(5 - \frac{8}{8}\right) + 9270 \times 2\right]$$

$$\text{HCV} = 31{,}295.4 \text{ kJ/kg}$$

The LCV is given by,

$$\text{LCV} = \text{HCV} - 9\text{H}_2 \times 2466$$

$$= 31{,}295.4 - 9 \times 0.05 \times 2466$$

$$\text{LCV} = 30{,}185.7 \text{ kJ/kg}$$

Example 2.5

Carbon monoxide is burnt in the presence of oxygen to form carbon dioxide. The reaction takes place according to the following equation:

$$2\text{CO} + \text{O}_2 \rightarrow 2\text{CO}_2$$

Calculate from first principles the theoretical air required for complete combustion and the mass of carbon dioxide formed per kilogram of carbon monoxide.

Solution: From the given relation

$$2\text{CO} + \text{O}_2 \rightarrow 2\text{CO}_2$$

On mass basis,

$$2 \times 28 \text{ kg CO} + 32 \text{ kg O}_2 \rightarrow 2 \times 44 \text{ kg CO}_2$$

or

$$1 \text{ kg CO} + \frac{4}{7} \text{ kg O}_2 \rightarrow \frac{11}{7} \text{ kg CO}_2$$

From the above relation, 1 kg of CO is burnt using kg of O_2.

Theoretical air required for complete combustion

$$= \frac{100}{23} \times \frac{4}{7} = 2.484 \text{ kg}$$

Mass of carbon dioxide formed per kilogram of carbon monoxide:

$$= \frac{11}{7} = 1.571 \text{ kg}$$

Example 2.6

The percentage composition of a fuel on mass basis is as follows: C = 90%; H_2 = 3.5%, O_2 = 1%; S = 0.5% and ash = 5%.

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Calculate (i) the minimum air required for complete combustion of 1 kg of fuel and (ii) the composition of dry flue gases on mass basis if 50 per cent excess air is supplied.

Solution:

(i) Minimum air required for complete combustion of 1 kg of fuel.

$$\text{Minimum air required} = \frac{100}{23}\left[\frac{8}{3}\text{C}+8\text{H}_2+\text{S}-\text{O}_2\right]$$

$$= \frac{100}{23}\left[\frac{8}{3}\times 0.9+8\times 0.035+0.005-0.01\right]$$

$$= 11.63 \text{ kg/kg of fuel.}$$

(ii) Composition of dry flue gases.

$$\text{Actual air supplied} = 11.63 \times 1.5$$

$$= 17.445 \text{ kg/kg of fuel}$$

$$\therefore \text{ Excess air supplied} = 17.445 - 11.63$$

$$= 5.815 \text{ kg/kg of fuel}$$

Writing the combustion equation,

(i) $\text{C} + \text{O}_2 \rightarrow \text{CO}_2$

$12 + 32 \rightarrow 44$

or $1\text{C}+\frac{8}{3}\text{O}_2 \rightarrow \frac{11}{3}\text{CO}_2$

$\therefore$ $\text{CO}_2 = \frac{11}{3}\times 0.90 = 3.3 \text{ kg}$

(ii) $\text{S} + \text{O}_2 \rightarrow \text{SO}_2$

$32 + 32 \rightarrow 64$

or $1\text{S} + 1\text{O}_2 \rightarrow 2\text{SO}_2$

$\therefore$ $\text{SO}_2 = 2 \times 0.005 = 0.01 \text{ kg}$

(iii) $\text{Excess O}_2 = \frac{23}{100}\times 5.815 = 1.337 \text{ kg}$

(iv) $\text{N}_2 = \frac{77}{100} \times \text{Actual air supplied}$

$$= \frac{77}{100}\times 17.445$$

$\text{N}_2 = 13.433 \text{ kg}$

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Note: The dry product analysis does not take into account H_2O formed, and hence it should be omitted.

On percentage basis

Total mass of dry flue gas $= CO_2 + SO_2 + O_2 + N_2$

$= 3.3 + 0.01 + 1.337 + 13.433$

$= 18.08$ kg/kg of flue gas

$$CO_2 = \frac{3.3}{18.08} \times 100 = 18.252\%$$

$$SO_2 = \frac{0.01}{18.08} \times 100 = 0.055\%$$

$$O_2 = \frac{1.337}{18.08} \times 100 = 7.395\%$$

$$N_2 = \frac{13.433}{18.08} \times 100 = 74.298\%$$

Example 2.7

The following is the percentage of a coal sample on mass basis: C = 82%; H_2 = 6%; O_2 = 9 and ash = 3%.

Determine (i) The minimum air required for complete combustion of coal and (ii) the volumetric analysis of the products supplied. Assume that air contains 23 per cent oxygen on mass basis.

Solution:

(i) Minimum air supplied

$$= \frac{100}{23}\left[\frac{8}{3}C + 8H_2 + S - O_2\right]$$

$$= \frac{100}{23}\left[\frac{8}{3} \times 0.82 + 8 \times 0.06 + 0 - 0.09\right]$$

$= 11.203$ kg/kg of fuel

(ii) Volumetric analysis

Actual air supplied $= 11.203 \times 1.1 = 12.323$ kg/kg of fuel

$\therefore$ Excess air supplied $= 12.323 - 11.203$

$= 1.12$ kg/kg of fuel

Products of combustion on mass basis are

$$CO_2 = \frac{11}{3}C$$

$$= \frac{11}{3} \times 0.82 = 3.01 \text{ kg/kg of fuel}$$

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$$\text{Excess } O_2 = \frac{23}{100} \times \text{Excess air}$$

$$= \frac{23}{100} \times 1.12 = 0.2576 \text{ kg/kg of fuel}$$

$$H_2O = 9H_2$$

$$= 9 \times 0.06$$

$$= 0.54 \text{ kg/kg of fuel}$$

$$N_2 = \frac{77}{100} \times \text{Actual air}$$

$$= \frac{77}{100} \times 12.323$$

$$= 9.4887 \text{ kg/kg or fuel}$$

Conversion of gravimetric analysis into volumetric analysis is essential to determine the volume analysis of flue gas.

S. no.	Constituent gas	Percentage Mass (a)	Molecular weight (b)	Proportional volume (c) = (a/b)	Percentage by volume (d) = (c/Σc) 100
1.	CO_2	3.01	44	0.0684	15.37
2.	H_2O	0.54	18	0.0300	6.74
3.	O_2	0.2576	32	0.0081	1.82
4.	N_2	9.4887	28	0.3389	76.07
				$\Sigma c = 0.4454$	100

Thus, volumetric analysis of products of combustion is as follows:

$$CO_2 = 15.37\%, \quad H_2O = 6.74\%, \quad O_2 = 1.82\%, N_2 = 76.07\%$$

Example 2.8

The fuel used in petrol engine contains 87 per cent carbon and 13 per cent H_2. The air supply is 75 per cent of that theoretically required for complete combustion. Assuming that all H_2 is burned and there is no free carbon left,

(i) Find the volumetric analysis of dry exhaust gases.

(ii) Also find the percentage of the gross CV of the fuel lost due to incomplete combustion. Take,
CV of C $\rightarrow$ CO_2 = 34,000 kJ/kg

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CV of C → CO = 10,000 kJ/kg
CV of H_2 → H_2O = 1,28,000 kJ/kg

Solution: Minimum mass of air required for complete combustion is given by

$$m_a = \frac{100}{23}\left[\frac{8}{3}\text{C} + 8\text{H}_2 + \text{S} - \text{O}_2\right] = 14.6 \text{ kg}$$

Actual air supplied, $m_a = 14.6 \times 0.75 = 10.95$ kg
Assuming mass of carbon burned to $CO_2 = x$
Mass of carbon burned to CO = y

$$x + y = 0.87 \tag{1}$$

Oxygen present in m_a kg of air = $\frac{23}{100} m_a$

Mass of O_2 available for combustion of carbon = $\frac{23}{100} m_a - 8\text{H}$

O_2 required to burn x kg of C to CO_2 = $\frac{8}{3}x$

O_2 required to burn y kg of C to CO = $\frac{4}{3}y$

$$\frac{8}{3}x + \frac{4}{3}y = \frac{23}{100} m_a - 8\text{H}$$

Substituting the values of H and m_a,

$$\frac{8}{3}x + \frac{4}{3}y = \frac{23}{100} \times 10.96 - 8 \times 0.13$$

$$2x + y = 1.11 \tag{2}$$

From (1) and (2)

$$x = 0.24 \text{ kg}$$
$$y = 0.63 \text{ kg}$$

Mass of dry flue gases formed

$$\text{CO}_2 = \frac{11}{3}x = \frac{11}{3} \times 0.24 = 0.88 \text{ kg}$$

$$\text{CO} = \frac{7}{3}y = \frac{7}{3} \times 0.63 = 1.47 \text{ kg}$$

$$\text{N}_2 = 10.96 \times \frac{77}{100} = 8.44 \text{ kg.}$$

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Volumetric analysis

S. no.	Constituent gas	Mass of gas formed per kilogram of fuel (a)	Molecular weight of gas (b)	Proportional volume $c = \frac{a}{b}$	Percentage by volume $\frac{c}{\Sigma c} \times 100$
1.	N_2	8.44	28	0.3014	80.61
2.	CO_2	0.88	44	0.02	5.35
3.	CO	1.47	28	0.0525	14.04
				$\Sigma c = 0.3739$	

Maximum possible CV of fuel when correct amount of air is supplied required for complete combustion is,

$$= 0.87 \times 34{,}000 + 0.13 \times 1{,}28{,}000 = 46{,}220 \text{ kJ/kg of air}$$

Percentage loss due to incomplete combustion

$$= \frac{0.63(34{,}000 - 10{,}000)}{46{,}220} \times 100$$

$$= 32.71\%$$

Example 2.9

In an engine test, the dry volumetric analysis of the products was $CO_2 = 0.0527$, $O_2 = 0.1338$ and $N_2 = 0.8135$. Assuming that the fuel is a pure hydrocarbon and that it is completely burnt, estimate the ratio of carbon to hydrogen in the fuel by mass and the air fuel ratio by mass.

Solution:

$$C_xH_y + aO_2 \rightarrow 5.27\ CO_2 + H_2O + 13.38 O_2$$

Carbon balance: $x = 5.27$

Oxygen balance: $2a = 5.27 \times 2 + b + 13.38 \times 2$

Hydrogen balance: $y = 2b$

$$\frac{21}{79} = \frac{O_2}{N_2} = a = 21.6$$

$\therefore$ $b = 5.9$

$y = 11.8$

$$C_{5.27}\ H_{11.8} + 21.6\ O_2 \rightarrow 5.27\ CO_2 + 5.9\ H_2O + 13.38\ O_2$$

$$\text{Percentage of carbon} = \frac{(CO)(C)}{CO + CO_2} = \frac{5.27 \times 12}{5.27 \times 12 + 11.8 \times 1} = 84.3\%$$

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Percentage of hydrogen = 15.7%

Complete combustion

$$C_{5.27}H_{11.8} + 8.22\ O_2 \rightarrow 5.27\ CO_2 + 5.9\ H_2O$$

$$O_2 \text{ required} = \frac{8.22 \times 32}{5.27 \times 12 + 11.8 \times 1} = 3.505 \text{ kg/fuel}$$

$$\text{Air supplied} = 3.505 \times \frac{100}{23} = 15.24 \text{ kg/kg of fuel}$$

Example 2.10

The fuel sample has the percentage analysis by mass as follows: C → 81%, O_2 → 5%, H_2 → 5%, moisture → 2% and ash → 7%.

Calculate theoretical minimum air required for complete combustion of 1 kg of the fuel. Also calculate the volumetric analysis of dry flue gases, if actual air supplied is 16 kg/kg of fuel. Assume that 80 per cent of carbon is burnt to CO_2 and remaining to CO. Hydrogen from the fuel burned completely.

Solution: Minimum amount of air for complete combustion is given by

$$m_a = \frac{1}{23}\left[\frac{8}{3}C + 8H_2 + S - O_2\right]$$

$$= \frac{1}{23}\left[\frac{8}{3} \times 81 + 8 \times 5 + 0 - 5\right]$$

$$= 10.91 \text{ kg}$$

Since the amount of air supplied (16 kg/kg) is more than minimum, excess air is supplied.

$$\text{Amount of } H_2 = 0.05 \text{ kg}$$

$$\text{Amount of } O_2 = 0.05 \text{ kg}$$

Amount of H_2 which is already mixed with O_2.

$$= \frac{0.05}{8}$$

$$= 0.00625 \text{ kg}$$

Amount of H_2 available for burning

$$= 0.05 - \frac{0.05}{8}$$

$$= 0.04375 \text{ kg}$$

16 kg of air contains $= 16 \times \frac{23}{100}$

$$= 3.68 \text{ kg } O_2 \qquad (1)$$

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O_2 required for burning 0.04375 kg $H_2 = 0.04375 \times 8 = 0.35$ kg
O_2 required for burning 0.81×0.8 kg C to CO_2

$$= 0.81 \times 0.8 \times \frac{8}{3}$$

$$= 1.728 \text{ kg}$$

O_2 required for 0.81×0.2 kg C to CO

$$= 0.81 \times 0.2 \times \frac{4}{3}$$

$$= 0.216 \text{ kg}$$

$\therefore$ Total O_2 consumed $= 0.35 + 1.728 + 0.216$

$$= 2.294 \text{ kg} \qquad (2)$$

Free O_2 available in exhaust gases from Equations (1) and (2),

$$= 3.680 - 2.294$$

$$= 1.386 \text{ kg}$$

Amount of N_2 in the exhaust gases:

$$= 16 \times \frac{77}{100}$$

$$= 12.32 \text{ kg}$$

Amount of CO_2 formed $= 0.81 \times 0.8 \times \frac{11}{3} = 2.376$ kg

Amount of CO formed $= 0.81 \times 0.2 \times \frac{7}{3}$

$$= 0.378 \text{ kg}$$

Volumetric analysis

S. no.	Constituent gases	Mass of gas formed per kilogram of fuel (a)	Molecular weight of gas (b)	Proportional volume $c = \frac{(a)}{(b)}$	Percentage by volume $= \left(\frac{c}{\Sigma c}\right)100$
1.	N_2	12.32	28	0.4400	79.88
2.	O_2	1.386	32	0.0433	7.86
3.	CO_2	2.376	44	0.0540	9.81
4.	CO	0.378	28	0.0135	2.45
				$\Sigma c = 0.5508$	

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Example 2.11

The volumetric analysis of a sample of producer gas supplied to an engine is

$H_2 \rightarrow 20\%$, $CH_4 \rightarrow 3\%$, $CO \rightarrow 22\%$, $CO_2 \rightarrow 8\%$ and $N_2 \rightarrow 47\%$

(i) Find the volume of air required for complete combustion of 1 m^3 of the gas

(ii) If 50 per cent excess air is supplied, find the percentage contraction in volume after the products of combustion have been cooled.

Solution: Minimum volume of air required for complete combustion of 1 m^3 of fuel

$$V_a = \frac{1}{21}\left[\frac{H_2}{2} + \frac{CO}{2} + 2CH_4 + 3C_2H_4\right]$$

$$= \frac{1}{21}\left[\frac{20}{2} + \frac{22}{2} + 2\times 3 + 3\times 0\right]$$

$$= 1.29 \text{ m}^3/\text{m}^3 \text{ of gas}$$

When combustion is completed with 50 per cent excess air, the dry constituent formed will be CO_2, N_2 and O_2.

$\therefore$ Volume of $CO_2 = 0.08 + CO_2$ formed from $CO + CO_2$ formed from CH_4

$= 0.08 + 0.22 \times 1 + 0.03 \times 1$

$= 0.33 \text{ m}^3$

Volume of $N_2 = 0.47 +$ nitrogen from the complete air supplied

$= 0.47 + 1.29 \times 1.5 \times 0.79$

$= 2 \text{ m}^3$

Volume of $O_2 =$ oxygen from the excess air only

$= 1.29 \times 0.5 \times 0.21 = 0.135 \text{ m}^3$

Total dry flue gases formed

$= CO_2 + N_2 + O_2$

$= 0.33 + 2 + 0.135 = 2.465 \text{ m}^3$

Total volume of gas before combustion = volume of gas + volume of air supplied

$= 1 + 1.29 \times 1.5$

$= 2.935 \text{ m}^3$

Percentage contraction in the volume after combustion and cooling

$$= \frac{2.935 - 2.465}{2.935} \times 100$$

$$= 16\%$$

Example 2.12

The percentage volumetric analysis of a sample of coal is $CO_2 = 10.0$, $CO = 0.3$, $O_2 = 8.0$ and $N_2 = 81.7$. Gravimetric percentage analysis of coal was C = 79, $H_2 = 5$, $O_2 = 3$ and incombustible = 13. Estimate (i) mass of dry flue gases per kilogram of fuel and (ii) mass of excess air per kilogram of fuel.

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Solution:

(i) Mass of dry flue gases per kilogram of fuel

Element	Mass (kg)	O_2 needed (kg)	Dry products (kg)
C	0.79	$0.79 \times \frac{8}{3} = 2.10$	$0.79 \times \frac{11}{3} = 2.89\ CO_2$
H_2	0.05	$0.05 \times 8 = 0.4$	
O_2	0.03		
		Total $O_2 = 2.5$	

Minimum mass of air needed for combustion = (2.50 – 0.03) = 2.47 kg

Convert volumetric analysis to gravimetric analysis,

S. no.	Constituent gas	Volume per m^3 of flue gas (*a*)	Molecular weight (*b*)	Relative mass $c = a \times b$	Mass per kilogram of flue gas $= \frac{c}{\Sigma c}$
1.	CO_2	0.10	44	4.4	0.147
2.	CO	0.003	28	0.084	0.0028
3.	N_2	0.817	28	22.876	0.7645
4.	O_2	0.08	32	2.56	0.08556
				$\Sigma c = 29.92$	

Amount of carbon present per kilogram of gases

= Amount of carbon in 0.147 kg of CO_2 + amount of carbon present in 0.0028 kilogram of CO.

$$= \frac{3}{11} \times 0.147 + \frac{3}{7} \times 0.0028 = 0.0413 \text{ kg}$$

But, carbon present in the fuel is 0.79 kg

$$\therefore \text{ Mass of dry flue gas per kilogram of fuel} = \frac{\text{Mass of carbon in 1 kg of fuel}}{\text{Mass of carbon in 1 kg of flue gas}}$$

$$= \frac{0.79}{0.0413} = 19.13 \text{ kg}$$

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(ii) Mass of excess air/kg of fuel

Mass of excess air per of fuel

Mass of excess oxygen per kilogram of flue gas $= 0.08556 - \left(0.08556 - \frac{4}{7} \times 0.0028\right)$

$= 0.084$ kg

Mass of excess O_2 per kilogram of fuel $= 19.13 \times 0.084 = 1.607$ kg

Mass of excess air per kilogram of fuel $= \frac{1.607 \times 100}{23}$

$= 6.987$ kg

Example 2.13

A fuel $C_{10}H_{22}$ is burnt using an air fuel ratio of 12:1 by mass. Air contains 77 per cent of nitrogen and 23 per cent of oxygen by mass. Assuming that the whole amount of hydrogen burns to form water vapour and there is neither any free oxygen nor any free carbon, determine the complete volumetric analysis of the products of combustion.

Solution: Writing the combustion equation,

$$2C_{10}H_{22} + 31O_2 = 20CO_2 + 22H_2O$$

$$2 \times 142 + 31 \times 32 = 20 \times 44 + 22 \times 18$$

Air required for complete combustion

$$= \frac{992}{284} \times \frac{100}{23}$$

$$= 15.187 \text{ kg/kg of fuel}$$

Actual air supplied $= 12$ kg/kg of fuel

Deficiency of air $= 15.187 - 12 = 3.187$ kg/kg of fuel

Due to incomplete combustion, carbon burns to CO and CO_2.

Now, 1 kg of C requires $\frac{4}{3} \times \frac{100}{23} = 5.8$ kg of less air to burn to CO instead of CO_2

i.e., $\frac{3.187}{5.8} = 0.549$ kg C is burnt to CO

and $\frac{12}{142} \times 10 - 0.549 = 0.296$ kg of C is burnt to CO_2

Mass of CO_2 formed $= 0.296 \times \frac{11}{3} = 1.085$ kg

Example 2.14

A petrol sample was formed to have 20 per cent hydrogen and 80 per cent carbon by weight. When used in an engine, the air supply is 92 per cent of that theoretically required for complete combustion. Calculate the percentage analysis of dry exhaust gases by volume. Assuming that all the hydrogen is burnt and that the carbon burns to carbon monoxide and carbon dioxide so that there is no free carbon.

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Solution:

Element	Mass (kg)	O_2 required (kg)
H_2	0.20	$0.20 \times 8 = 1.6$
C	0.80	$0.80 \times = 2.13$
		Total O_2 required = 3.73

Minimum amount of air required for complete combustion

$$3.73 \times \frac{100}{23} = 16.217 \text{ kg/kg of fuel}$$

Less air required = 1.29 kg/kg of fuel

Quantity of air saved by burning carbon to carbon monoxide instead of carbon dioxide

$$\frac{4}{3} \times \frac{100}{23} = 5.8 \text{ kg/kg of carbon}$$

Due to incomplete combustion,

Mass of carbon burning to carbon monoxide $= \frac{1.29}{0.58} = 0.223$ kg

Mass of carbon burning to $CO_2 = 0.80 - 0.223 = 0.577$ kg

$$CO_2 \text{ formed} = 0.577 \times \frac{11}{3} = 2.115 \text{ kg}$$

$$CO \text{ formed} = 0.223 \times \frac{7}{3} = 0.520 \text{ kg}$$

Nitrogen in the air supplied $= 16.217 \times 0.92$

$= 11.48$ kg.

Volumetric analysis of dry exhaust gases is done as follows:

S. no.	Product	Mass kg (a)	Molecular weight (b)	Proportional volume $(c) = \frac{(a)}{(b)}$	Percentage volume $= \left(\frac{c}{\Sigma c}\right)100$
1.	N_2	11.48	28	0.41	86.03
2.	CO	0.520	28	0.01857	3.9
3.	CO_2	2.115	44	0.048	10.07
				$\Sigma c = 0.4766$	

Example 2.15

The following data refer to an experimental determination of the CV of a sample of coal containing 87 per cent C and 3.0 per cent H_2. Mass of coal = 0.85 gm, mass of wire 0.03 gm of CV 6700 J/gm and mass of water in the calorimeter = 1950 gm. Water equivalent of calorimeter = 380 gm observed temperature rise = 3.0°C. Cooling correction = +0.017°C. Find HCV and LCV of the coal.

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Solution: Corrected temperature rise = $(t_2 - t_1) + t_C$

t_C = cooling correction = +0.017°C

$t_2 - t_1$ = observed temperature rise = 3 + 0.017

= 3.017°C

Heat received by water = $(m + m_c)\ 4.18 \times [(t_2 - t_1) + t_c]$

= (1950 + 380) 4.18 × 3.017

= 29,383.77 J

Heat given out by fuse wire = mass of fuse wire × CV

= 0.03 × 6700

= 201 J

Heat produced due to combustion of fuel

= 29,383.77 − 201

= 29,182.77 J

HCV of fuel,

$$\text{HCV} = \frac{29{,}182.77}{0.85}$$

= 34,332.67 J/gm

= 34,332.67 kJ/kg

Steam produced per kilogram of coal = 9 × 0.03

= 0.27 kg

Lower calorific value of coal,

LCV = HCV − 2465 × 0.27

= 34,332.67 − 665.55

= 33,667.12 kJ/kg.

Example 2.16

The following results were obtained when a sample of gas was tested:

Gas burnt in the calorimeter = 0.07 m^3
Pressure of gas supply = 5 cm of water
Temperature of gas = 12°C
Temperature of water at inlet = 10°C
Temperature of water at outlet = 24°C
Steam condensed = 0.09 kg
Weight of water heated by gas = 27 kg
Barometer pressure = 75 cm of Hg
Determine the higher and LCV of fuel per cubic metre of gas at a temperature of 14°C and barometric pressure of 76 cm of Hg.

Solution: Using ideal gas relation,

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

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$$V_2 = \frac{p_1 V_1}{T_1} \times \frac{T_2}{p_2} \qquad \left(p_1 = 75 + \frac{5}{13.6} = 75.37 \text{ cm of Hg}\right)$$

$$= \frac{75.37 \times 0.07}{2.85} \times \frac{287}{76} = 6.99 \text{m}^3$$

Heat received by water

$$m \times c_p \times \Delta T = 27 \times 4.186 \times (24 - 10) = 1582.30 \text{ kg}$$

Higher calorific value of fuel $= \frac{1582.30}{0.07} = 22604.286 \text{ kg/m}^3$

Amount of steam condensed per cubic metre of gas burnt $= \frac{0.07}{0.07} = 1$ kg

Lower calorific value,

$$\text{LCV} = \text{HCV} - 2465 \times 1$$
$$= 22604.286 - 2465$$
$$= 20139.286 \text{ kg/kg}$$

Example 2.17

A sample of fuel has the following composition:

H = 9%, S = 2%, C = 83%, O = 4% and ash = 2%.

For an air–fuel ratio of 12:1, calculate the following:

(i) Mixture strength as a percentage rich or weak
(ii) Volumetric analysis of the dry products of combustion

Solution:

Element	Mass (kg)	O_2 needed (kg)
H_2	0.09	$0.09 \times 8 = 0.72$
S	0.02	$0.02 \times \frac{1}{1} = 0.02$
C	0.83	$0.83 \times \frac{8}{3} = 2.21$
O_2	0.04	
		Total $O_2 = 2.95$

(i) Mixture strength

Mass of oxygen to be supplied per kilogram of fuel = 2.95 − 0.04

= 2.91 kg

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Mass of minimum air required for complete combustion

$$= \frac{2.91 \times 100}{23} = 12.65 \text{ kg}$$

Correct air fuel ratio = 12.65:1

Actual air fuel ratio = 12:1

$$= \frac{12.65}{12} \times 100 = 105.42\%.$$

It shows that mixture strength is 5.42% rich.

(ii) The percentage composition of day flue gases

Deficient amount of air = 12.65 − 12 = 0.65 kg

Amount of air saved by burning 1 kg of C to CO instead of CO_2:

$$= \left[\frac{8}{3}CO_2 - \frac{4}{3}CO\right] \times \frac{100}{23} = 5.8 \text{ kg}$$

Mass of carbon burning to CO

$$= \frac{0.65}{5.8} = 0.112 \text{ kg}$$

Mass of carbon burning to CO_2 = 0.83 − 0.112 = 0.718 kg

$$CO_2 \text{ formed} = 0.718 \times \frac{11}{3} = 2.63 \text{ kg}$$

$$CO \text{ formed} = 0.112 \times \frac{7}{3} = 0.261 \text{ kg}$$

SO_2 formed = 0.02 × 2 = 0.04 kg

N_2 supplied = 12 × 0.77 = 9.24 kg

S. no.	Dry products	Weight kg (a)	Molecular weight (b)	Proportional volume $c = (a)/(b)$	Percentage volume $= \frac{c}{\Sigma c} \times 100$
1.	N_2	9.24	28	0.33	82.52
2.	SO_2	0.04	64	0.000625	0.1563
3.	CO_2	2.63	44	0.0598	14.95
4.	CO	0.261	28	0.00932	2.33
				$\Sigma c = 0.3999$	

2.5 PROPERTIES OF COAL, INDIAN COALS

Coal is classified into three major types, namely anthracite, bituminous and lignite. However, there is no clear demarcation between them, and coal is also further classified as semi-anthracite, semi-bituminous and sub-bituminous.

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The chemical composition of coal has a strong influence on its combustibility. The properties of coal are broadly classified as follows: physical properties and chemical properties.

1. *Physical properties*

The following physical properties of coal are evaluated by proximate analysis, which is discussed in the next sub-section.

Heating value

The heating value of coal varies from coal field to coal field. The typical HCVs for various coals are given in the Table 2.6.

Table 2.6 *HCV(GCV) for Various Coals*

Parameter	Lignite (Dry basis)	Indian coal	Indonesian coal	South African coal
Coal HCV (kcal/kg)	4500	4000	5500	6000

2.5.1 Analysis of Coal

There are two methods used in common practice to analyse the coal: ultimate analysis and proximate analysis. The ultimate analysis is made to determine all component elements of coal, namely solid or gaseous ones. The proximate analysis of coal is made to determine the fixed carbon, volatile matter, moisture and ash percentages. The ultimate analysis can be done in a laboratory by a skilled chemist, whereas the proximate analysis can be determined with a simple apparatus.

Proximate analysis

Proximate analysis indicates the percentage by weight of the fixed carbon, volatiles, ash and moisture content in coal. The amounts of fixed carbon and volatile combustible matter directly contribute to the heating value of coal. The higher the percentage of fixed carbon, the greater the heat generated during burning. High volatile matter content indicates easy ignition of fuel. The percentage of ash is considered while designing the furnace grate, combustion volume, pollution control equipment and ash-handling systems. Table 2.7 shows a typical proximate analysis of various coals.

Table 2.7 *Typical Proximate Analyses of Various Coals (in Percentage)*

Parameter	Indian coal	Indonesian coal	South African coal
Moisture	5.98	9.43	8.50
Ash	38.63	13.99	17.00
Volatile matter	20.70	29.79	23.28
Fixed carbon	34.69	46.79	51.22

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(i) *Measurement of moisture*

Moisture percentage is found by placing a sample of powdered raw coal of 200-μ size in an uncovered crucible in an oven kept at 108 ± 2°C along with the lid. Then the sample is cooled to room temperature and weighed again. The loss in weight represents moisture.

(ii) *Measurement of volatile matter*

To measure the volatile matter, fresh sample of crushed coal is weighed, placed in a covered crucible and heated in a furnace at 900 ± 15°C. The sample is then cooled and weighed. The loss of weight represents moisture and volatile matter. The remainder is fixed carbon and ash known as coke.

(iii) *Measurement of carbon and ash*

To measure the content of carbon and ash, the cover from the crucible used in the volatile matter test is removed, and the crucible is heated over the Bunsen burner until all the carbon is burnt. The residue is weighed, which yields incombustible ash. The difference in weight of the remainder of volatile matter test and this test gives the fixed carbon. In practice, fixed carbon or FC is determined as follows:

$$FC = 100 - (\% \text{ of moisture, volatile matter and ash})$$

Significance of various parameters in proximate analysis

(a) *Fixed carbon*

Fixed carbon is the solid fuel left in the furnace after volatile matter is distilled off. It consists mostly of carbon in addition to traces of hydrogen, oxygen, sulphur and nitrogen not driven off with the gases. Fixed carbon gives a rough estimate of the heating value of coal.

(b) *Volatile matter*

Volatile matters contain gaseous methane, hydrocarbons, hydrogen and carbon monoxide, and incombustible gases such as carbon dioxide and nitrogen found in coal. The volatile matter is an index of the gaseous form of fuels present and varies between 20 and 35 per cent.
Volatile matter

- ➤ Ensures easy ignition of coal and proportionately increases flame length
- ➤ Sets minimum limit on the furnace height and volume
- ➤ Influences secondary air requirement and distribution aspects
- ➤ Influences secondary oil support

(c) *Ash content*

It is an impurity that will not burn and is removed periodically from the furnace. It varies from coal to coal from 5 to 40 per cent.

Presence of ash results in the following ill effects:

- ➤ Forms clinkering and slagging of coal
- ➤ Influences combustion efficiency and boiler efficiency
- ➤ Reduced fuel-handling and burning capacity
- ➤ Increased fuel-handling costs

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(d) *Moisture content*

Moisture in coal accumulates during transportation, handling and storage. It replaces the combustible matter, and hence reduces the heat content per kilogram of coal. A typical range of moisture content is 0.5–10 per cent. The presence of moisture

- ➤ Increases heat loss, due to evaporation and superheating of vapour
- ➤ Helps, to a limit, in binding fines
- ➤ Aids radiation heat transfer

(e) *Sulphur content*

Sulphur content in coal ranges between 0.5 and 0.8 per cent normally. The presence of sulphur

- ➤ Affects clinkering and slagging tendencies
- ➤ Corrodes chimney and other equipments such as air heaters and economizer and ESPs
- ➤ Limits exit flue gas temperature

1. *Chemical properties*

The ultimate analysis is done in oder to determine the chemical properties. The analysis gives the various elemental chemical constituents such as carbon, hydrogen, oxygen, sulphur, etc., present in the fuel. It is useful in determining the quantity of air required for combustion and the volume and composition of the combustion gases. The data thus obtained are used for the calculation of flame temperature, flue duct design, etc. Typical ultimate analyses of various coals are given in Table 2.8.

Table 2.8 *Typical Ultimate Analyses of Coals*

Parameter	Indian coal (%)	Indonesian coal (%)
Moisture	5.98	9.43
Mineral matter (1.1 × ash)	38.63	13.99
Carbon	41.11	58.96
Hydrogen	2.76	4.16
Nitrogen	1.22	1.02
Sulphur	0.41	0.56
Oxygen	9.89	11.88

2.5.2 Indian Coals

Coal is the most important fossil fuel and is available in abundance in India, and it accounts for 55 per cent of India's energy need. In India, coal is largely mined in the eastern and the central regions of the country. Coal has been recognized as the most important source of energy for electricity generation in India. About 75 per cent of the coal is consumed in India in the power sector alone. In addition, other industries such as steel, cement, fertilizers, chemicals, paper and thousands of medium- and small-scale industries are also dependent on coal for their process and energy requirements.

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The coal reserves of India up to the depth of 1200 m have been estimated by the Geological Survey of India at 247.85 billion tonnes (bt) as on 1 January 2005 of which 92 bt are proven. Hard coal deposits spread over 27 major coalfields that are mainly confined to eastern and south central parts of India.

The lignite reserves in India are estimated at around 36 bt, 90 per cent of which is available in the southern State of Tamil Nadu. A total of 4150 million tonnes (mt) spread over 480 sq km is in the Neyveli Lignite fields in the Cuddalore District. Geological reserves of about 1168 mt of lignite have been identified in Jayamkondacholapuram of Trichy District of Tamil Nadu. In Mannargudi and East of Veeranam, geological reserves of around 22,661.62 mt and 1342.45 mt, respectively, of lignite have been estimated. Lignite deposits have been located in other states such as Rajasthan, Gujarat, Kerala, Jammu and Kashmir and the Union Territory of Pondicherry.

Despite various policy initiatives taken by the Indian Government to diversify the fuel mix in the backdrop of limited reserve potentiality of petroleum and natural gas, eco-conservation restriction on hydel project and geo-political perception of nuclear power, coal will continue to occupy centre stage of India's energy scenario. Indian coal offers a fuel source to domestic energy market for the next century and beyond. Based on estimates, the consumption of coal is projected to rise by nearly 40 per cent over the next five years and almost to double by 2020.

2.6 SELECTION OF COAL IN THERMAL POWER STATION

The common coals used in Indian industry are bituminous and sub-bituminous coal. The gradation of non-coking coal is based on useful heat value (UHV), the gradation of coking coal is based on the ash content; and for semi-coking/weakly coking coal, it is based on ash plus moisture content, as in vogue as per notification. The gradation of Indian coal based on the information of Ministry of coal, Government of India is as follows (Tables 2.9–2.12):

Table 2.9 *Grades of Coking Coal*

Grade	Ash content
Steel grade – I	Not exceeding 15%
Steel grade – II	Exceeding 15%, but not exceeding 18%
Washery grade – I	Exceeding 18%, but not exceeding 21%
Washery grade – II	Exceeding 21%, but not exceeding 24%
Washery grade – III	Exceeding 24%, but not exceeding 28%
Washery grade – IV	Exceeding 28%, but not exceeding 35%

Table 2.10 *Indian Coal Based on Grades and Calorific Value*

S. no.	Grade	Calorific value range (in kcal/kg)
1.	A	Exceeding 6200
2.	B	5600–6200
3.	C	4940–5600

(*Continued*)

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Table 2.10 (*Continued*)

S. no.	Grade	Calorific value range (in kcal/kg)
4.	D	4200–4940
5.	E	3360–4200
6.	F	2400–3360
7.	G	1300–2400

Normally, D, E and F coal grades are available to Indian industry.

Table 2.11 *Grades of Semi-Coking and Weakly Coking Coal*

Grade	Ash + moisture content
Semi-coking grade – I	Not exceeding 19%
Semi-coking grade – II	Exceeding 19%, but not exceeding 24%

Table 2.12 *Grades of NEC Coal*

Grades	HCV (kcal/kg)	Corresponding Ash % + moisture %
A	6200–6299	18.85–19.57
B	5600–6199	19.58–23.91

2.6.1 Geological Resources of Coal in India

Based on the exploration carried out up to the maximum depth of 1200 m by the GSI, CMPDI, SCCL, MECL, DGM (Maharashtra) and DGM (Chhattisgarh), etc., a cumulative total of 2,93,497 mt of Geological Resources of Coal have so far been estimated in the country as on 1 April 2012. The details of state-wise geological resources of coal are given as under (see Tables 2.13–2.14):

Table 2.13 *Gondwana Coalfields (in mt)*

	Geological resources of coal			
State	Proved	Indicated	Inferred	Total
Andhra Pradesh	9566.61	9553.91	3034.34	22,154.86
Assam	0	2.79	0	2.79
Bihar	0	0	160.00	160.00
Chhattisgarh	13,987.85	33,448.25	3410.05	50,846.15
Jharkhand	40,163.22	33,609.29	6583.69	80,356.20
Madhya Pradesh	9308.70	12,290.65	2776.91	24,376.26
Maharashtra	5667.48	3104.40	2110.21	10,882.09

(*Continued*)

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Table 2.13 (*Continued*)

State	Geological resources of coal			
	Proved	Indicated	Inferred	Total
Orissa	25,547.66	36,465.97	9433.78	71,447.41
Sikkim	0	58.25	42.98	101.23
Uttar Pradesh	884.04	177.76	0	1061.80
West Bengal	12,425.44	13,358.24	4832.04	30,615.72
Total	1,17,551.01	1,42,069.51	32,383.99	2,92,004.51

Source: Geological Survey of India.

Table 2.14 *Tertiary Coalfields ((in mt)*

State	Geological resources of coal				
	Proved	Indicated	Inferred (exploration)	Inferred (mapping)	Total
Arunachal Pradesh	31.23	40.11	12.89	6.00	90.23
Assam	464.78	42.72	0.50	2.52	510.52
Meghalaya	89.04	16.51	27.58	443.35	576.48
Nagaland	8.76	0	8.60	298.05	315.41
Total	593.81	99.34	49.57	749.92	1492.64

Source: Geological Survey of India.

2.6.2 Status of Coal Resources in India during the Past Five Years

Further, based on the regional, promotional and detailed exploration by GSI, CMPDI and SCCL, etc., the estimation of coal resources of India has reached to 2,93,497 mt. The estimates of coal resources in the country during the past five years are given in Table 2.15.

Table 2.15 *Geological Resources of Coal (in mt)*

As on	Geological resources of coal			
	Proved	Indicated	Inferred	Total
1 April 2007	99,060	1,20,177	38,144	2,57,381
1 April 2008	1,01,829	1,24,216	38,490	2,64,535
1 April 2009	1,05,820	1,23,470	37,920	2,67,210
1 April 2010	1,09,798	1,30,654	36,358	2,76,810
1 April 2011	1,14,992	1,37,471	34,390	2,85,862

Source: Geological Survey of India.

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2.7 QUESTIONS

2.7.1 Objective Questions

1. An Orsat apparatus is used to
   (a) determine the smoke density
   (b) analyse the wet products of combustion
   (c) analyse the dry products of combustion
   (d) determine the volume of excess air
2. Arrange the following liquid fuels in the ascending order based on density:
   (i) Gasoline (ii) Kerosene (iii) Diesel (iv) Heavy oil
   (a) 1, 2, 3, 4 (b) 4, 3, 2, 1 (c) 1, 3, 2, 4 (d) 2, 3, 4, 1
3. Orsat flue gas analysis gives
   (a) percentage by mass of dry products of combustion
   (b) percentage by volume of dry products of combustion
   (c) percentage by mass of fuel composition
   (d) percentage by volume of fuel composition
4. In Orsat apparatus, pyrogallic acid is used to absorb
   (a) CO_2 (b) O_2 (c) N2 (d) CO
5. In Orsat apparatus, KOH solution is used to absorb
   (a) CO_2 (b) O_2 (c) N_2 (d) CO
6. Bomb calorimeter is used to determine the CV of
   (a) solid fuels and liquid fuels (b) liquid fuels only
   (c) gaseous fuels (d) none of the above
7. Lewis Thomson calorimeter is used to determine the CV of
   (a) solid fuels (b) liquid fuels
   (c) gaseous fuels (d) none of the these
8. In Orsat apparatus, cuprous chloride solution is used to absorb
   (a) CO_2 (b) O_2 (c) N_2 (d) CO
9. In a Lewis Thomson calorimeter, oxygen is supplied for combustion by
   (a) potassium chlorate (b) potassium nitrate
   (c) both of them (d) none of these
10. In the presence of oxygen, heat is liberated by
    (a) potassium chlorate (b) potassium nitrate
    (c) both of them (d) none of these
11. Which of the following is a common constituent between water gas and producer gas?
    (a) CO_2 (b) H_2 (c) N_2 (d) CO

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Answers

1. c 2. a 3. b 4. b 5. a 6. a 7. a 8. d 9. c 10. a 11. b

2.7.2 Review Questions

1. List down the merits and demerits of liquid fuels over solid fuels.
2. Discuss the advantages and disadvantages of gaseous fuels over liquid fuels.
3. Define calorific value of a fuel.
4. Differentiate between HCV and LCV of a fuel and state which one of the two is used in practical calculations and why.
5. What are the methods used for finding the e of a fuel? Explain any one method with a neat sketch.
6. The percentage composition by weight of a coal used in power plant is C = 90%, H_2 = 3.3%, O_2 = 3%, N_2 = 8%, S = 0.9% and ash = 2%.
   (a) Calculate the minimum quantity of air required to burn 1 kg of coal completely.
   (b) If 50 per cent excess air is supplied, then determine the percentage composition of the dry flue gases by volume.

   Minimum volume of air required for complete combustion of 1 m^3 of fuel
7. The dry product of combustion of an IC Engine has the following volumetric analysis

   $CO_2 \rightarrow 11\%$; $CO \rightarrow 3\%$; $CH_4 \rightarrow 4\%$; $H_2 \rightarrow 1\%$; $O_2 \rightarrow 5\%$ and $N_2 \rightarrow 76\%$

   Assuming the fuel used is a pure hydrocarbon and nothing was left unburnt. Calculate the percentage of carbon and hydrogen by mass in fuel.
8. The percentage composition by mass of a crude oil is given as follows:

   $C \rightarrow 90\%$, $H_2 \rightarrow 3.3\%$, $O_2 \rightarrow 3\%$, $N_2 \rightarrow 0.8\%$ and $S \rightarrow 0.9\%$ and the remaining is incombustible. If 50 per cent excess air is supplied, find the percentage of dry exhaust gases formed by volume.
9. A hydrocarbon fuel used in IC engine contains $C \rightarrow 85\%$ and $H_2 \rightarrow 15\%$. It is burnt with 60 per cent excess air. Engine develops 150 kw with brake thermal efficiency 30 per cent cv of fuel used = 44,500 kg/kg. Calculate the air supplied to the engine/min.
10. The volemic composition of a coal gas is given as H_2 = 27%, CO = 7%, CH_4 = 48%, $C_2H_4|$ = 13%, CO_2 = 3%, and N_2 = 2%. If 50 per cent excess air supplied, find
    (a) The percentage composition by volume of the products assuming H_2O produced is perfect gas
    (b) The percentage contraction in column on combustion for uncoiled gases and for cooled gases.
11. In a CI engine, $C_{12}H_{26}$ is used as a fuel. At full-load condition, 35 per cent excess air is consumed. The pressure and temperature of air at the start of ignition arc are 35 bar and 500°C, respectively. If the combustion takes place at constant pressure without any heat loss, calculate the temperature at the end of combustion. Neglect the effect of dissociation and variation of specific heats. Take the values of CP of the products as CO_2 = 0.865 kg/kg K; H_2O = 1.98 kg/kg K; O_2 = 0.91 kg/kg K; and N_2 = 1.33 kg/kg K CV of fuel used = 44 MJ/kg.
12. The Orsat analysis of the products of combustion of unknown hydrocarbon fuel is given below:

    $CO_2 \rightarrow 12.2\%$, $CO \rightarrow 0.9\%$, $O_2 \rightarrow 3.8\%$ and $N_2 \rightarrow 83\%$. Calculate (a) composition of fuel (b) A:F ratio and (c) Excess or deficiency of air.

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13. Find an expression for the height of air supplied per kilogram of fuel when Carbon context of the fuel and the volumetic analysis of the fuel gas is known.

    Show that carbon burnt to CO per kilogram of fuel is $\frac{(CO)(C)}{CO+CO_2}$, where CO and CO_2 represent the percentage of carbon monoxide and carbon dioxide in dry flue gas by volume and C represents the percentage by weight of carbon that is actually burnt.

14. In a boiler trial, the analysis of the coal used is as follows: C = 20%, H_2 = 4%, O_2 = 6% and the remainder being incombustible matter. The dry flue gas has the following composition by volume:

    CO_2 = 9%, CO = 2.5%, N_2 = 80% and O_2 = 8.5%, determine

    (a) minimum mass of air required per kilogram of coal

    (b) percentage of excess air

15. The percentage composition of a sample of fuel was found to be C = 83%, H_2 = 9%, S = 2%, O_2 = 3.5% and ash = 2.5%. For an air fuel ratio of 13:1, calculate

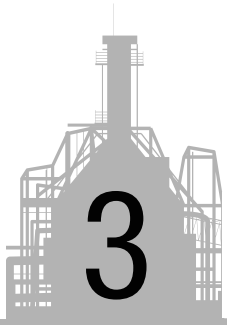
    (a) the mixture strength as a percentage rich or weak

    (b) the volumetric analysis of the dry products of combustion

16. A fuel contains by mass, C = 86%, H_2 = 4%, O_2 = 4.5% and the remaining is ash. Determine the minimum quantity of air required for complete combustion of 1 kg of fuel. If the actual air supplied is 1.5 times the minimum air required for complete combustion, estimate percentage volumetric composition of dry gases.

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Fuel-Handling Systems



3

Contents

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3.1 COAL HANDLING

It is absolutely essential to have an efficient fuel-handling system in power plants. Since a majority of the power plants operate using coal as a fuel, it is necessary to study about coal-handling system. Coal can be handled either *manually* or *mechanically*. But, with respect to the capacity of modern power plants, it is almost impossible to handle coal manually. That means *mechanical handling of coal is inevitable*.

3.1.1 Outplant Handling of Coal

With the commissioning of more and more numbers of large-capacity steam power plants, it is very essential to have an effective means of transporting coal from coal mines to the plant site and subsequently store it to meet emergency situations. Outplant handling of coal mainly deals with *transporting* coal from coal mines to the storage area of the plant by *sea* or *river*, or *rail* or *road*.

1. *Transportation by sea or river*

An ideal site for a power plant is near a water source. If the plant is situated near the sea or river, coal can be economically transported in bulk quantity by ships or boats or barges. After unloading the coal by mechanical means, coal is either stored in the plant site (coal yard) or conveyed to the plant.

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2. *Transportation by road*

This type of transportation system is ideal for transporting coal directly to the point of consumption. Such a means is suitable for supplying coal to a small capacity plant. Trucks or tippers are used to supply coal for this purpose.

3. *Transportation by rail*

If the power plant is accessible by railway, coal can be transported to the coal yard much more economically. The coal can be unloaded using cranes, bucket conveyers or wagon tippers.

3.1.2 Storage of Coal

To meet the continuous demand of coal and to face any emergencies arising out of short supply of coal, it is always desirable to have a coal storage yard near the plant. The capacity of such a yard may be about 10 per cent of annual consumption to a maximum of 25 per cent of annual consumption.

Even though coal storage is essential for the reasons mentioned above, still it is undesirable because of the following reasons:

(i) High tendency to ignite spontaneously
(ii) Higher rate of deterioration due to oxidation
(iii) Higher maintenance cost involved to protect the area against natural calamities
(iv) Higher capital cost and interest on it in addition to insurance cost

The following precautions should be taken care of while storing coal:

Coal should be dumped (stored) on a solid ground completely free from standing water. It should be protected from direct sunlight preferably and should be stored using the following methods to reduce the chances of oxidation and combustion.

1. *Stocking in heaps*

Coal is piled in heaps of 10–12 m height on a concrete floor, by compacting it in layers of 20–30 cm thickness using bulldozers. This prevents air circulation in the interior of the heap. The heap top is given a gentle slope to drain off rain water immediately without washing the coal.

In another method, the heat of oxidation is removed by circulating air uniformly through layers. Air thus circulated carries away the excess heat, maintaining coal below its combustion temperature of 343 K. Oxidation can also be avoided by sealing the coal heap using asphalt, fine coal dust or bituminous powder.

2. *Underwater storage*

Dock basins are used to store coal underwater in order to prevent slow oxidation and subsequent combustion of coal.

3.1.3 Inplant Handling of Coal

For easy, smooth and rapid operation, mechanical handling system is embraced for inplant handling of coal. A flow chart of such a system is shown in Figure 3.1. Inplant handling begins right from the coal delivery point upto the furnace firing system. The following are guidelines for an inplant coal-handling system:

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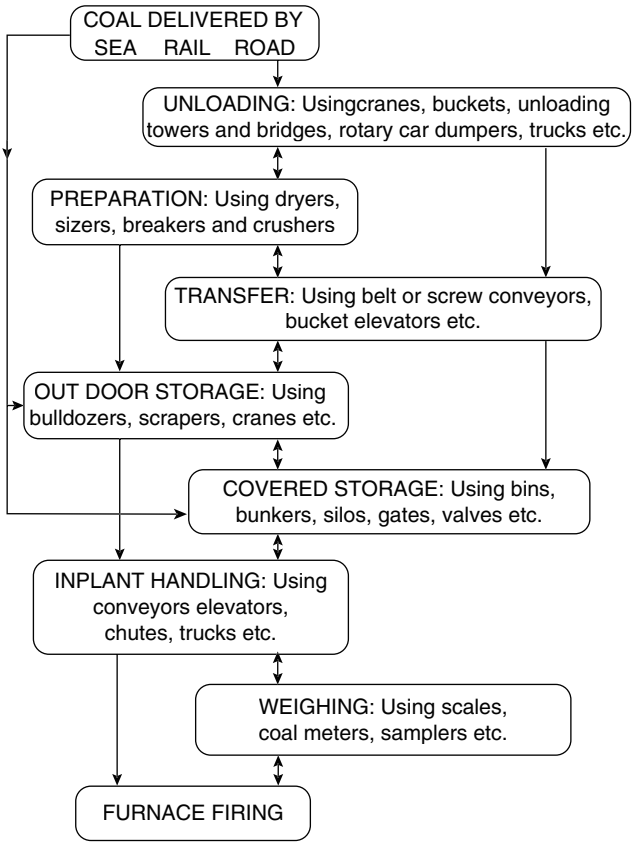


Fig. 3.1 *Stages in Coal Handling*

(i) It should be easy to handle, simple in construction and effective in working.

(ii) For easy inspection and maintenance, it should have centralized handling units.

(iii) It should have minimum handling operations and should be free from repetitive handling.

(iv) The working parts should be free from erosion and corrosion.

(v) Its capacity should be well above the requirement of coal during peak hours.

3.1.4 Inplant Handling System

This system feeds coal from live storage to the furnace using similar equipments as explained above. Coal delivered to the plant site is weighed on wagon balance and then delivered to underground hoppers or bunkers either manually or mechanically. The coal is now lifted to the transfer tower and then to the crusher through magnetic separators and screens. Coal is crushed to the required size (20–30 mm for stoker firing and 10–20 mm for pulverized fuel firing). In the

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case of pulverized firing, coal is further crushed into fine powder in mill and then carried to the transfer tower from where it is conveyed to a boiler bunker. From the boiler bunker, coal now flows by gravity to boiler hopper and ultimately to the boiler furnace. Depending on the load, fuel flow is regulated by using gates and valves in the system.

3.1.5 Stages of Coal Handling

The various stages of coal handling are shown in Figure 3.2.

1. *Unloading*

Depending on the mode of transport, namely, by sea or road or rail, coal unloading is done using the following equipments:

Coal towers, unloading bridges and self-unloading boats are used if the outplant handling of coal is by sea or river.

Car shakers, rotary car dumpers and coal accelerators are used if the mode of transport is by rail.

Lift trucks with scoop are used to unload the coal if the mode of transport is by road.

2. *Coal preparation*

Since coal is not properly sized initially, a coal preparation plant is needed to get the uniform size of the coal before it is fed to the plant. The preparation plants consist of the following equipments:

(i) *Coal crusher*

The function of the coal crusher is to crush the coal to uniform size (about 8–25 mm). The capacity of the unit depends on the size of the plant.

(ii) *Sizer*

The function of the sizer is to pick the coal of required size, which is coming from the crusher. The unsized coal is fed back to the crusher.

(iii) *Dryer*

The function of the dryer is to remove excess moisture present in the coal. The moisture is removed by passing hot flue gases through the cold storage in closed spaces.

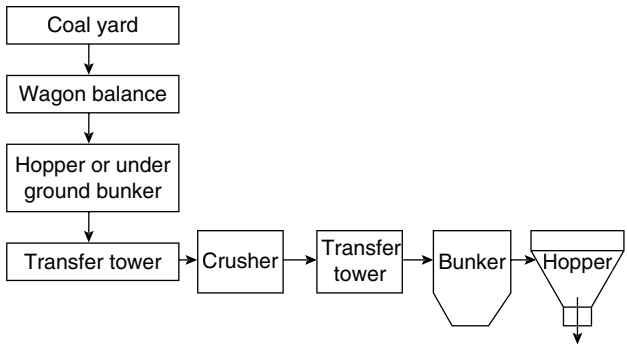


Fig. 3.2 *Inplant Handling System*

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(iv) *Magnetic separator*

The function of the separator is to remove iron particles, which otherwise may choke the burners besides damaging other equipments. The iron particles are trapped by the magnetic pulley while the belt carrying coal passes over it. It is collected in a chute.

A typical sketch of a coal preparation plant is shown in Figure 3.3.

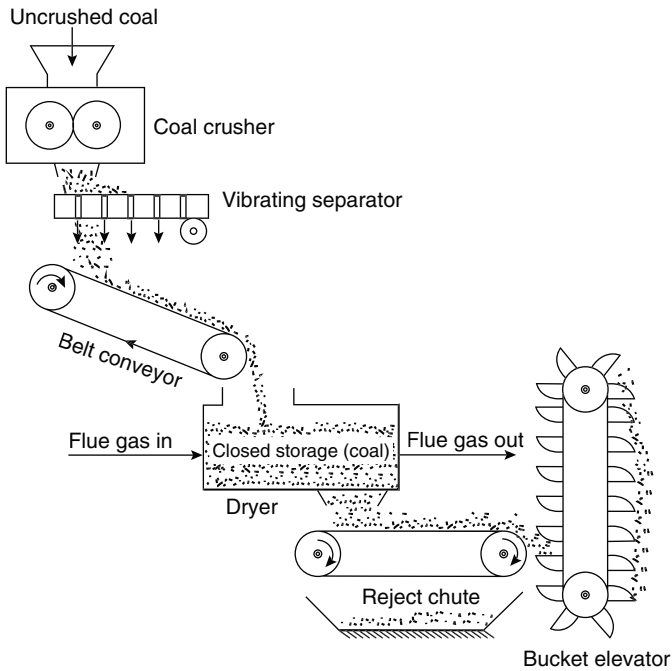


Fig. 3.3 *Coal Preparation Plant*

3. *Transfer of coal*

Coal transfer includes transferring of coal from unloading point to the storage site using the coal-handling equipments.

3.2 CHOICE OF HANDLING EQUIPMENT

Many coal-handling equipments are available, the choice of which depends on various considerations. Coal transfer from unloading point to the storage site can be done using the following equipments:

1. *Bucket elevators*

Figure 3.4 shows centrifugal and continuous types of bucket elevators used for lifting coal to moderate heights (about 30 m). It consists of buckets fixed to a chain moving over two sprockets. The buckets carry coal from the bottom and discharge to the top at a speed ranging between 15 and 36 m/min. The continuous type of elevator is used to convey more amount of coal.

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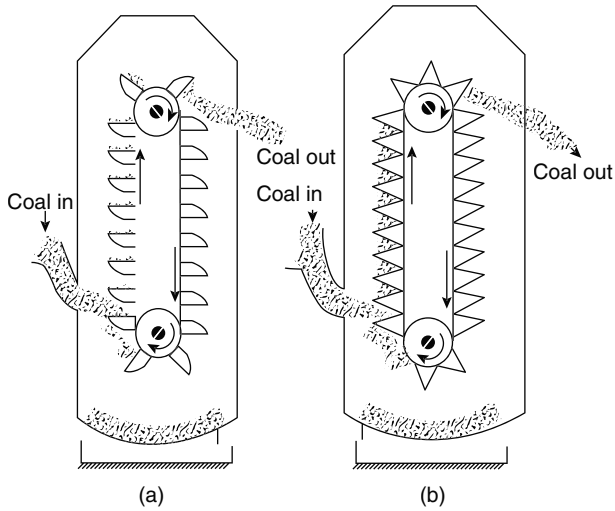


Fig. 3.4 *Bucket Elevators: (a) Centrifugal; (b) Continuous*

2. *Grab bucket conveyor*

Figure 3.5 shows a grab bucket conveyor that can grab as well as convey coal from one place to another. It is used with a crane or tower and it has different capacities. It is very essential when all other means of transporting coal are not viable. It has high initial cost but low operating cost.

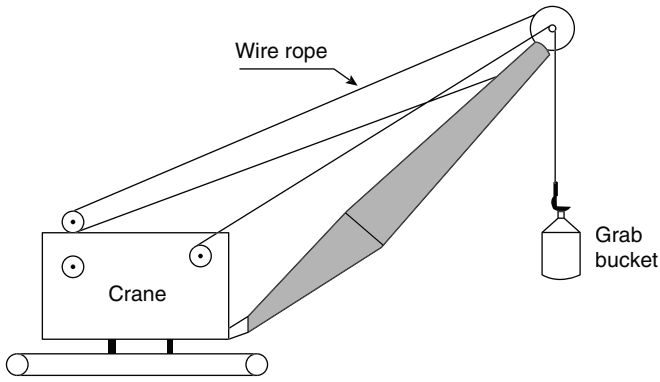


Fig. 3.5 *Grab Bucket Conveyor*

3. *Belt conveyor*

Figure 3.6 shows section through a belt conveyer used for transporting large quantities of coal over long distances. It consists of an endless belt made of *rubber*, *balata* or *canvas* running over pulleys or end drums. The belt is supported by a series of rollers known as idlers at regular

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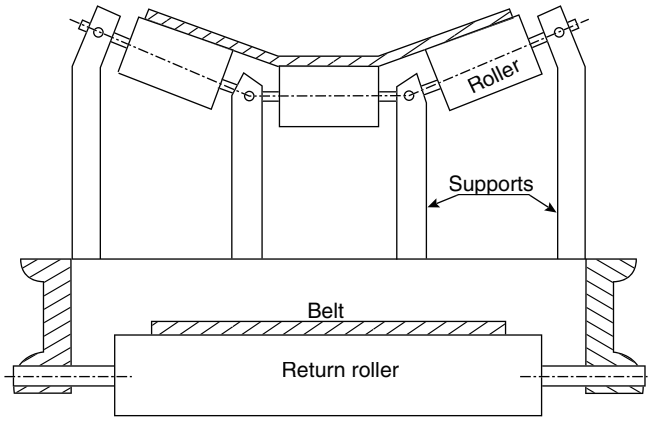


Fig. 3.6 *Belt Conveyer*

intervals. The empty belt is supported by return rollers as shown. These conveyers can be used with inclination up to 20° to the horizontal moving at an average speed of 60–90 m/min. Upto 50–100 t/h of the load can be carried over a distance of about 500 m.

It is an economical means of coal transportation at a rapid rate with minimum maintenance cost. By providing overhead covers, coal can be protected by rain, wind, etc. However, it cannot be used for carrying coal at greater heights.

4. *Screw conveyor*

Figure 3.7 shows a screw conveyer used to transmit coal. It consists of an endless helicoid screw of diameter 150–500 mm wound around a rotating shaft. One end of the shaft is connected to the drive, whereas the other end is supported by a bearing. The screw along with the shaft rotates at about 60–120 rpm and carries coal from one end to another. Screw conveyors are designed to suit different load capacities.

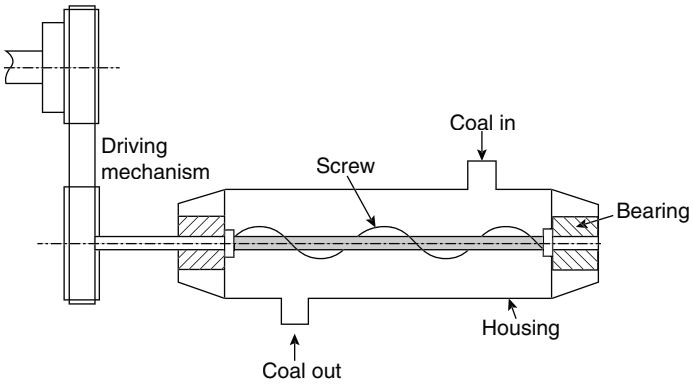


Fig. 3.7 *Screw Conveyor*

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5. *Flight or scraper conveyers*

Figure 3.8 shows a flight conveyer used for filling a large number of storage bins situated under the conveyor. It consists of steel scrapers attached to chain mechanism. Steel scrapers scrap the coal through a trough and discharge it at the bottom in to storage bin. Due to excessive wear and tear of the scrapers, this conveyor has shorter life span.

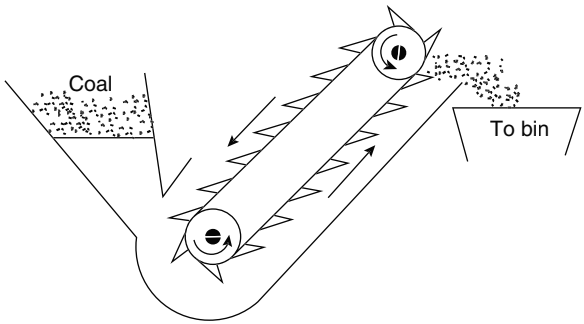


Fig. 3.8 *Scraper Conveyor*

3.3 FUEL BURNING

One of the most important factors in the economical working of a power plant is the efficient combustion of fuel. Two commonly used methods for burning the fuel are *stoker firing* and *pulverized fuel firing*.

Methods of fuel firing: The solid fuels are fired into the furnace by the following methods:

1. Hand firing
2. Mechanical firing

1. *Hand firing*

This is a simple method of firing coal into the furnace. It requires no capital cost.

2. *Mechanical (stoker) firing*

This is used in medium- and large-size power plants. Operating cost is high.

3.3.1 Overfeed and Underfeed Fuel Bed Stokers

Stoker is a power-operated fuel feeding mechanism. The working of different types of stokers is based on the following two principles:

1. *Overfeed principle*

Figure 3.9 shows the operating principle of an overfeed stoker. The primary air enters the grate from the bottom. The air while moving through the grate openings gets and picks up additional energy. The air then passes through a layer of incandescent coke where O_2 reacts to form CO_2 and H_2O vapour reacts with coke to form CO_2, CO and free H_2. The gases leaving the incandescent coke consist of CO_2, CO, H_2, N_2 and H_2O and volatile matter of raw fuel. Then additional air known as secondary air is supplied to burn the combustible gases. The combustion of gases

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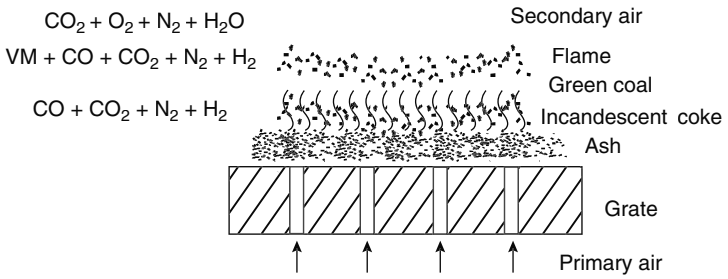


Fig. 3.9 *Overfeed Stoker Principle*

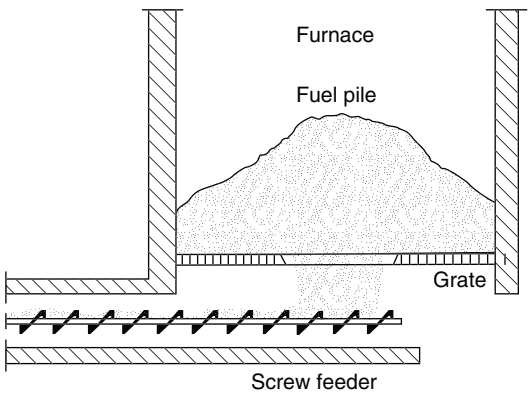


Fig. 3.10 *Schematic of an Overfeed Stoker*

entering the combustor is not complete. The combustion of the remaining combustible gases is completed in the combustion chamber. A schematic diagram of an overfeed stoker is shown in Figure 3.10.

The advantages and disadvantages of this stoker are as shown below:

Advantages

(i) The clinkering difficulties are reduced even with coals having high clinkering tendencies, by the spreading action.

(ii) A wide variety of coal varying from lignite to semi-anthracite as well as high ash coal can be burnt easily.

(iii) The coking tendency of the coal is reduced before it reaches the grate by the release of volatile gases, which burn in suspension.

(iv) It gives quick response to load change similar to pulverized fuel system because there is only a small amount of fuel on the grate at any time and most of the heat is released during burning of the coal in suspension.

(v) The use of high-temperature preheated air is possible.

(vi) This form of fixing provides thin and even fire bed and results in a high rate of combustion (350 kg/m^2 h). Therefore, it gives quick response of the load change and with less sensitivity to the swelling characteristics of the fuel.

(vii) Its operation cost is considerably low, as 0.6 kW/h fired.

(viii) Due to fire bed equal pressure drop and proper air distribution, combustion can be completed with minimum quantity of excess air.

Disadvantages

(i) It is always difficult to operate spreader with varying sizes of a coal and with varying moisture content.

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(ii) A natural result of suspension burning of fine fuel particles is the entrainment of ash in the products of combustion. To avoid the nuisance of fly ash, a dust collector is essential with this stoker.

(iii) Many fine unburnt carbon particles are also carried with the exhaust gases, and it is necessary to trap these and return to the furnace for burning. Otherwise, it would add as a loss to the combustion system.

2. *Underfeed principle*

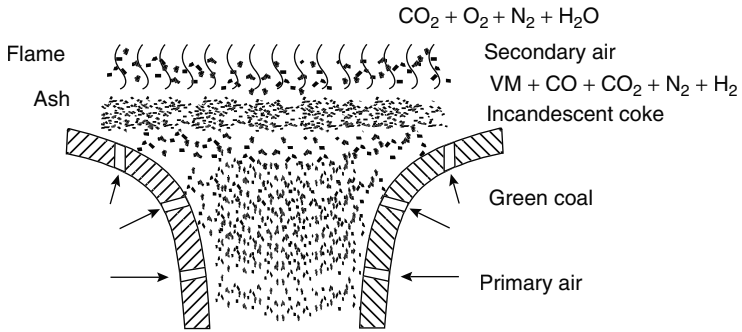


Fig. 3.11 *Underfeed Stoker Principle*

Figure 3.11 shows underfeed principle. In this case, air entering through the holes in the grate comes in contact with the volatile matter in raw coal is given off of by distillation. Then it passes through the incandescent coke where reactions similar to overfeed stoker take place. The gases produced then pass through a layer of ash. The secondary air is supplied to burn the combustible gases. The advantages and disadvantages of the underfeed stokers. Figure 3.12 shows a schematic diagram of an overfeed stoker.

Advantages

(i) This gives higher thermal efficiency compared with chain grate stokers.

(ii) The part load efficiency is high with multiple retort stoker.

(iii) The combustion rate is considerably higher.

(iv) Sufficient amount of coal always remains on the grate so that the combustion is continued in the event of temporary breakdown of the coal supply system.

(v) The grate is self-cleaning.

(vi) Different varieties of coals can be used with this type of stoker.

(vii) Tuyeres, grate bars and retorts are not subjected to high temperature; they remain always in contact with fresh coal.

(viii) The use of forced draft and relatively large quantities of fuel on the stoker make them responsive to rapid changes in load.

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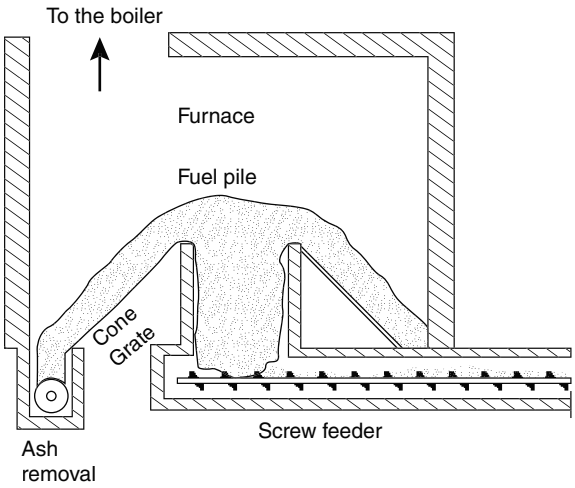


Fig. 3.12 *Schematic of an Underfeed Stoker*

(ix) The coal is continuously agitated by the plunger and pusher plates and due to this, the fuel bed remains porous and free from clinkers.

(x) Smokeless operation is possible even at very light load.

(xi) It can be used with all refractory furnaces because of non-exposure of stoker mechanism to the furnace. Under the existing conditions in the furnace of this type, it would not be possible to use other types.

(xii) Underfeed stokers are suitable for non-clinkering high volatile and low ash content coals.

Disadvantages

(i) The initial cost of the unit is high.

(ii) It requires large building space.

(iii) The clinker troubles are usually present.

(iv) Low-grade fuels with high ash content cannot burn economically. The anthracite coals with relative low ash fusion temperatures are not suited to the underfeed stoker.

3.4 EQUIPMENT FOR BURNING COAL IN LUMP FORM

Figure 3.13 shows how stokers based on the above principles can be further classified.

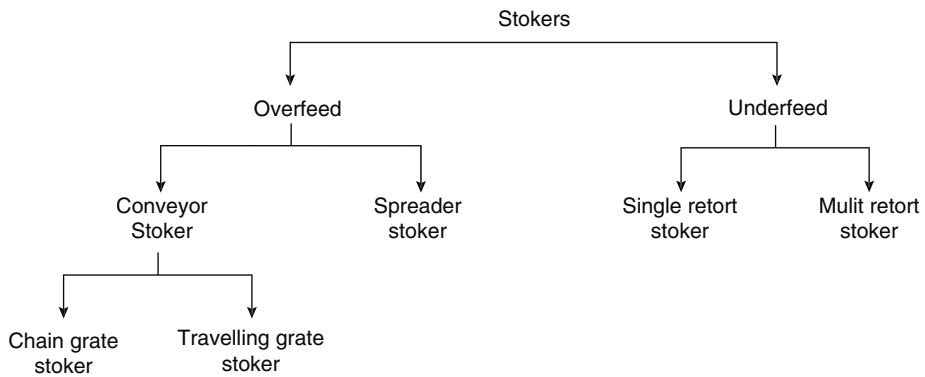


Fig. 3.13 *Types of Stokers*

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3.4.1 Chain Grate Stoker

Chain grate stoker consists of an endless chain, which forms a support for the fuel bed. The chain travels over two sprocket wheels one at the front and one at the rear of the furnace. The chain receives coal at its front through a hopper and carries it into the furnace. The ash is carried over the rear end of the stoker and deposited in the ash pit. The air enters through the air inlets situated below the grate. Initial cost is high and operations and maintenance cost are low.

3.4.2 Travelling Grate Stoker

It differs from chain grate stoker only in grate construction. It carries small grate bars, which actually support the fuel bed.

The stokers are suitable for low ratings because the fuel must be burnt before it reaches the rear of the furnace.

The travelling stoker may be chain grate type or bar grate type. These two differ only in the details of grate construction. The grate surface of a bar grate stoker is made of a series of cast iron sections mounted on carrier bars. The carrier bar rides on two endless-type drive chains.

A travelling-type chain grate is shown in Figure 3.14. The chain grate stoker consists of an endless chain that forms a support for the fuel bed. The chain travels over two sprocket wheels: one at the front and other at the rear of furnace as shown in figure. The front sprocket is connected to a variable speed drive mechanism.

The coal is fed by gravity from a hopper located in front of the stoker. The depth of the fuel on the grate is regulated by a hand-adjusted gate as shown in figure. The speed of the grate varies at the rate at which the coal is fed to the furnace. The combustion control automatically regulates the speed of the grate to maintain the required steam generation rate. The ash containing a small amount of combustible material is carried over the rear end of the stoker and deposited in the ash pit as shown in Figure 3.14.

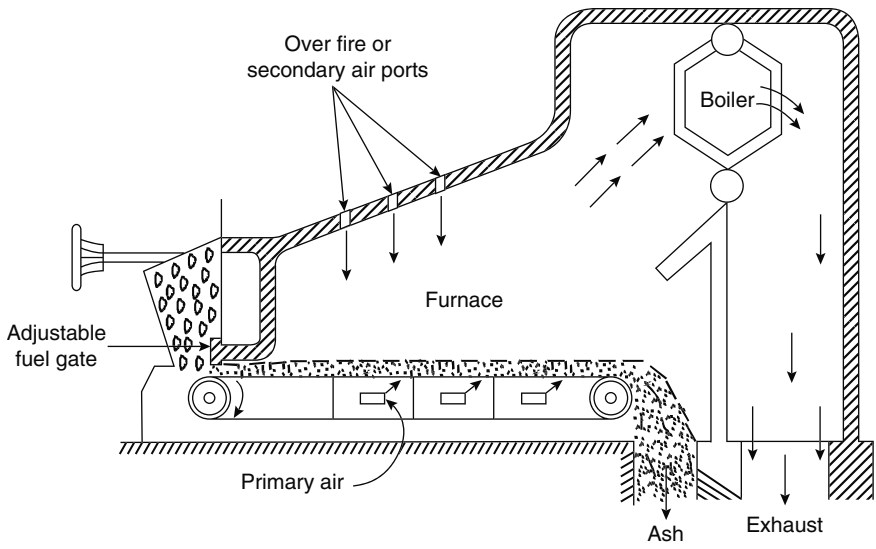


Fig. 3.14 *Travelling Chain Grate Stoker*

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The air required for combustion is supplied through the air inlets situated below the grate. The secondary air is supplied through the openings provided in the furnace well above the grate. The combination of primary air and overfire air supplied provide turbulence required for rapid combustion. The primary air is brought in from the sides and then forced through the upper grate. The air-ducts under the stoker are divided into sections. The air supply to different parts of the stoker is regulated to meet the changing demand through these sections. The air openings in the grate depend on the kind of coal burned and vary from 20 to 40 per cent of the total grate area. Air dampers are provided to control air supply to the various zones. The operator controls the rate of burning in different zones to minimize the coke carried over into the ash pit, or coal supplied to the grate is regulated by varying either the depth of coal on the grate with the help of grate valve or the rate of grate travel. These grates are suitable for low rating of fuel because the fuel must be burnt before it reaches the rear end of the furnace. The rate of burning with this stoker is 200–300 kg/m^2/h when forced drought is used. Any type of fuel except caking bituminous coal can be used with chain grate stoker because of the formation of large percentage of fine particles resulting in increased carbon loss.

The advantages and disadvantages of chain grate stoker are listed below:

Advantages

(i) It is simple in construction and its initial cost is low.
(ii) It is more reliable in service, therefore maintenance charges are low.
(iii) It gives high heat release rates per unit volume of the furnace.
(iv) The heat release rates can be controlled just by controlling the speed of chain.
(v) It is self-cleaning stoker.

Disadvantages

(i) The amount of coal carried on the grate is small as the increase in grate size creates additional problems. This cannot be used for high-capacity boilers 200 t/h or more.
(ii) The temperature of preheated air is limited to 180°C.
(iii) The clinker troubles are very common.
(iv) Ignition arches are required.
(v) There is always some loss of coal in the form of fine particles carried with the ashes.

3.4.3 Spreader Stoker

Figure 3.15 shows a spreader stoker. This is an overfeed-type stoker. The coal burns on this stoker remains partly in suspension and partly on the grate as shown in Figure 3.15.

In this stoker, coal from the hopper is fed on to a feeder, which measures the coal in accordance to the requirements. Feeder is a rotating drum fitted with blades. From the feeder the end drops on to spreader or distributor, which spreads the coal over the furnace. The spreader stoker distributes the coal evenly over the entire grate area. The spreader speed depends on the size of coal.

The spreader stoker consists of variable feeding device, a mechanism for throwing the coal uniformly on the grate and suitable openings for admitting the air. The coal feeding and distributing mechanism is located in the front wall above the grate. A portion of coal supplied containing fine

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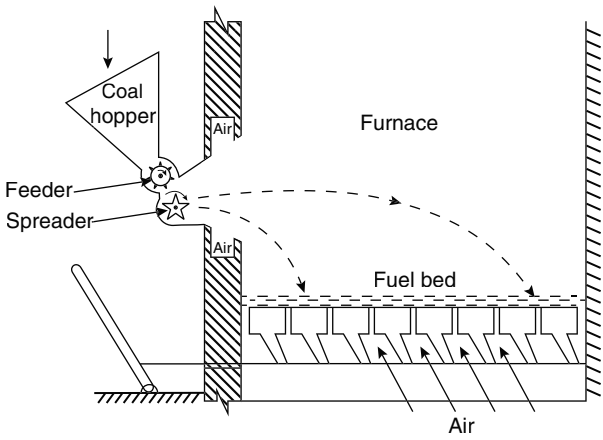


Fig. 3.15 *Spreader Stoker*

particles of coal burns in suspension and remaining falls to the grate. The air supplied by FD fan enters the furnace through the openings provided in the grate. A portion of this air is used to burn the fuel on the grate and the remaining air is utilized to burn the volatile matter and fine particles in suspension. Overfire or secondary air is supplied through the nozzle. The secondary air creates high turbulence and completes the combustion of volatile matter and fine particles of the coal. The unburnt coal and ash are deposited on the grate, which should be removed periodically.

The feeder is a slow-speed rotating drum on which a large number of small blades is mounted. It supplies coal to the spreaders in a continuous stream. The speed of the feeder can be adjusted as per the load on the plant. The feeders are operated with variable speed drive to control the combustion as per requirement. The feeders may be reciprocating ram, endless belt or spiral worm.

The spreader consists of a rapidly rotating shaft carrying blades. These blades are twisted to provide uniform distribution of the coal over the grate. The fast rotating blades hit the coal particles coming from the feeder and throw it into the furnace. The distribution of coal over the grate depends on the rotating speed of the spreader and on the size of the coal. Variations in performance of the spreader due to the changes in coal size and moisture content can be done by means of an external hand adjustment of the mechanism.

The selection of the size of the coal used in spreader stoker should be in between 6 and 36 cm. Stationary grates are used up to 10 MW capacity plant while moving grates are used in the range of 10–30 MW capacity plant.

The spreader stoker has wide applications with respect to the fuels used as well as to the boiler sizes. A wide variety and poor quality coal can be burnt efficiently with this type of stoker. This type of stoker can be used for boiler capacities from 80 to 150 tons of steam per hour. The heat release rate of 40×10^6 kJ/m^2 h is possible with stationary grate and 80×10^6 kJ/m^2 h is possible with travelling grate.

Advantages

(i) A wide variety of coal varying from lignite to semi-anthracite as well as high ash coal can be burnt easily.

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(ii) The clinkering problems are reduced even with coals that have high clinkering tendencies due to the spreading action.

(iii) The high-temperature preheated air can be used.

(iv) The coking tendency of the coal is reduced before it reaches the grate by the release of volatile gases, which burn in suspension.

(v) This system provides thin and even fire bed and results in high rate of combustion (350 kg/m^2 h) giving quick response to the load change and with less sensitivity to the swelling characteristics of the fuel.

(vi) It gives quick response to load change similar to pulverized fuel system due to small amount of fuel on the grate at any time and most of heat is released during burning of the coal in suspension.

(vii) The fire bed gives equal pressure drop and proper air distribution so that combustion can be completed with minimum quantity of excess air.

(viii) Its operation cost is considerably low.

Disadvantages

(i) It is difficult to operate the system with varying sizes of coal and with varying moisture content.

(ii) Due to suspension, burning of fine fuel particles is the entrainment of ash in the products of combustion; a dust collector is necessary with this stoker.

(iii) Many fine unburnt carbon particles are also carried with the exhaust gases resulting in a loss to the combustion system.

3.4.4 Retort Stoker

In underfeed stokers, the fuel is fed from underneath the fire and moves gradually upwards. The primary air is supplied from below where combustion takes place. The fuel releases the volatile matter as it passes through the initial fuel bed from bottom. The released volatile matter mixes with fresh air and enters into the combustion zone. The entire combustion process is highly efficient and gives high rates of heat release. Bituminous and semi-bituminous coals with small ash content and fusing temperature above 1300°C (caking or non-caking) can be burnt very effectively using these stokers.

The underfeed stokers can be divided into the following two main types:

1. Single retort stoker
2. Multi-retort stoker

1. *Single retort stoker*

Figure 3.16 shows the arrangement of a single retort stoker. The fuel is placed on a large hopper on the front of the furnace. Then it is further fed by reciprocating ram or screw conveyor into the bottom of the horizontal trough. The air is supplied through the tuyeres provided along the upper edge of the grate. The ash and clinkers are collected on the ash plate provided with dumping arrangement. The coal-feeding capacity of a single retort stoker varies from 100 to 2000 kg/h.

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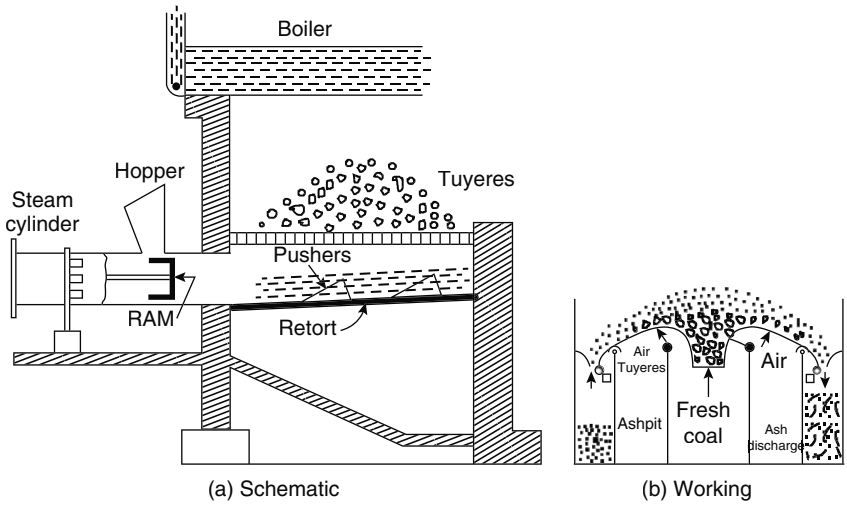


Fig. 3.16 *Single Retort Stoker*

Due to the inability of obtaining even air distribution from the sides of retorts, multi-retort stokers are generally used for increasing the burning capacity of the stoker.

2. *Multi-retort stoker*

The multi-retort stoker is shown in Figure 3.17. It consists of a series of alternate retorts and tuyere boxes for supply of air. Each retort is fitted with a reciprocating ram for feeding and pusher plates for the uniform distribution of coal. The coal falling from the hopper is pushed forward during the inward stroke ram. Then the distributing rams (pushers) push the entire coal down the length of the stoker. The ash formed is collected at the other end as shown in figure. The number of retorts may vary from 2 to 20 with local burning capacity ranging from 300 to 2000 kg/h/retort.

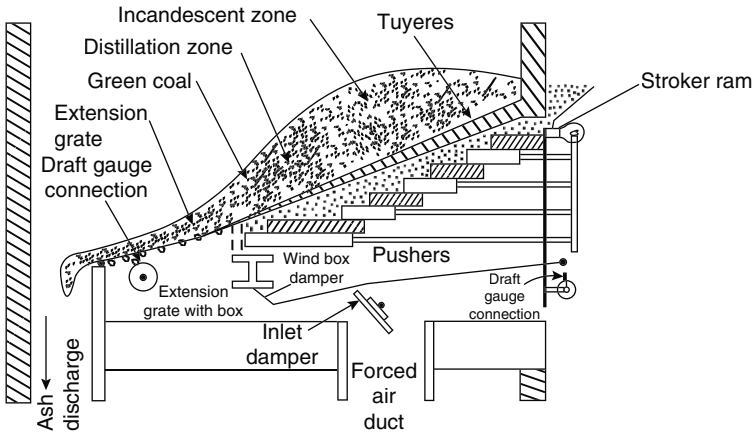


Fig. 3.17 *Multi-Retort Stoker*

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Underfeed stokers use forced draft for maintaining sufficient air flow through the fuel bed. The primary air is supplied from main wind box to the fuel bed situated below the stoker. The partly burnt coal moves on to the extension grate. The low-pressure air enters into the extension grate and wind from main wind box is supplied to the thinner fuel bed on the extension grate. The quantity supplied can be regulated by an air damper. The air pressure in the main wind box under the stoker is also varied to meet variable load. Provision is also made for varying the air pressure under the different sections of the stoker in order to correct for irregular fuel-bed conditions. Forced draft causes rapid combustion. It is also necessary to introduce 'overfire air' when high volatile coals are used to prevent the smoke formation. Combustion control is introduced into the stoker drive by either varying the ram stoke or changing the rate of reciprocation. On–off control is used on motor-driven stokers.

3.5 ADVANTAGES AND DISADVANTAGES OF STOKER FIRING OVER PULVERIZED SYSTEM OF FIRING

Advantages

(i) There is no necessity of coal preparation plant as the coal obtained from mines can be directly used. Sometimes it is necessary to size (crush) the coal in order to suit the furnace conditions.

(ii) This can be used for medium capacity plant more economically.

(iii) It is free from danger of explosion.

(iv) The building space requirement compared with pulverized system is less.

(v) The capital investment as compared to pulverized system is less.

(vi) The maintenance and operating costs are less.

(vii) The auxiliary plant required is considerably reduced.

(viii) It also works for few hours in the event of coal-handling plant failure as large amounts of coal are stored on the grate.

(ix) The dust collection problems are less severe compared with pulverized system as most of the ash is removed from the grate.

(x) The stoker firing systems are more reliable.

Disadvantages

(i) The loss of coal is more through riddling.

(ii) There is heavy wear and tear of moving parts due to abrasive action of coal.

(iii) The troubles of clinkering of combustion chamber walls are very common.

(iv) The sudden variations of load cannot be met to the same degree of efficiency as in the case of pulverized fuel firing.

(v) The furnaces need fire arches, which increase the construction cost and create troubles during operation.

(vi) Standby losses are considerably more.

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3.6 PREPARATION AND BURNING OF PULVERIZED COAL

Coal is pulverized (powdered) to increase its surface exposure thereby permitting rapid combustion. Coal is reduced to a fine powder in grinding mill or pulverizers.

There are two methods used to feed pulverized fuel to the furnace.

3.6.1 Unit or Direct System

Figure 3.18 shows the schematic diagram of a unit system. In the unit system, a separate pulverizing unit is provided for each furnace and the fuel is fired directly into the furnace without being stored. The raw coal from overhand coal bunker falls into a feeder and there it is dried by supplying hot air. The coal is then transferred to the pulverizing mill where it is pulverized. Primary air is supplied to the mill by the fan. The mixture of pulverized coal and primary air then flows to burner where secondary air is added.

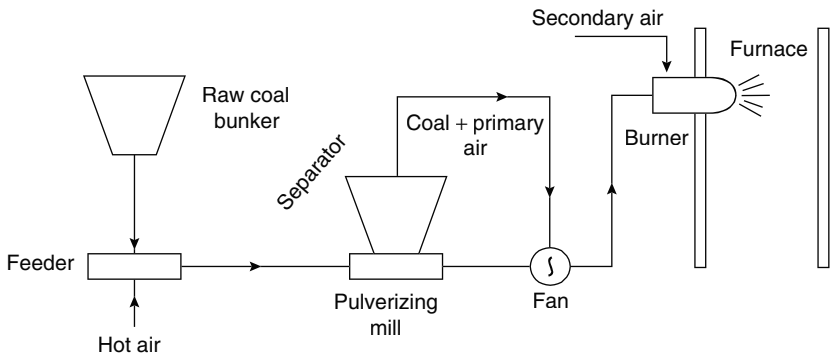


Fig. 3.18 *Unit System*

Advantages

(i) System is simple and cheaper.
(ii) Direct control of combustion from pulverizing mill is possible.
(iii) Coal transportation system is simple.

Disadvantages

(i) Mill operates at variable load.
(ii) System is less flexible.
(iii) No reserve.
(iv) Fan handles air + coal mixture.

3.6.2 Bin or Central System

Figure 3.19 shows the schematic diagram of a bin system. In central system, the coal is ground in a central grinding plant and stored in bins or bunkers. From the bunkers it is distributed in various burners through separate feeders in accordance with load demands.

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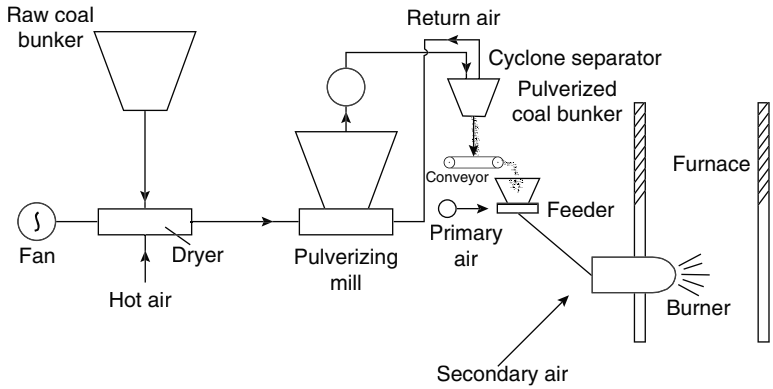


Fig. 3.19 *Bin System*

Coal from the raw coal bunker is fed by gravity to a dryer where it is dried by the hot air mill. The pulverized coal from the mill is transferred to pulverized coal bunker. The air from the coal is separated in the cyclone separator and returned to the mill. The primary air is mixed at the feeder and the mixture is supplied to the burner.

Advantages

(i) Pulverizing mill grinds the coal at steady rates irrespective of the load.

(ii) There is always some coal in reserve. Thus, failure of the mill will not affect the coal fed to the burner.

(iii) Exhaust fan handles clean air.

(iv) For a given boiler capacity, pulverizing mill of smaller capacity will be required.

Disadvantages

(i) Initial cost of the system is high.

(ii) Coal transportation system is complicated.

(iii) System requires more space.

3.6.3 Advantages and Disadvantages of Pulverized Coal Burning

The advantages and disadvantages of pulverized coal burning are as follows:

Advantages

(i) By breaking, a given mass of coal into smaller pieces exposes more coal surface area for combustion. The increase of surface area exposed per unit volume with the decreasing diameter of coal particle is shown in Figure 3.20. This increase varies rapidly after 0.01 inch diameter of the coal particle and there is no substantial increase after 0.001 inch diameter.

(ii) Greater surface area of coal per unit mass of coal allows faster combustion as more coal surface is exposed to heat and oxygen. This reduces the excess air required to ensure complete combustion and the fan power also.

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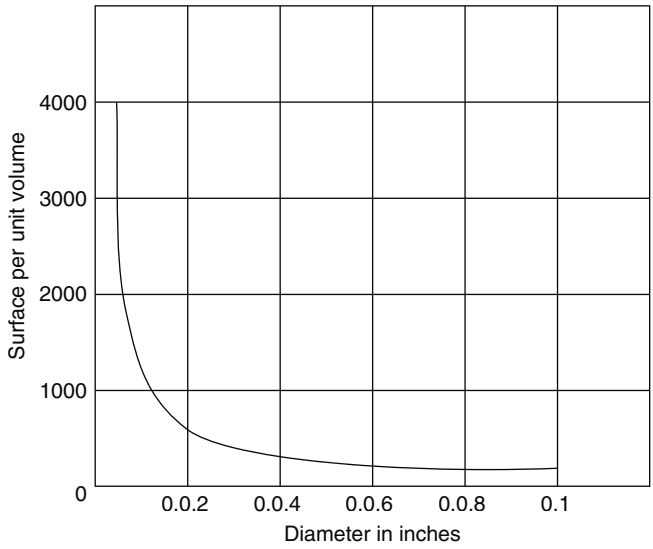


Fig. 3.20 *Surface Area versus Coal Size*

(iii) The system is perfectly free from clinker and slagging problem.

(iv) This system works successfully with or in combination with gas and oil.

(v) Large amount of heat release is possible and with such rate of heat generation; each boiler of pulverized fuel-fired system can generate as large as 2000 tons of steam per hour.

(vi) Wide variety and low-grade coal can be burnt more easily.

(vii) It gives fast response to load changes as rate of combustion can be controlled easily and immediately. Automatic control applied to pulverized fuel-fired boilers is effective in maintaining an almost constant steam pressure under wide load variations.

(viii) The external heating surfaces are free from corrosion and fouling as smokeless combustion is possible.

(ix) The banking losses are low compared with stoker firing system.

(x) The boilers can be started from cold very rapidly and efficiently.

(xi) Practically no ash-handling problems exist.

(xii) It is possible to use highly preheated secondary air (350°C), which helps in rapid flame propagation.

(xiii) The pulverizing system can be repaired without cooling the unit as the equipment is located outside the furnace.

(xiv) The furnace volume required is considerably less as the use of burners, which produce turbulence in the furnace, makes it possible to complete combustion with minimum travel of flame length.

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(xv) Rate of feed of the fuel can be regulated.

(xvi) The smaller quantity of excess air and thorough mixing of air and fuel produce a high furnace temperature with little smoke.

(xvii) The system can withstand high combustion air temperature.

(xviii) The life of the system is more and the operation is less troublesome due to the absence of moving parts in the furnace.

Disadvantages

(i) As additional equipment is essential to pulverize the coal, the initial cost is more.

(ii) Pulverized coal firing produces fly ash, which requires separate fly ash removal equipment.

(iii) The furnace has to be designed to withstand high temperature.

(iv) There will be a chance of explosion.

(v) Special equipment is required for starting the system.

3.7 PULVERIZED FUEL FURNACES (BURNERS)

Burners are used to burn the pulverized coal. The pulverized coal burners should satisfy the following requirements:

(i) It should mix the coal and primary air thoroughly and project the same in the furnace with secondary air.

(ii) It should create proper circulation and maintain stable combustion of coal.

(iii) It should control the flame shape and its travel in the furnace.

(iv) The mixture of coal and air should move away from the burner at a rate equal to flame travel in order to avoid the flash back with the burner.

(v) The burner should avoid overheating, internal fires and excessive abrasive wear.

3.8 PULVERIZED MILLS

There are various types of mills used for pulverizing the coal. These are:

(i) Ball mill

(ii) Ball and Race mill

(iii) Impact or Hammer mill

(iv) Bowl mill

1. *Ball mill*

The ball mill shown in Figure 3.21 consists of a large cylinder partly filled with various sized steel balls (2.5–5 cm in diameter). Coal (6 mm) is fed into the cylinder that mixes with these balls. Pulverization takes place as a result of action between the balls and the coal, as the cylinder is rotated (130 m/min peripheral velocity).

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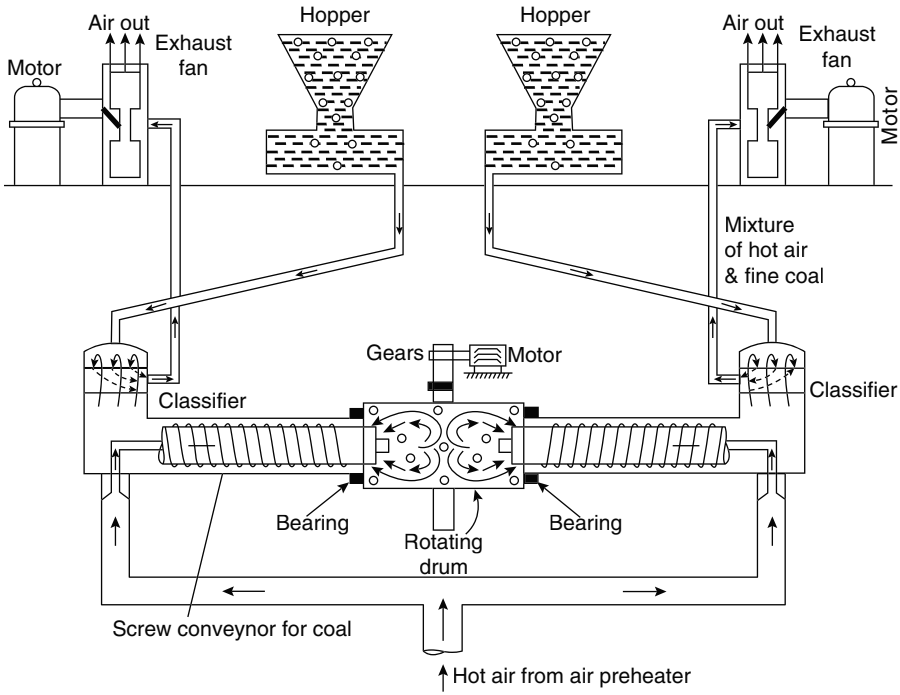


Fig. 3.21 *Double Classifier Ball Mill*

The mill consists of coal feeder, pulverizer, classifier and exhauster. The feeders supply coal to the classifier, which is further passed to the pulverizer with the help of screw conveyor. A mixture of tempering air and hot air from air-preheater is introduced in the pulverizer. These streams of air carry the pulverized coal and pass through the classifier. The oversized particles are thrown out of the air stream in the classifier and fine material is passed to the burner through exhaust fan.

The output of the mill can be controlled by the dampers located in the exhaust fan inlet duct. These dampers vary the flow of air through the mill, thus controlling the rate of fuel removed from the mill. The dampers are operated by the boiler's automatic combustion control. The feeder output is regulated by the coal level in the cylinder. When coal level in the cylinder attains sufficient height to seal off the lower channel, the differential control operates to stop the coal feed.

A ball mill capable of pulverizing 10 tons of coal per hour containing 4 per cent moisture requires 28 tons of steel balls and consumes 20–25 kW hour energy per ton of coal.

The principal features of this pulverizer are:

(i) The grinding elements are not seriously affected by metal scrap and other foreign material in the coal unlike the grinders in most other pulverizers.

(ii) There is considerable quantity of coal, which acts as a reservoir. This pulverizer prevents the fire from going out when there is slight interruption in fuel feed caused by coal clogging in the bunker.

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(iii) A wide range of fuels including anthracite and bituminous coal, which are difficult to pulverize, can be used in this mill.

(iv) The system is simple to operate, low in initial cost but has high operating cost.

2. *Ball and race mill*

This is also known as contact mill. It consists of two elements that have a rolling action with respect to each other. The coal passes between the rotating elements time and again until it has been pulverized to the desired degree of fineness. The pulverization is completed by a combination of crushing impact and attrition between grinding surfaces. The coal is crushed between two moving surfaces: balls and races. The upper stationary race and lower rotating race driven by a worm and gear hold the balls between them. The coal is supplied through the rotating race driven by a worm and gear, and hold the balls between them. The coal is supplied through the rotating table feeder at the upper right to fall on the inner side of the races. The moving balls and races trap coal between them to crush it to powder. Spring hold down the upper stationary race and adjust the force needed for crushing.

A forced draft fan supplies hot air to the mill through the annular space surrounding the races. The air picks up the coal dust as it flows between the balls and races and then enters into the classifier above. The fixed vanes make the entering air to form a cyclonic flow throwing the oversized particles to the wall of the classifier. The oversized particles slide down for further grinding in the mill. The coal particles of required size are taken to the burners with air from the top of the classifier. A typical Ball and Race mill is shown in Figure 3.22.

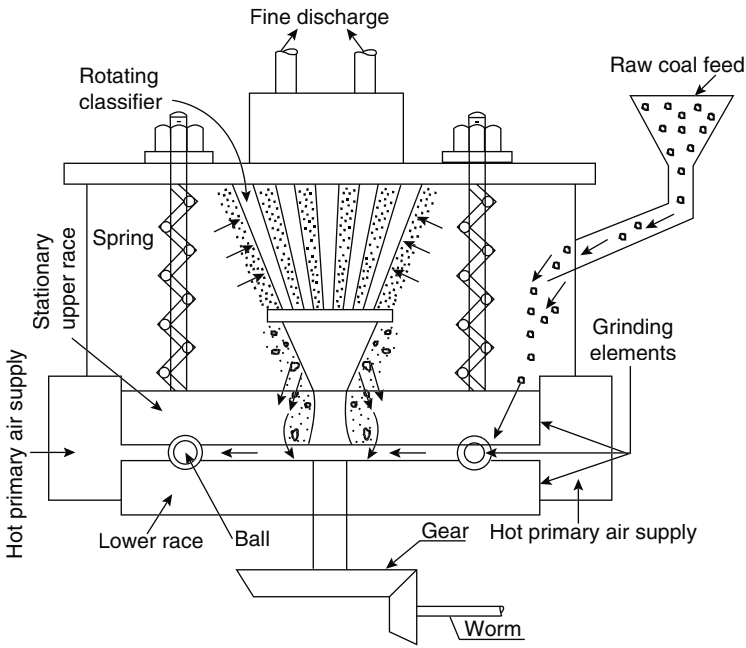


Fig. 3.22 *Ball and Race Mill*

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The mill is provided with a means of separating heavy impurities from the coal and thus reducing wear and possible damage to the grinding element. These heavy particles resist the upward thrust caused by the primary air stream and collect in a compartment in the base of mill, and then they are removed periodically. The automatic combustion control regulates the flow of primary air through the pulverizer and feeder and maintains the coal supply. When more coal is required, the primary air flow is increased automatically and its higher velocity in the mill carries additional coal in the furnace. This action reduces the amount of coal in the pulverizer and decreases the pressure drop, thus causing the feeder controller to supply more coal.

The fan used with this mill handles only air, therefore the blade erosion by coal particles is eliminated. As the casing of the pulverizer is under pressure, the leakage of fine coal through the mill casing causes the pulverized fuel to be blown out into the boiler room. This mill can handle coals containing as much as 20 per cent moisture. Mill, feeder and fan need nearly 15 kW hour energy per ton of coal pulverized.

Advantages and disadvantages

These pulverizers have greater wear compared to other pulverizers. However, the following advantages have outweighed the wear problem and these pulverizers have found general acceptance.

Advantages

(i) lower space is occupied

(ii) lower power consumptions in kW h/ton of coal pulverized

(iii) lower weight and lower capital cost

3. *Impact or hammer mill*

This is also known as impact mill as pulverization takes place by impact. The coal in pulverizer remains in suspension during the entire pulverizing process. All the grinding elements and the primary air fan are mounted on a single shaft as shown in Figure 3.23. The primary air fan induces flow of air through the pulverizer carrying the coal to the primary stage of grinding. In the primary stage of grinding, the coal is reduced to fine granular state by impact with a series of hammers and then into the final stage where pulverization is completed by attrition. The final stage of grinding consists of pegs carried on a rotating disk and travelling between stationary pegs. The finely pulverized coal in the final stage of grinding is carried with the air to the burner through the rotating scoop-shaped rejecter arms, which throw the large particles back into the grinding section.

The output of the pulverizer is controlled by varying the coal feed and the flow of primary air by either hand or automatic control.

Advantages

(i) The pulverizer is directly connected to the motor drive and therefore operates at high speed.

(ii) The power required to drive the pulverizer is nearly proportional to the coal pulverized over a wide range of rating.

(iii) This mill requires minimum floor area due to high speed of the pulverizer and the fan being integral with the pulverizer.

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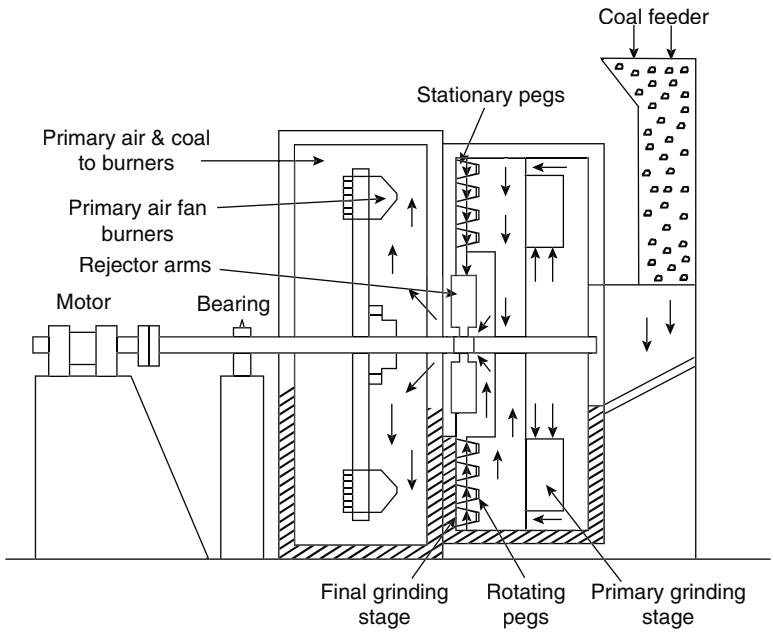


Fig. 3.23 *Impact Mill*

4. *Bowl mill*

The bowl mill shown in Figure 3.24 consists of stationary rollers and a power-driven bowl in which pulverization takes place as the coal passes between the sides of the bowl and the rollers. A primary air-induced draught fan draws a stream of heated air through the mill, carrying the pulverized coal into a stationary classifier located at the top of the pulverizer. The classifier returns the coarse particles of coal to the bowl for further grinding through the centre cone of the classifier. The coal pulverized to the desired fineness is carried to the burner through the fan. The impurities in coal containing heavy particles are thrown over the side by centrifugal force as these enter the rotating bowl. The heavy particles thrown out fall into the space below the bowl. These are discharged from the mill through a spout. The automatic control charges the supply of coal to the bowl to the mill by adjusting the feeder speed and the flow of primary air by regulating the damper in the line from the pulverizer to the fan.

Some features of this pulverizer are:

(i) Classifier may be adjusted to change the coal fineness while the pulverizer is operating.

(ii) Leakage of coal from the mill casing is completely eliminated as the mill operates under negative pressure.

Selection of the bowl mill is based on the following of parameters:

(i) Total moisture content, used to determine the mill drying capacity.

(ii) Sulphur and mineral content, used to determine pyrite removal sizing requirements.

(iii) Grindability, used to select the mill size.

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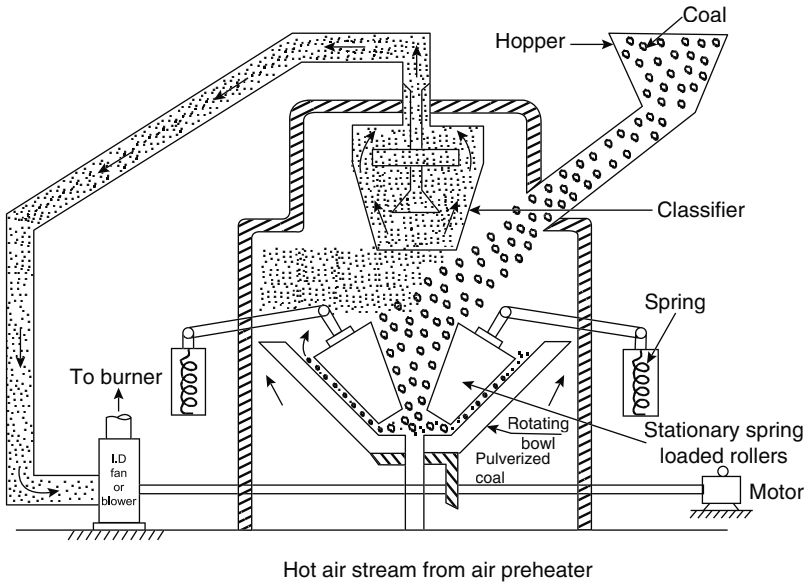


Fig. 3.24 *Bowl Mill*

(iv) Volatile matter, used to determine the required coal fineness.

(v) Higher heating value of the coal, which affects the ability of a pulverizer to meet the boiler requirement for maximum continuous rating.

A typical modern pulverizer can grind most of the fuel to at least 98–99 per cent passing 50 mesh and 60–70 per cent passing 200 mesh. The capacity of the pulverizer is also dependent on the raw coal feed size, grindability, moisture content and fineness required of the final coal.

3.8.1 Causes for Mill Fires

Mill fires are most often caused due to the following reasons:

(i) Pyrite building from pyrite plow failure

(ii) Hot start up with loaded mill

(iii) Accumulation of coal dust in mill devices

(iv) Localized hot spots from poor primary air flow

The causes of fires are also dependent on coal properties (volatile matter and moisture) that exceed pulverizing rating. Wet coal always blocks feeders, reduces system input and produces very lean fuel–air mixture in the mill. Better coal-handling method and effective trash separation could reduce feeder problems and the likelihood of mill fire. Closing of the multiport outlet valve can prevent the spread of fire from mill to burner lines. Three major types of quenching methods used are water, steam and CO_2 injection. Use of water has proved inadequate in some deep-seated mill fires. Steam injection is effective but explosive CO–H_2 mixture may result from steam coke reaction.

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3.9 FUEL-BURNING EQUIPMENTS

Different types of fuel burners are used to burn solid, liquid and gaseous fuels. Solid fuel mainly used is pulverized coal. Liquid fuels are burnt in oil burners, whereas gaseous fuels are burnt in gas burners.

3.9.1 Coal Burners

The function of the coal burners is to burn pulverized coal in the presence of primary and secondary air in a boiler furnace. Coal burners are classified according to their design and arrangement in the furnace. The following types of coal burners are used in practice:

1. *Stream-line burner*

It is also known as *long-flame* or *U-flame* burner. It is designed to fire downwards in the furnace without any turbulence. Thin flat streams of air and fuel mixture are discharged in the furnace, which travel a considerably long distance producing a long flame and ensuring complete combustion. Heated secondary air, which is fed at right angles to the flame, ensures proper mixing of air and fuel. This type of burner is suitable for burning fuels with low volatile matter as fuel flows at low velocity (15–45 m/s) and hence takes more time to travel in the furnace (Figure 3.25). A U-flame burner is shown in Figure 3.25.

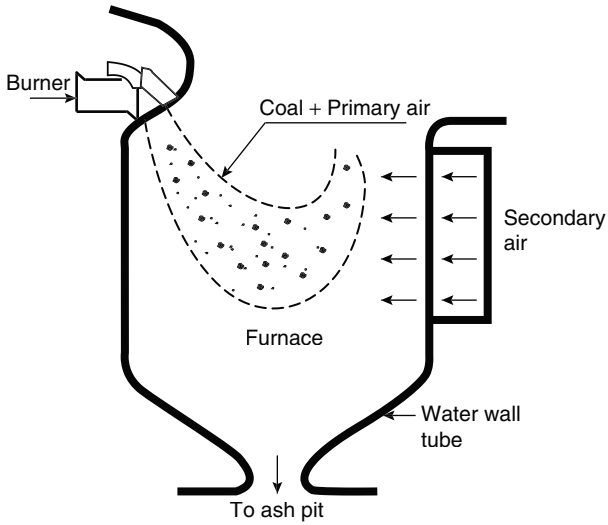


Fig. 3.25 *U-Flame Burner*

2. *Turbulent burner*

Schematic of a tangential burner is shown in Figure 3.26. It is made up of circular discharge orifices or with straight narrow outlets. It is placed in the wall of water-cooled furnaces. The fuel and air mixture pass through the burner such that they mix properly at the furnace entrance. Fuel burns in the furnace producing a short turbulent flame. These burners are very useful to burn high volatile matter fuels.

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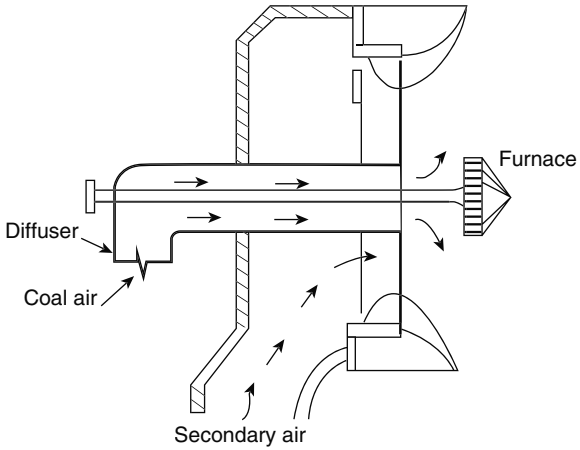


Fig. 3.26 *Turbulent Burner*

3. *Tangential burner*

In tangential burners, fuel and air streams are ejected in a horizontal direction tangent to an imaginary circle at the centre of the furnace. The burners are placed at the corners of the furnace to ensure intense turbulence and thorough mixing of fuel and air. All the fuel and air nozzles can be fitted at an angle of 20–25° above and below the horizontal plane. Schematic of a tangential burner is shown in Figure 3.27.

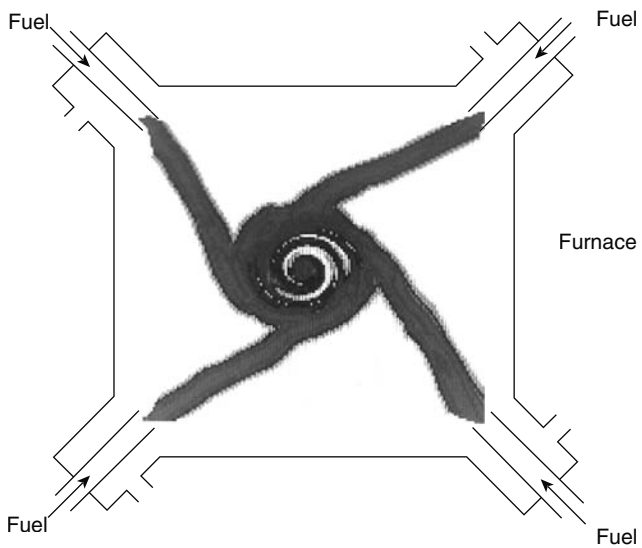


Fig. 3.27 *Tangential Burner*

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4. *Cyclone burner*

Cyclone burners are used to burn crushed coal instead of pulverized coal. This is to eliminate fly ash-handling problems. It consists of a cyclone furnace as shown in Figure 3.28. Crushed coal of about 8 mm size is fed from the feeder to the centre of the cyclone along with primary air. This creates a vortex flow in the furnace. The secondary air is supplied separately to increase the vortex flow. The ash formed due to rapid burning of coal drains down the inner wall of the furnace as molten slag. Hot flue gases along with 10–20 per cent of fly ash enter the furnace. Due to centrifugal action most of the fly ash is thrown against the furnace wall and is drained out along with molten slag. Thus, flue gases leaving the furnace are clean.

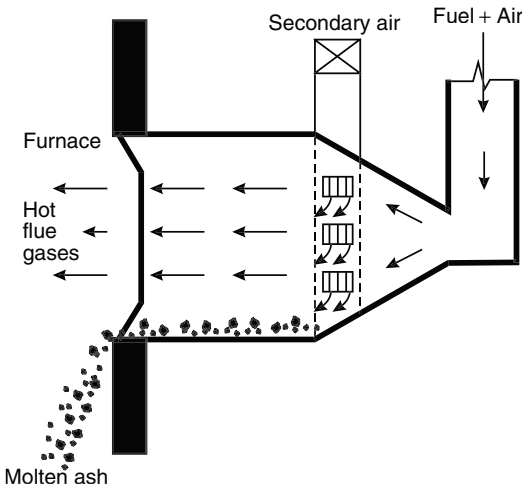


Fig. 3.28 *Cyclone Burner*

A schematic diagram of a cyclone furnace showing the circulation of primary, secondary and tertiary air is shown in Figure 3.29 below.

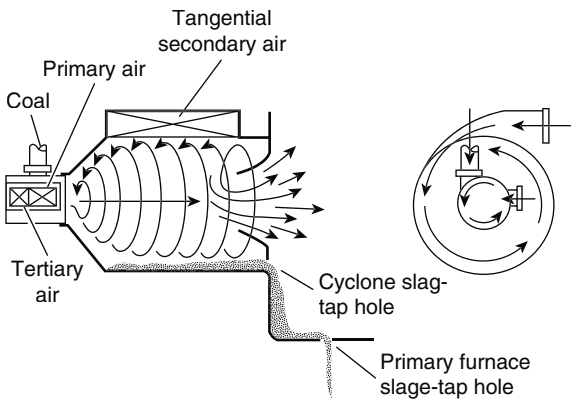


Fig. 3.29 *Schematic Diagram of a Cyclone Furnace*

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3.9.2 Oil Burners

Oil burners can be classified into two different types, namely, (i) vapourizing type and (ii) atomizing type.

The *vapourizing burners* are used for domestic purpose especially in portable air heaters, cooking stoves, etc. The *atomizing burners* are commonly used in industrial heating.

1. *Vapourizing burner*

Petromax stove, which burns kerosene, is a classical example of a vapourizing burner. Here, an upward moving high-velocity fuel jet is allowed to strike against hot moving plate. The vapours burn with a short intense flame. The fuel is stored in a fuel tank below the fuel line and a hand-driven piston is used to increase the oil pressure by compressing the air into the reservoir.

2. *Atomizing burner*

These burners are very commonly used in oil-fired furnaces of steam generators. The oil is divided into fine particles to increase the surface area of contact with the combustion air. Due to atomization, a homogeneous mixture of oil and air is produced, which is fired into the furnace. Atomization is achieved by the following two methods:

(i) *Steam or air jet atomization*

Steam jet atomization is used in power appliances, whereas air jet atomization is used for heating work.

(ii) *Mechanical atomization*

In mechanical atomization, oil is pressurized by using a positive displacement pump or a rotating wheel that displaces the oil centrifugally. It is used in high-capacity steam-generating furnaces.

Figure 3.30 shows a high pressure steam or air atomizing burner. In this burner, oil is delivered under pressure through a central tube. Steam or air is supplied along the annular space between the oil pipe and the concentric outer tube. Near the tip of the burners the two streams impinge resulting in atomization of fuel. To improve combustion efficiency and flow conditions, the viscosity of the oil is reduced by heating it to about 120°C. The heated oil is then supplied to the burner.

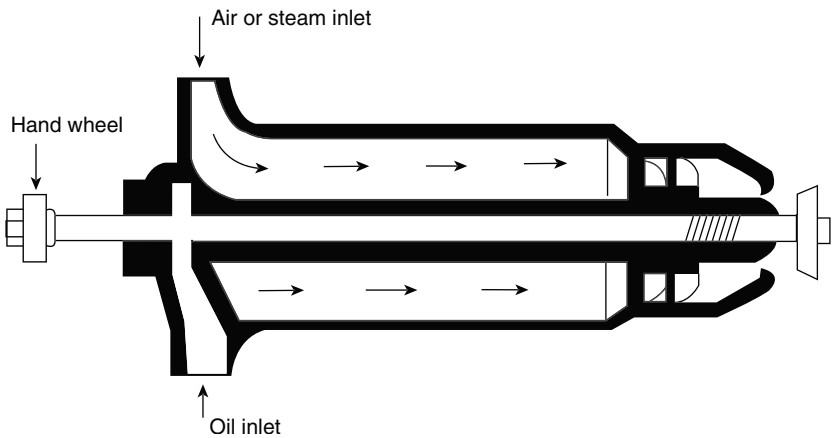


Fig. 3.30 *Atomizing Burner*

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3.9.3 Gas Burners

As the fuel is readily available in vapour form, gas burners are simpler in construction. Different types of gas burners are used in practice. Figure 3.31(a) shows a *ring burner* that can be used to control the flame length. The rate of gas flow can be controlled by using the dampers. The gas flows through an annular ring, whereas air is induced both around and within the annular ring. Figure 3.31(b) shows a typical burner used for feeding gas and air under pressure. To prevent the back flow of flame, the velocity of the gas is set at a higher rate compared to the air velocity.

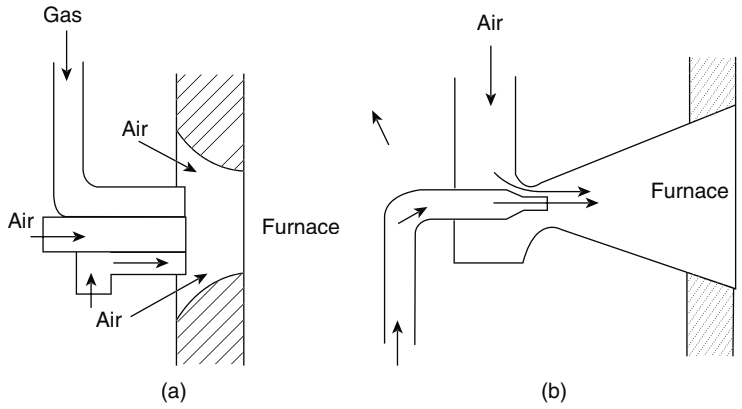


Fig. 3.31 *Gas Burners*

3.10 FLUE GAS ANALYSIS

The percentage by volume of carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2) and nitrogen (N_2) in a flue gas is determined by Orsat apparatus.

The flue gas analysis carried out by the Orsat apparatus (Figure 3.32) gives fairly accurate results upto ±0.5 per cent of CO_2 content. In its simplest form, the apparatus is arranged to determine the composition of CO_2, CO_2, CO and N_2 by difference. It contains a Eudiometer (A) graduated upto 100 cc and surrounded by water jacket. The base of the eudiometer is connected to an aspirator bottle (B) by means of a flexible rubber tube. By lowering or raising the aspirator bottle, the eudiometer can be charged or discharged. The flasks 1, 2 and 3 with duplicate flasks behind them contain KOH solution (caustic potash with KOH:H_2O, 1:2 by weight) for absorbing; alkaline solution of pyrogallic acid (5 g of pyrogallic acid in 15 cc of water is mixed with 120 g of KOH in 80 cc of water) for absorbing O_2; and cuprous chloride (CuO dissolved in 20 times its weight in concentrated HCl with copper wire immersed till it becomes colourless) for absorbing CO. Small glass tubes packed in the flasks 1, 2 and 3 accelerate the absorption of gases.

3.10.1 Procedure

The chemical reagents in the flasks 1, 2 and 3 are freshly prepared and filled. The eudiometer jackets are filled with water. The aspirator bottle is filled with salt water so that water does

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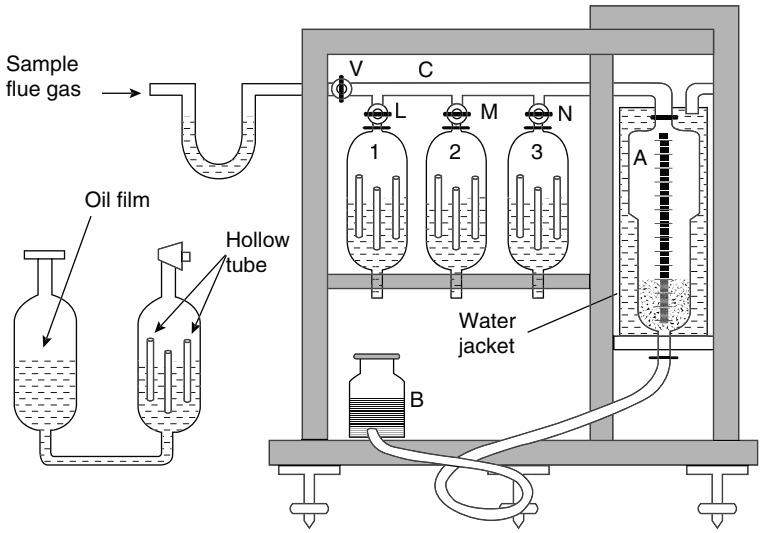


Fig. 3.32 *Orsat Apparatus*

not absorb the gases. The three way cock 'V' is opened to the atmosphere and the aspirator bottle is raised till the salt water level in the eudiometer tube reaches 100 cc mark and then the cock is closed. Now cock 'L' is opened and the aspirator bottle is lowered. The reagent in the flask 1 raises and touches the mark on the capillary tube. Cock 'L' is immediately closed. Due to this action, all gases including air are expelled from the flask. Same procedure is repeated with flasks 2 and 3. The aspirator bottle is now raised to expel all the gases removed from the flasks. It can be noted that the amount of gases present in the capillary tube 'C' is negligible.

The sample gas tube is now connected to the three-way cock and the cock is opened to the flue gases. Aspirator bottle is lowered to aspirate exactly 100 cc of sample gas (till the water level touches 0 cc mark). Cock 'V' is closed. To ensure no leakage, the aspirator bottle is raised again. If there is leakage, water level in the eudiometer goes on rising. The sample of gas is aspirated through a U-tube containing wool to ensure only filtered gas. The water jacket of the eudiometer ensures that all the moisture condenses and a dry sample of 100 cc gas is obtained. The aspirator bottle is now brought near the eudiometer so that the water levels in the eudiometer and the bottle remain the same. This ensures that the gas sample collected is at atmospheric pressure.

The cock 'L' is opened and the aspirator bottle is raised to pass the sample of gas into flask 1, which contains KOH solution. During this period, the reagent passes to the duplicate flask to accommodate the sample gas and the wetted surface of the glass tube absorbs CO_2 gas. The aspirator bottle is now lowered so that the reagent rises, washing the glass tubes. The aspirator bottle is raised and lowered a few times so that all CO_2 is absorbed. Cock 'L' is closed. The aspirator bottle is brought near the eudiometer again such that the level of water in the bottle and the eudiometer remains same. The reading on the eudiometer tube is noted, which gives percentage by volume of CO_2.

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The same procedure is repeated by opening cocks 'M' and 'N' one by one. The corresponding readings on the eudiometer tube give percentage by volume of oxygen (O_2) and carbon monoxide (CO), respectively.

volume per cent of nitrogen (N_2) is determined as follows:

volume per cent of N_2 = 100 – volume per cent of ($CO_2 + O_2 + CO$)

3.11 ASH HANDLING SYSTEM

Coal that is available in nature has some percentage of ash. When coal is burnt, about 10–20 percent of it is converted into ash. Considering the large coal-burning capacity of modern power plant, the amount of ash that is produced annually accounts to thousands of tonnes per annum. Thus, it is necessary to have modern ash-handling systems.

An ash-handling system should perform the following operations in sequence as shown in Figure 3.33.

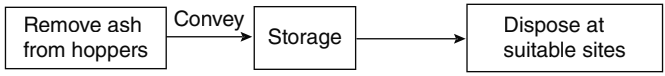


Fig. 3.33 *An Ash-Handling System*

Figure 3.34 shows the general layout of an ash and dust handling system used in modern power plants.

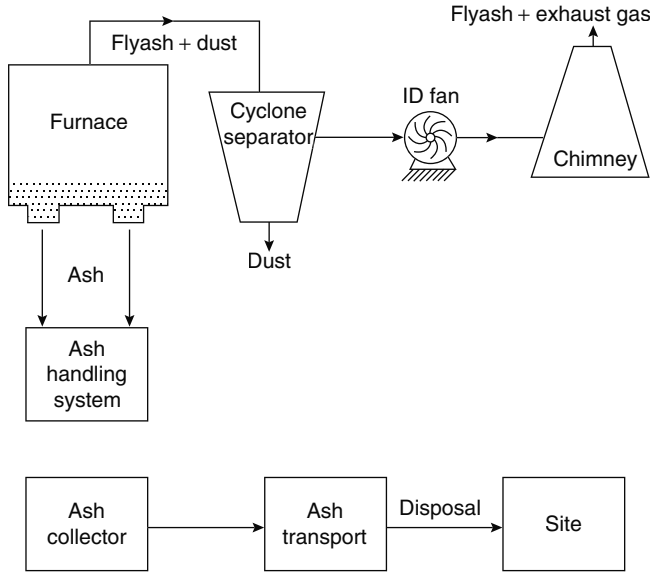


Fig. 3.34 *Ash and Dust Handling System*

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Modern ash-handling system may be classified into the following four groups:

1. *Mechanical handling system*

Figure 3.35 shows a mechanical dust-handling system. This system is used to handle limited amount of ash especially in small plants. Hot ash from the boiler furnace is made to fall through a water seal over the belt conveyor. The cooled ash is carried away by the belt to the dumping site or to an overhead bunker. The ash is disposed off from the bunker by means of trucks. The system can deliver ash up to 3 t/h with a speed of 30 cm/min for a period ranging from 5 to 10 years. Power consumption using this system is low.

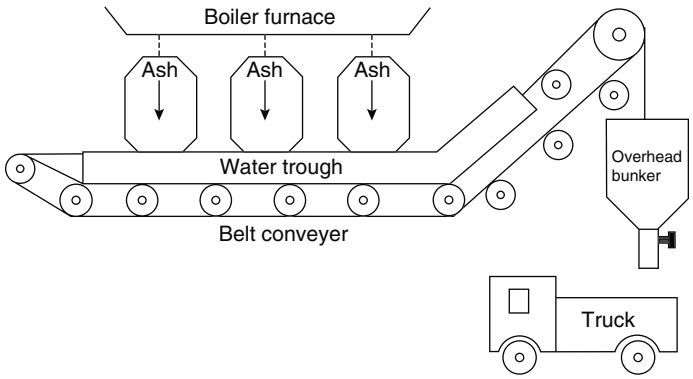


Fig. 3.35 *Mechanical Ash-Handling System*

2. *Pneumatic system*

Figure 3.36 shows a typical pneumatic ash-handling system. In this system, air is used as a medium to carry ash over long distances at the rate of 5–30 t/h. The system consists of ash crushers, separators, hopper and ash filter.

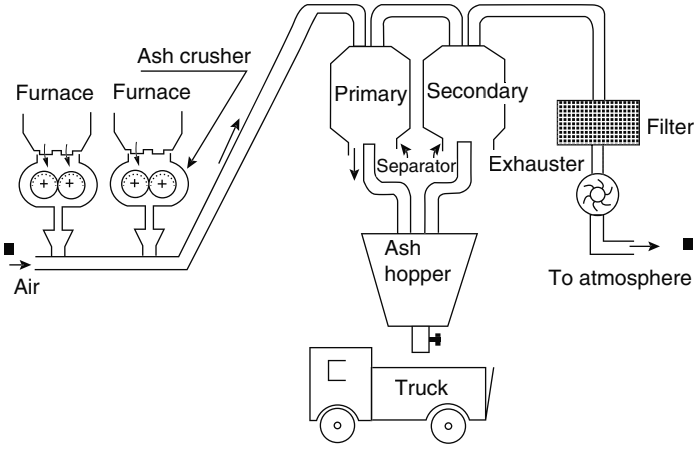


Fig. 3.36 *A Typical Pneumatic Ash-Handling System*

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The ash collected in the ash hopper is crushed by using ash crushers. This ash is carried to the primary and secondary cyclone separators using air. Heavier ash settles down due to centrifugal action and is collected in the ash hopper. Air leaving the secondary separator is further cleaned by passing through filter. Clean air is exhausted to the atmosphere using exhaust fan. This system is prone to rapid wear and tear resulting in failure of conveying pipes; it is noisy in operation.

3. *Hydraulic system*

In this system, water is used as the medium to carry ash at high velocity through channels. Depending on water pressure or velocity, the system is subdivided as *low-pressure system* and *high-pressure system*.

(i) *Low-pressure system*

In this system, ash falling from the boiler furnace is carried away by water moving at a low velocity of 3–5 m/s into sumps. In the sump, ash and water are separated by passing over a screen. Water is pumped back to the trough and used again. The ash collected is removed using carriages. This system is continuous and can handle about 40–50 t/h of ash through a distance about 500 m. Figure 3.37 shows a low-pressure system.

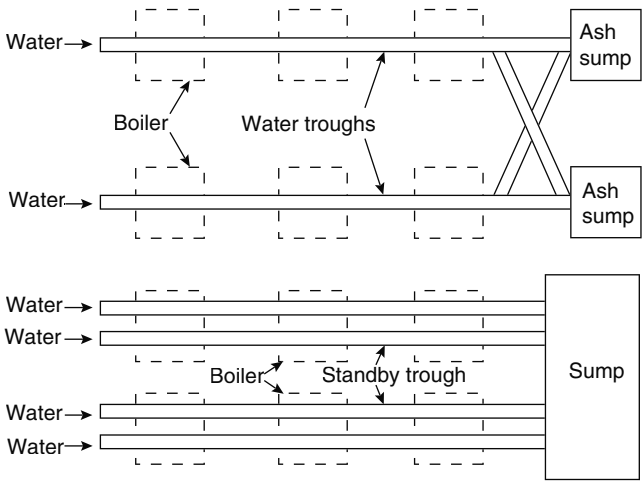


Fig. 3.37 *Low-Pressure System*

(ii) *High-pressure system*

In high-pressure system, the hoppers below the boilers are fitted with water nozzles at the top and on the sides. Ash is quenched by top nozzle spray and is carried away by the side nozzle spray. The quenched ash flows along with water at high velocity and gets collected in the sump. Water is separated by ash by passing it over screen and is re-circulated. Figure 3.38 shows a high-pressure system.

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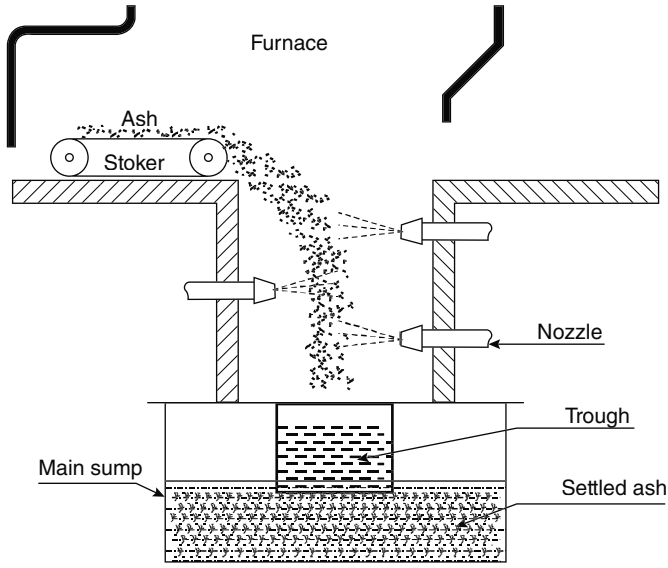


Fig. 3.38 *High-Pressure System*

Advantages of hydraulic system

(i) It is clean, dustless and totally enclosed.

(ii) Its ash-carrying capacity is considerably large, therefore it is more suitable for large thermal power plants.

(iii) It can discharge ash to a considerable distance (1000 m) from the power plant.

(iv) It can also be used to handle a stream of molten ash.

(v) The unhealthy aspects of ordinary ash basement work are eliminated.

(vi) The whole system is clean and healthy.

(vii) The important feature of the system is the absence of working parts in contact with the ash.

Low-velocity hydraulic system is gaining more popularity in modern thermal power stations as it is simple, able to handle large ash quantities, and is clean and comparatively economical.

4. *Steam jet system*

In this system, high-pressure jet of steam is used as a medium to carry away the ash from boiler hopper. This system can remove ash through a horizontal distance of 200 m and a vertical distance of 30 m. This system is useful particularly when there is limited space so that no other handling plants can be installed. As pipes carry ash that is highly abrasive, it is necessary to use nickel alloy as lining to prevent wear and tear. This system does not require any other auxiliary drives except steam generated by the boiler itself. For each tonne of ash removal, about 110 kg of steam is required.

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3.11.1 Advantages and Disadvantages of Wet and Dry Ash-Handling Systems

Advantages of wet system

(i) Transportation of ash by pipelines eliminates noise, dust and traffic problems.
(ii) Use of manned equipment is eliminated.
(iii) The system is unaffected by transportation strikes.

Disadvantages of wet system

(i) Large quantities of leachate under a positive pressure head in pond pose a constant threat to ground water quality. This is prevented by surface preparation and artificial lining, which are very costly.
(ii) The transport water is normally recycled. This requires additional pipelines, pumping equipment, treatment facilities and substantial operating and maintenance costs.
(iii) A larger area is required. Area of wet system may be twice that of the dry system.
(iv) Water requirements are very large.
(v) Scaling and cementation within pipeline, particularly when the slurry contains calcium, magnesium and sulphate ashes, may render this system unsuitable in certain cases.
(vi) It is not flexible to relocate the other discharge site.

Advantages of dry system

(i) Leachate quantities are significantly reduced. Liners to disposal area can be eliminated by fixation of ash. Ash piles can be designed to provide drainage at different levels.
(ii) Water and power requirements are considerably less.
(iii) Compacted ash is a structural material that can be sold.
(iv) Required storage volume and area are reduced considerably. The density of compacted ash is 1400 kg/m^3 against 900 kg/m^3 of loose ash.
(v) The ash disposal site has wider choice of land after closure.
(vi) This system offers greater flexibility in operation as ash is transported by vehicles to different sites.

Disadvantages of dry system

(i) Use of trucks makes this system totally dependent.
(ii) It presents increased visual impact along transportation route.
(iii) Wetting of ash containing calcium and magnesium forms lumps, which may stick to conveying belt.

3.12 DUST COLLECTION

The exhaust gases leaving the boiler contain fine particles of solid matter in suspension in the form of smoke, dust, soot, fly ash or cinder. The amount of these solid particles depends on the method of firing. In the case of pulverized fuel firing, about 80 per cent of total ash produced in

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the furnace escapes through the chimney in the form of fine particles (particulates varying from 1 to 80 μ). Such emissions from the power plant are highly objectionable considering their ill effects on environment and living things. Dust collection and removal poses a serious problem in power plants.

Dust collectors may be either mechanical or electrical type.

3.12.I Mechanical Dust Collectors

Mechanical dust collectors may be either dry or wet type. In the case of wet-type collectors, dust is washed away by spraying water on it.

1. *Gravitational separators*

Figure 3.39 shows different types of gravitational separators. In Figure 3.39(a), the gas velocity reduces due to increased duct area. Heavier dust particles settle at the bottom in Figure 3.39(b); the flow direction of the flow gas is suddenly changed and heavier particles settle at the bottom. In Figure 3.39(c), some baffles are placed in the direction of flow of flue gases. The dust particles strike the baffles and settle.

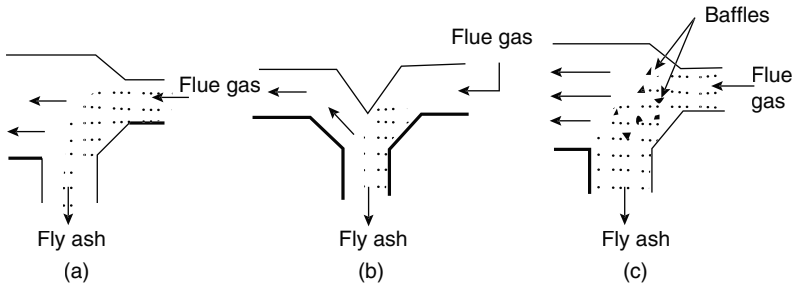


Fig. 3.39 *Gravitational Separators*

2. *Cyclone separator*

In a cyclone separator shown in Figure 3.40, high-velocity flue gas carrying dust is made to pass through a conical separator in a tangential direction. This results in a centrifugal action (whirling motion), throwing away the heavier dust particles to the sides of the conical chamber. Dust-free gas from the conical shell is passed through a secondary chamber to the chimney. Heavier dust particles accumulate on the inner surface of the shell and settle at the bottom. For efficient handling of dust, multiple cyclone dust collectors are used in both stoker firing and pulverized fuel firing installations.

3. *Baghouse filters*

To remove the dust particles from the flue gas, fabric filters are used in power plants. They are made of porous material that retain particulate matter as gas passes through the voids. A fabric filter element is made up of long hollow cylindrical tube that provides a large surface per unit of gas volumetric flow rate. A system consisting of a large number of such filter elements

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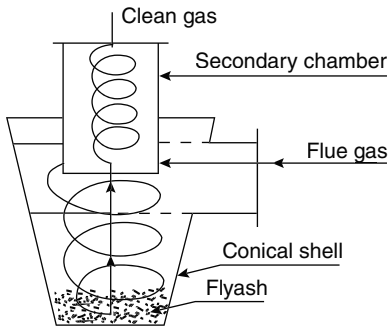


Fig. 3.40 *Cyclone Separator*

arranged in parallel rows is called a *baghouse*. The diameter of the hollow cylinder varies from 130 to 350 mm with height up to 12 m.

Dirty gases from the air heater enter the filter elements at the bottom and deposit the particulate matter on the inner surface of the tubes. Gases then pass laterally through the fabric and exit to the outlet manifold. The gases enter the chimney afterwards. The element has an open bottom and closed top and rests on a sheet above a dirty air plenum. The collection efficiency of such filters is about 99.99 per cent. Figure 3.41 shows a baghouse filter.

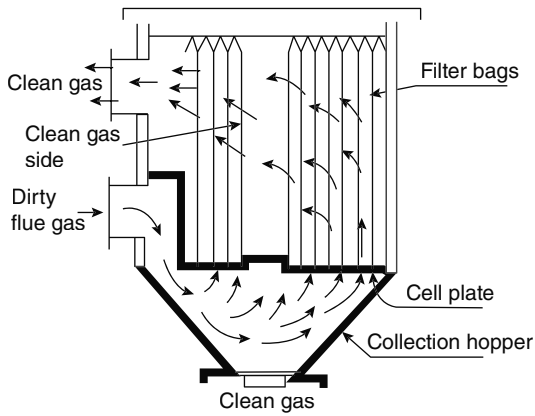


Fig. 3.41 *A Baghouse Filter*

4. *Wet scrubber*

Figure 3.42 shows principle of working of a wet-type dust collector, also known as *gas scrubber*. In this system, dirty flue gas enters the vertical chamber tangentially and follows a vortex path in the chamber and finally escapes out. During this period, water is being sprayed into the gas using a spray manifold placed at the centre of the chamber. The dust in the flue gas is washed off and drains in the form of slurry from the bottom. To minimize corrosion, the chamber and pipes

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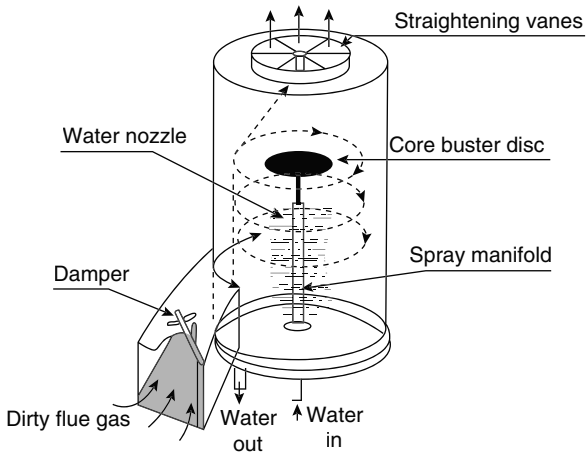


Fig. 3.42 *A Wet-Type Dust Collector*

are lined with lead and rubber, respectively. The spray nozzles are usually made of vitrified materials. The efficiency of this system is about 90 per cent, but it is susceptible to corrosion and requires a large quantity of water.

3.12.2 Electrical Dust Collector (Electrostatic Precipitators)

With stringent air quality restriction by the governments, a majority of the power plants in the world use *electrostatic precipitators* (ESPs) for dust removal. An ESP can handle large volumes of flue gases accompanied low pressure drop and high collection efficiency (99.5 per cent). It facilitates easy removal of dust particles.

Figure 3.43 shows a general arrangement of an ESP. It consists of two sets of electrodes: the *emitting or discharging electrode* and *collecting electrode*. In the case of a *tubular-type* precipitator, emitting electrodes are placed in the centre of the pipe whereas in the case of *plate-type* precipitator, emitting electrodes are placed midway between the two plates. The emitting electrodes are connected to negative polarity of high-voltage (20–100 kV) DC source. The collecting electrodes are connected to the positive polarity of the source and earthed.

When high voltage is applied, it generates a unidirectional non-uniform electric field having greater magnitude at the discharge electrodes. This results in a blue luminous glow, called a *corona* around them. This corona is an indication of negatively charged ionized gas molecules that travel from discharge electrodes to grounded collection electrodes. The dust particles thus get deposited on the collector electrodes and lose their charge. The remaining dust particles cling to the electrode surface due to electrical resistivity, and are removed by rapping the electrodes using rapping motors.

Working principle

The principal components of ESPs are two sets of electrodes as shown in Figure 3.44. The first set comprises rows of electrically grounded vertical parallel plants, called collection

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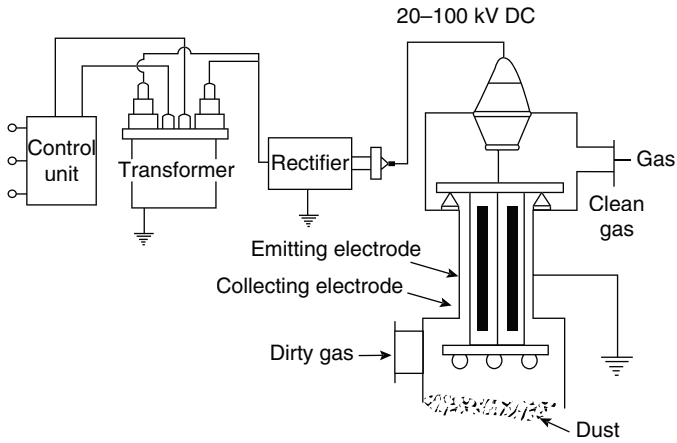


Fig. 3.43 *A General Arrangement of an ESP*

electrodes. The gas to be cleaned flows between these plates. The second set, called discharge electrodes, consist of wires. These are located centrally between each pair of parallel plates. The wires carry a unidirectional, negatively charged high-voltage (between 20 and 100 kV) current from an external source. This generates a non uniform, unidirectional electric field with greater magnitude near the discharge electrodes. When the voltage is high enough, a blue luminous glow, called corona, is produced around them. It is an indication of negatively charged gas ions. These ions travel from the wires to the grounded collection electrodes due to the presence of strong electric field between them.

Electrical forces in the corona accelerate the free electrons present in the gas, which in turn ionize the gas molecules, forming additional electrons and positive gas ions. The new electrons in turn again ionize the gas ions and this chain reaction continues.

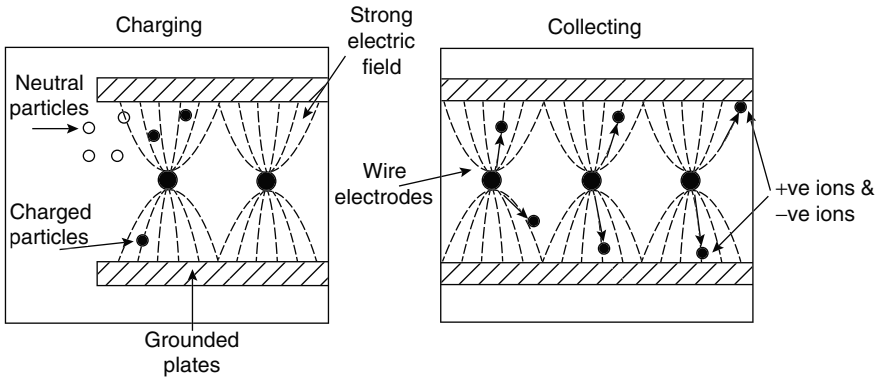


Fig. 3.44 *Working Principle of ESP, Charging and Collecting*

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The +ve ions migrate to the negatively charged wire electrodes. The electrons follow the electric field towards the grounded electrodes. Their velocity decreases as they move away from corona region.

The –ve ions migrating along electric field lines collide with the particulate matter in the gas and charge them with negative potential. The particles after acquiring sufficient charge move towards the grounded electrodes.

Migration velocity is given by

$$V_{\mathrm{m}} = \frac{2.95 \times 10^{-12}\, p(E/S)^2 d}{\mu_g}$$

where p = a function of particle dielectric constant
$= 1.50 \rightarrow 2.40$
E = applied voltage, V
S = distance between charging and collecting electrodes
d = particle diameter, m
μ_g = gas viscosity, kg/m s

When the particles collect on the grounded plates, they lose their charge to ground. The electrical resistivity causes only partial discharging. High resistivity results in holding of charges and increases the forces holding the particles to the plates low resistivity results in quick grounding of charges and hence re-entraining of particles.

When dust builds up on the plates, it deposits in a layer of increased thickness. This results in possible re-entry into the gas stream unless periodically removed. This is done by rapping the plates so as to cause shock vibrations that shake the dust into the hoppers at the bottom of the precipitator. The discharging and rapping processs are shown in Figure 3.45

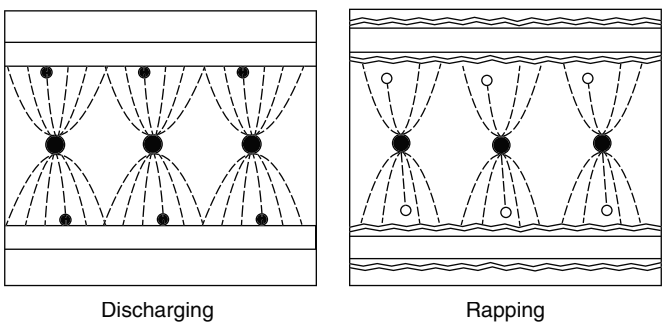


Fig. 3.45 *Working Principle of ESP, Discharging and Rapping*

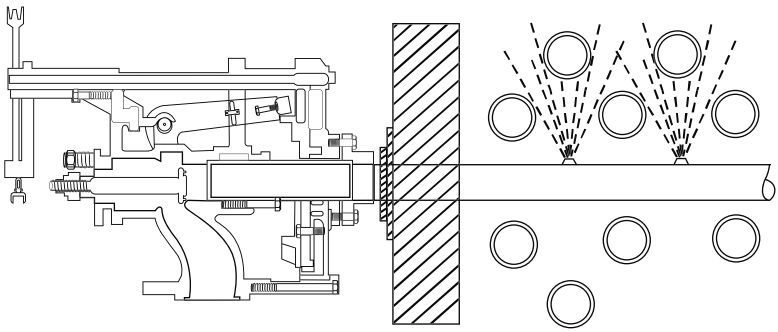
3.12.3 Soot Blowers

The products of combustion, namely, dust, soot, particulates, fly ash, carbon ash (cinder), etc., cling to the water tubes, economizer tubes, air preheater tubes and superheater tubes. This results in reduced heat transfer and boiler efficiency. Hence, these deposits should be regularly removed

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for efficient working of boilers. This method of removing soot is known as *soot blowing*. Soot blower consists of a set of tubes fitted with nozzles carrying high-velocity jet of steam or air. In the hotter parts of the boiler, the blower advances into the furnace, cleans the heat transfer surfaces and comes back. This type of soot blower is known as *retractable soot blower*.

The usual form for the smaller boilers (Figure 3.46) is a dead-end tube projecting through the setting and extending into the tube bank. A number of small lateral nozzles are provided and the external head is arranged so that high-pressure steam can be admitted to the tube and at the same time the tube rotated around its axis. Externally, the device incorporates a turning mechanism (manual or motorized), a valve and the packing gland necessary between fixed steam supply and rotating tube. Steam jets clean soot from heating surface in their vicinity, but the action is limited in extent in a close array such as a tube bank; hence, several elements may be needed to clean a boiler adequately. It is quite common to see 8–16 units in a large boiler. These would be tedious to operate manually, and hence must not all be operated simultaneously; the operating heads are electric motor powered and operated by remote control on an automatic sequential basis. On some boilers, the automatic draft system controls have to be positioned for abnormally high draft before beginning the soot blowing operation; otherwise, the release of steam into the setting may blow soot and smoke into the boiler room.



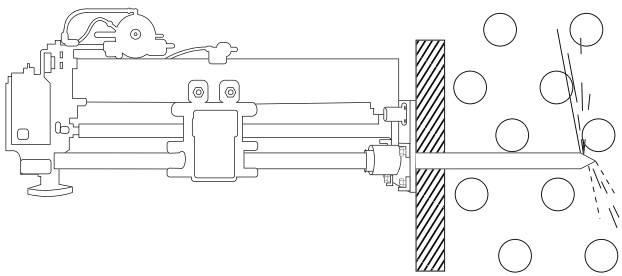


Fig. 3.46 *Soot Blower*

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When not in use the soot blower tube of the system would be dry uncooled, and subject to gas temperatures. Even with alloy tubes, the deterioration is relatively rapid on units normally experiencing high gas temperatures. Retractable elements, although more expensive initially, have the advantage of protection from high temperature when not in use. Also, using only two travelling nozzles, the jet size can be larger without excessive steam drain, and the projected blast distance greater. The eroding action of blower jets on boiler tubes, sometimes seen with the fixed rotary type, is said to be much less with the retractable type. Principal disadvantages are (i) clearances required around outside of setting and (ii) more difficult steam-packing job (travel as well as rotation). Despite the cost of compressed air, some retractable elements have been operated on air to reduce boiler water make up, and blower packing gland maintenance.

Soot blowers should be operated frequently enough to keep the tubes clean. Instead of a once a shift or twice a shift rule of thumb routine, the operating engineer should observe the flue gas temperature drop, estimate the quantity of steam per blow and calculate the frequency on an economic basis. Obviously, if the temperature drops 24°C after blowing, the interval might well be decreased, whereas if only 12°C were observed, it could represent a net loss because the value of the steam used was more than that of the increment of flue gas heat.

3.13 QUESTIONS

3.13.1 Objective Questions

1. The equipment used in Orsat apparatus is
   (a) Eudiometer (b) Planimeter
   (c) Manometer (d) None
2. The flue gas analysis is done by using
   (a) Orsat apparatus (b) Visometer
   (c) Bomb calorimeter (d) Junker's gas calorimeter
3. In Orsat Gas Analysis, KOH solution is used for absorbing
   (a) O_2 gas (b) CO_2 gas
   (c) CO gas (d) H_2 gas
4. In Orsat Gas Analysis, alkaline solution is used for absorbing
   (a) O_2 gas (b) CO_2 gas
   (c) CO gas (d) H_2 gas
5. In Orsat Gas Analysis, cuprous chloride solution is used for absorbing
   (a) O_2 gas (b) CO_2 gas
   (c) CO gas (d) H_2 gas

Answers

1. a 2. a 3. b 4. a 5. c

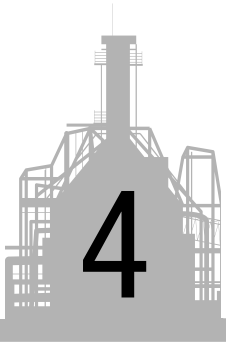
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3.13.2 Review Questions

1. Write a short note on the following.
   (a) Coal burners (b) Oil burners (c) Gas burners
2. What are the merits and demerits of pulverized coal?
3. Explain with a neat sketch the working principle of double classifier ball mill.
4. What are the principal requirements of a good ash-handling plant?
5. Differential between forced drought and induced drought system.
6. What are the benefits of a air preheater?
7. What is meant by 'overfeed' and 'underfeed' principles of firing coal?
8. List the advantages and disadvantages of using pulverized coal.
9. Explain the unit system and bin system of pulverized coal firing.
10. What are the requirements of good coal-handling plant?
11. Explain various ash-handling systems.
12. Give the general layout of ash-handling and dust collection system.
13. Draw a line diagram of inplant coal handling and indicate the names of equipments used at different stages.
14. What is meant by outplant handling of coal? What are its advantages and disadvantages?
15. What is the necessity of coal storage? Discuss the different methods used for coal storage of plant.
16. What are the different types of coal conveyors. Indicate the use of each and justify that its use is essential at that stage.
17. What are the different methods used for coal handling at plant site?
18. What are the advantages of burning the fuels in pulverized term?
19. Name the different types of coal pulverizing mill and discuss the outstanding features of each, which decides its benefit over the others.
20. Draw a neat sketch of cyclone burner and describe its working and what are its outstanding features compared to other burners. Why such burners are useful for Indian coals?
21. What are the different methods used for supplying pulverized fuel to the combustion chambers. Discuss its merits and demerits.
22. What are different ash-handling systems? Discuss their merits and demerits.
23. Draw a line diagram of hydraulic ash-handling system used for modern power plant. Discuss their merits and demerits.

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Steam Power Plant



4

Contents

4.1 Introduction
4.2 General layout of modern thermal power plant
4.3 Rankine and modified Rankine cycles
4.4 Working of different circuits
4.5 Selection of site for steam power plants
4.6 Questions

4.1 INTRODUCTION

Generation of power in a country mainly depends on the utilization of natural resources available in addition to geographical location. Majority of power generated in the world comes from hydro-electric plant, thermal power plant and nuclear power plant. Although power available by a hydro-electric plant is seasonal and depends on the availability of natural sites and the hydrological cycle, nuclear power has raised its own concern about the safety, especially in the backdrop of natural disaster. Due to this installing a thermal power plant has become more attractive. A steam power plant can be commissioned if water and fuel resources are available nearby.

The available data from CEA, as on 31 March 2011 says, at present 54.09 per cent or 9.4 GW of total electricity production in India, are from coal-based thermal power plants. In a thermal power plant, coal is used as the chief source of fuel, and burnt in the boiler furnace, by raising the steam in the steam generators. In the process, chemical energy of coal is converted into heat energy. The high pressure steam thus generated is expanded in the turbine. The turbine is coupled to the generators, converting heat energy into electricity.

Figure 4.1 shows the vital input and out parameters of a typical 1000 MW power plant. The initial investment of such a plant is approximately ₹ 5000 crores.

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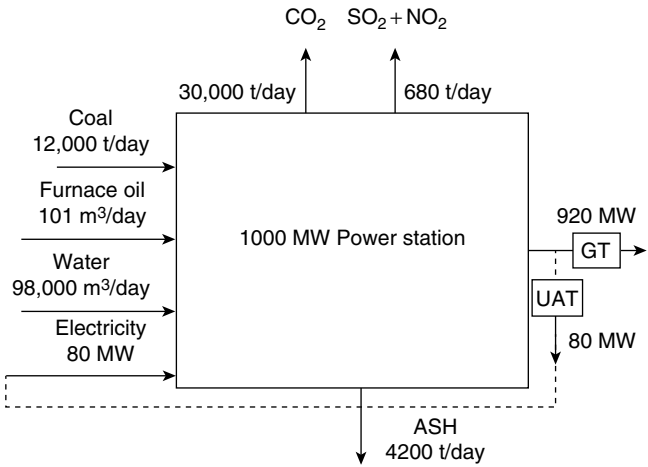


Fig. 4.1 *Vital Input and Output Parameters of a Typical 1000-MW Power Plant*

4.2 GENERAL LAYOUT OF MODERN THERMAL POWER PLANT

In a typical thermal power plant, coal is transported from coal mines to the power plant mainly by rail, through sea route or by road, if the coal is available at a nearby location. Coal is stored in a coal yard and is further treated in the coal-handling system. As the raw coal is not uniform in size, it is crushed initially in coal crushers to a size of 10–20 mm. The crushed coal is then either stored in dead storage (generally, 45–90 days coal supply meant to meet emergency) or taken to the live storage (8 hours/1 shift coal supply) in the raw coal bunker in the boiler house. The stored coal from the raw coal bunker is then taken to the pulverizing mill using a raw coal feeder. The pulverizer mill comes in as a single unit with the boiler, and the coal is crushed here to powdered form (60–80 μm). The pulverized coal is supplied to the boiler furnace in pipes by high-pressure hot air. The pulverized coal air mixture is burnt in the boiler furnace. The hot mixture of gases produced in the boiler furnace is utilized to convert water into steam. The saturated steam from the steam generator is taken through a series of super heater tubes. The superheated steam from the final super heater enters the high-pressure steam turbine (HPT), where high-pressure steam expands in the nozzle, generating work. After expansion in the HPT as the steam becomes wet, it is reheated in a reheater raising its temperature again. The reheated steam is now passed through the intermediate pressure turbine for extracting the work further and finally to the low-pressure turbine (LPT). The expanded steam from the LPT is condensed in a steam condenser by circulating cooling water. The condensate formed is collected in the hot well and is reused as feed to the boiler. The rotational energy imparted to the turbine by high-pressure steam is converted to electrical energy in the generator.

Figure 4.2 shows the general layout of a modern steam power plant

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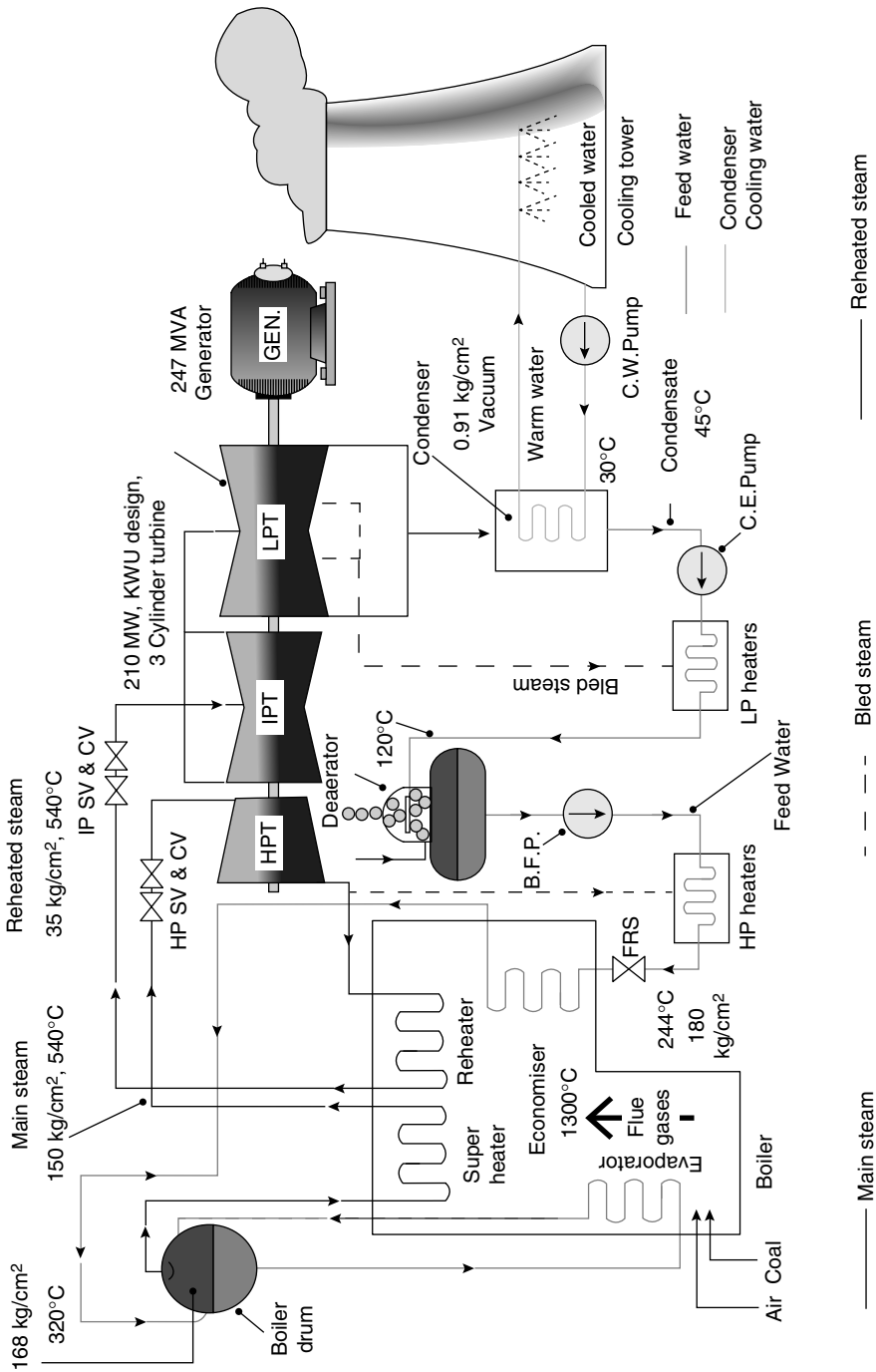


Fig. 4.2 *General Layout of a Modern Steam Power Plant*

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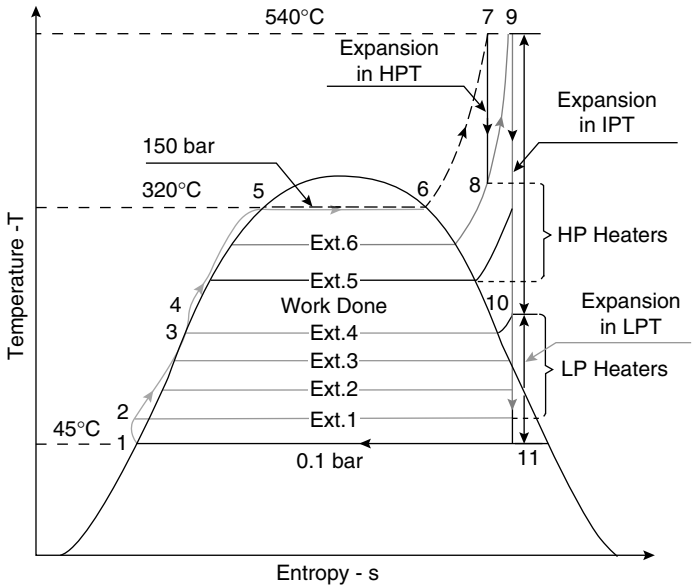


Fig. 4.3 *Temperature–Entropy Diagram*

Expansion of steam in different stages viz. HP, JP and LP and corresponding state points are shown on the T-β diagram Figure 4.3.

The major components of a thermal power station are as follows:

1. *Fuel-handling system*

It is absolutely essential to have an efficient fuel-handling system in power plants. As majority of the power plants operate using coal as a fuel, it is necessary to study about coal-handling system. Coal can be handled either *manually* or *mechanically*. But, citing the capacity of modern power plants, it is almost impossible to handle coal manually. That means *mechanical handling of coal is inevitable*.

2. *Storage of coal*

To meet the continuous demand of coal and to face any emergencies due to short supply of coal, it is always desirable to have a coal storage yard near the plant. The capacity of such a yard may be about 10 per cent of the annual consumption to a maximum of 25 per cent of the annual consumption.

3. *Inplant handling system*

This system feeds coal from live storage to the furnace using similar equipments as explained above. Coal delivered to the plant site is weighed on a wagon balance and then delivered to underground hoppers or bunkers either manually or mechanically. The coal is now lifted to the transfer tower and then to the crusher through magnetic separators and screens. Coal is crushed to the required size (20–30 mm for stoker firing and 10–20 mm for pulverized fuel firing). In the case of pulverized firing, coal is further crushed into fine powder in mill and then carried to the transfer tower from where it is conveyed to boiler bunker. From boiler bunker, coal now flows by gravity to boiler hopper and ultimately to the boiler furnace. Depending on the load, fuel flow is regulated by using gates and valves in the system.

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4. *Ash-handling system*

Coal that is available in nature has some percentage of ash. When coal is burnt about 10–20 per cent of it forms ash. Considering the large coal-burning capacity of modern power plant, the amount of ash that is produced annually accounts to thousands of tonnes per annum. Thus, it is a must to have modern ash-handling systems.

5. *Dust collection*

The exhaust gases leaving the boiler contain fine particles of solid matter in suspension in the form of smoke, dust, soot, fly ash or cinder. The amount of these solid particles depends on the method of firing. In the case of pulverized fuel firing, about 80 per cent of total ash produced in the furnace escapes through the chimney in the form of *fines* (particulates varying from 1 to 80 μm). Such emissions from the power plant are highly objectionable considering their ill effects on environment, and living things. Dust collection and removal poses a serious problem in power plants.

6. *Steam turbines*

One of the principal prime movers used for electric power generation is steam turbine. The steam turbine obtains its motive power from change of momentum of a jet of steam flowing over a curved vane. The steam jet while moving over the curved surface of the blade exerts a pressure on the blade owing to its centrifugal force. The centrifugal pressure is exerted normal to the blade surface and acts along the whole length of the blade. The resultant of centrifugal pressure, plus the effect of change of velocity, is the motive force on the blade.

In steam turbine, numbers of small blades are fixed to the rim of a revolving wheel or rotor. High-velocity jets of steam are obtained by expansion through nozzles and are directed on to the blades. The effective force of these jets, acting on the blades, rotates the wheel.

7. *Boilers and mountings*

Function of boiler is to generate steam at constant pressure as per the process requirement. The steam generated may be nearly dry saturated or superheated steam. Constant pressure in the boiler is maintained by generating the steam equal to the steam flowing out of the boiler.

In modern power plants, it is very common to use a single boiler unit per turbine. This leads to simpler piping systems and relatively easier boiler and turbine control. These boilers are usually designed to operate at *critical pressure* (221.2 bar) or above or below the critical pressure. If the boilers are designed to operate above the critical pressure, then they are known as *super critical* boilers or *once-through* boilers. If the boilers are designed to operate below the critical pressure, they are known as *sub-critical* or drum boilers. The high-pressure boilers may be further classified as natural circulation, forced circulation and once-through boilers.

(i) *Economizer*

The function of the economizer is to recover a portion of heat of the exhaust gases before the flue gases enter the chimney and discharged to the atmosphere. Economizer is placed in the path of the flue gases in between the boiler exit and entry to the chimney. Feed water coming from the feed pump when passed through the economizer tubes absorbs the heat in the exhaust gases. This increases the temperature of water entering the boiler. Due to the high temperature of feed water, fuel consumption reduces. This increases the overall efficiency of the boiler.

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(ii) *Super heater*

The function of the super heater is to increase the temperature of steam above its saturation temperature. As heat contained in unit mass of superheated steam is more than dry saturated or wet steam, it is extensively used in steam power plants. Steam from the boiler drum is passed through super heater tubes. Super heater tubes are placed in the furnace along the passage of flue gases. The temperature of steam is thus raised above the saturation temperature.

(iii) *Air preheater*

The function of the air heater is to recover the heat of a portion of exhaust flue gases before the flue gases enter the chimney. It is placed along the passage of the exhaust flue gases in between the economizer and the chimney.

Air from the forced draught fan is passed over the pre-heater tubes containing flue gases. The temperature of air is increased and this high temperature air enters the furnace. Due to higher air temperature, combustion of the fuel becomes rapid and fuel consumption becomes less. This increases the overall efficiency of the boiler.

(iv) *Feed water pump*

The function of the feed pump is to pump water at high pressure to the water space of the boiler drum. Many types of feed pumps are used.

(a) Rotary Pumps
They are either driven by electric motors or small steam turbines. Water is pumped due to rotary action of the impeller.

(b) Reciprocating Pumps
They are continuously run by steam from the same boiler to which the water is fed. Water is pumped by reciprocating action.

(v) *Pressure-reducing valve*

The function of the pressure-reducing valve is to maintain constant pressure on the delivery side of the valve with the fluctuating boiler pressure. Whenever the steam demand is fluctuating, it becomes very difficult to maintain uniform pressure. In such cases, a pressure-reducing valve is connected to the steam supply line.

8. *Condenser*

Condenser is an essential component of a modern steam-condensing plant, which increases the plant efficiency. It is a closed vessel in which high-pressure steam from the steam turbine condenses to low pressure. Different types of steam condensers are used in a steam-condensing plant. When steam condenses, it forms hot water in the condenser. The heat contained in hot water coming from the condenser is rejected in either a cooling tower or a cooling pond before re-circulating it in the water-circulating system.

9. *Cooling ponds and towers*

The cooling water system is one of the most important systems of power plant and availability of cooling water decides the plant site. In a water-cooling system, cooling ponds and towers are mainly used. In large power plants, cooling towers are used in place of cooling ponds. Acooling tower is a wooden or metallic rectangular structure, with packed baffling devices. The hot water is delivered to the top of tower and falls down through the tower and is broken into small

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particles while passing over the baffling devices. Air enters the towers at the bottom and flows upward and cools the water. The hot air leaves the tower at the top. The cooled water falls into a tank below the tower from where it can be again circulated to the condenser. In modern power plants, condensate water is cooled by using a cooling tower. Cooling towers are basically classified as *wet cooling tower* and *dry cooling tower*. Further, they may be either forced draught (mechanical draft) or natural draft cooling towers depending on the mode of cooling.

4.3 RANKINE AND MODIFIED RANKINE CYCLES

Before making the analysis of Rankine cycle, it is essential to understand the working of a Carnot cycle, which puts restrictions on the operating parameter, namely temperature.

4.3.1 Carnot Cycle

It is the simplest cycle having highest thermal efficiency operating between two temperature limits. However, this cycle is practically not achievable because of the limitations in achieving isentropic conditions.

Carnot cycle consists of two constant pressure and two isentropic processes as shown on the *p–V* and *T–s* diagrams in Figure 4.4.

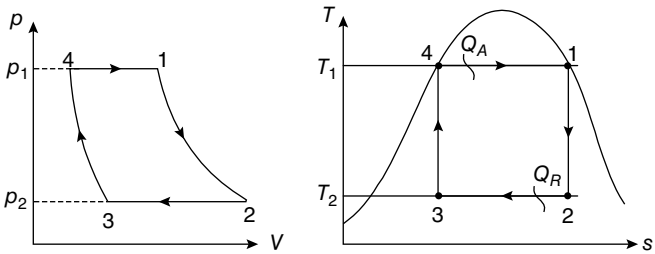


Fig. 4.4 *Carnot Cycle*

Net work done during the cycle,

$$= Q_A - Q_R$$
$$= T_1 (s_1 - s_4) - T_2 (s_2 - s_3)$$
$$= (T_1 - T_2) (s_2 - s_3) (s_1 = s_2; s_3 = s_4)$$

Thermal efficiency

$$\eta = \frac{T_1 - T_2}{T_1}$$

Some of the limitations of Carnot cycle are as follows:

(i) It is difficult to operate the cycle with superheated steam, as heat should be added to steam at constant temperature instead of constant pressure (process 4–1).

(ii) Efficiency of the cycle is dependent on maximum operating temperature, T_1. As steam has critical temperature of 374°C, maximum operating temperature is limited if the cycle operates in wet region.

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(iii) Difficulty in controlling the quality of condensate so as to maintain state 3.

(iv) Difficulty in isentropic compression of wet vapour to maintain saturation point 1.

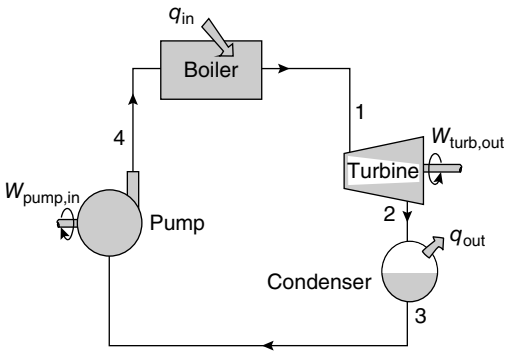


Fig. 4.5 *Rankine Cycle*

4.3.2 Rankine Cycle

Figure 4.5 shows a Rankine cycle, which forms the basic working cycle of steam power plants. In this cycle, feed water supplied by a multistage feed pump is raised into steam in a boiler. The high-pressure steam is expanded in the turbine, generating work. After expansion, the steam is condensed in a condenser and the cycle repeats. In a real cycle, due to irreversibility, losses are present and the cycle efficiency decreases.

Figure 4.6 shows a simple Rankine cycle on *p–V* and *T–s* diagrams. In this cycle, either saturated steam enters the turbine at point 1 or superheated steam enters the turbine at point 1′. Cycle 1–2–3–4–*b*–1 is a saturated Rankine cycle and cycle 1′–2′–3–4–*b*–1′ is a superheated Rankine cycle.

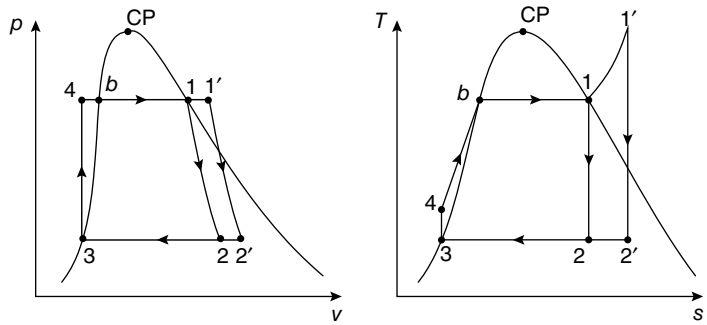


Fig. 4.6 *Rankine Cycle on p–V and –s Diagrams*

The cycle has the following reversible processes:

(i) **1–2 or 1′–2′:** Adiabatic reversible expansion through the turbine. The exhaust vapour at the end of expansion at 2 or 2′ is usually in the two-phase region.

(ii) **2–3 or 2′–3:** A two-phase mixture constant temperature and pressure process. Heat is rejected in the condenser at constant pressure.

(iii) **3–4:** Adiabatic reversible compression. The pump increases saturated liquid at condenser pressure at 3, to subcooled liquid at the steam generator pressure, 4. Line 3–4 is a vertical line as the liquid is incompressible and pump work is adiabatic reversible.

(iv) **4–1 or 4–1′:** Heat is added at constant pressure in the steam generator. It is shown by the line 4–*b*–1–1′. During the process 4–*b*, subcooled liquid at point 4 becomes

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saturated at point b. This conversion occurs in the economizer of the steam generator. During the process b–1, saturated liquid at point b is heated at constant pressure and temperature to saturated vapour at point 1. This change occurs in the evaporator or boiler. Process 1–1′ corresponds to the conversion of saturated vapour at point 1 to superheated vapour at point 1′ in the super heater.

Let h_1 = enthalpy of saturated vapour at 1
$h_{1'}$ = enthalpy of superheated vapour at 1′
h_2 = enthalpy of steam vapour at 2
$h_{2'}$ = enthalpy of steam vapour at 2
$h_3 = h_{f3}$ = enthalpy of water at point 3
$h_4 = h_{f4}$ = enthalpy of water at point 4
$h_b = h_{fb}$ = enthalpy of water at point b

(v) Heat added per unit mass at constant pressure process 4–b–1 or 4–b–1′ is given by

$$q_A = h_1 - h_{f4} \text{ kJ/kg saturated vapour cycle}$$
$$q_A = h_{1'} - h_{f4} \text{ kJ/kg superheated vapour cycle}$$

(vi) Heat rejected at constant pressure process 2–3 or 2′–3 is

$$q_R = h_2 - h_{f3} \text{ kJ/kg saturated vapour cycle}$$
$$q_R = h_{2'} - h_{f3} \text{ kJ/kg superheated vapour cycle}$$

(vii) Turbine work during the process 1–2 or 1′–2 is

$$w_T = h_1 - h_2 \text{ kJ/kg saturated vapour cycle}$$
$$w_T = h_{1'} - h_2 \text{ kJ/kg superheated vapour cycle}$$

(viii) Pump work during the process 3–4 is

$$w_p = h_{f4} - h_{f3} \text{ kJ/kg}$$
$$= v_3 (p_4 - p_3) \text{ kJ/kg}$$

1. *Net work done*

Net work for saturated vapour cycle
= Heat added – Heat rejected
$= (h_1 - h_{f4}) - (h_2 - h_{f3})$
$= (h_1 - h_2) - (h_{f4} - h_{f3})$ kJ/kg
$= (h_1 - h_2) - w_p$

Net work for superheated vapour cycle
= Heat added – Heat rejected
$= (h_{1'} - h_{f4}) - (h_{2'} - h_{f4})$
$= (h_{1'} - h_{2'}) - (h_{f4} - h_{f3})$ kJ/kg
$= (h_{1'} - h_{2'}) - w_p$

2. *Thermal efficiency*

Thermal efficiency is given by

$$= \frac{q_w}{q_A}$$

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Thermal efficiency for saturated vapour cycle

$$= \frac{(h_1 - h_2) - w_p}{(h_r - h_{f4})}$$

Thermal efficiency for superheated vapour cycle

$$= \frac{(h_{1'} - h_{2'}) - w_p}{(h_{r'} - h_{f4})}$$

4.3.2.1 *Actual and Ideal Processes for a Rankine Cycle*

As mentioned earlier, in an actual Rankine cycle, due to the irreversibilities in various components arising out of fluid friction and heat loss to the surroundings, pressure drop in the boiler, condenser and piping system is inevitable. To overcome the pressure drop, it is essential to supply feed water to the system at sufficiently higher pressure than the ideal cycle. This results in higher power consumption. Both ideal and actual cycles are shown in Figure 4.7(a) and (b).

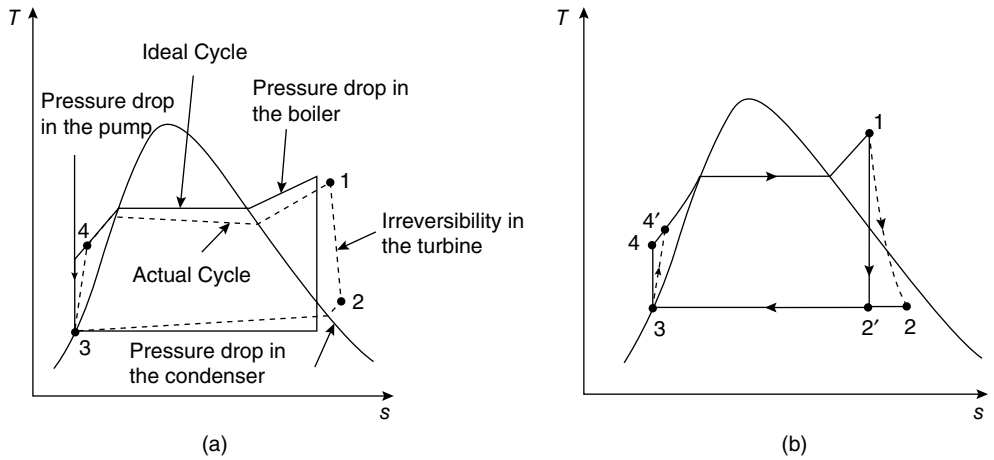


Fig. 4.7 *Actual and Ideal Processes for a Rankine Cycle*

Due to the heat loss to the surroundings, turbine work decreases resulting in lower efficiency, given by the following equations. Pump efficiency for an actual cycle is given by

$$\eta_p = \frac{q_w}{q_A} = \frac{h_{4'} - h_1}{h_4 - h_1}$$

Turbine efficiency for an actual cycle is give by

$$\eta_T = \frac{q_A}{q_w} = \frac{h_1 - h_2}{h_1 - h_{2'}}$$

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4.3.2.2 *Improving the Cycle Efficiency*

It is possible to increase the Rankine cycle efficiency by varying the pressure and temperature. Effect of pressure and superheating with the following parameters improves the cycle efficiency:

(i) Lowering the condenser pressure raises the thermal efficiency of the cycle.

(ii) Raising the boiler pressure and temperature raises the thermal efficiency of the cycle.

(iii) Raising steam temperature (superheating) raises the thermal efficiency of the cycle.

1. *Lowering the condenser pressure*

Condition of steam is a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. Lowering the condenser pressure also lowers the temperature of the steam, enabling heat rejection at a lower temperature. The effect of lowering the condenser pressure on the Rankine cycle efficiency is shown on the *T*–*s* diagram in Figure 4.8. However, decreasing the condenser pressure increases the moisture content of the steam at the final stages of the turbine leading to erosion and corrosion of blades.

Considering the efficiency equation,

$$\eta = \frac{(h_1 - h_2) - (h_{f4} - h_{f3})}{(h_1 - h_{f4})}$$

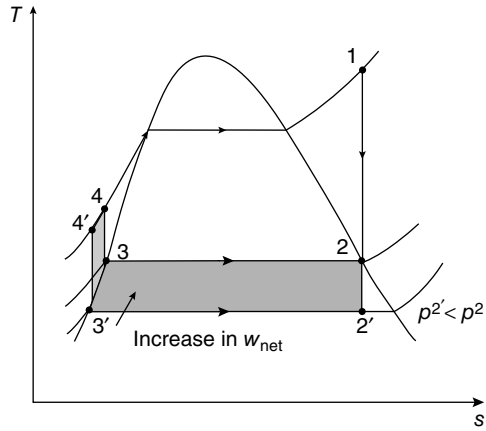


Fig. 4.8 *Lowering the Condenser Pressure*

In the above equation, the turbine work increases from $(h_1 - h_2)$ to $(h_1 - h_2')$ with corresponding decrease in pump work and hence efficiency increases. However, the lowest pressure of condenser under ideal conditions is limited to the saturation temperature of cooling water or air. In actual practice, this value is not achieved as some difference in temperature has to be maintained for heat transfer to take place.

The variation of Rankine cycle efficiency with condenser back pressure is shown in Figure 4.9 below.

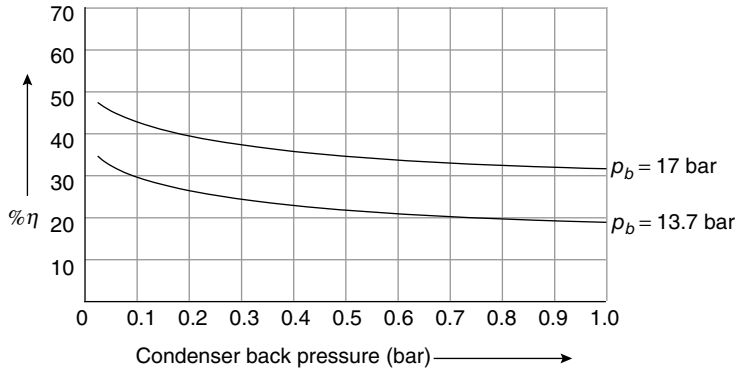


Fig. 4.9 *Rankine Cycle: Efficiency versus Condenser Pressure*

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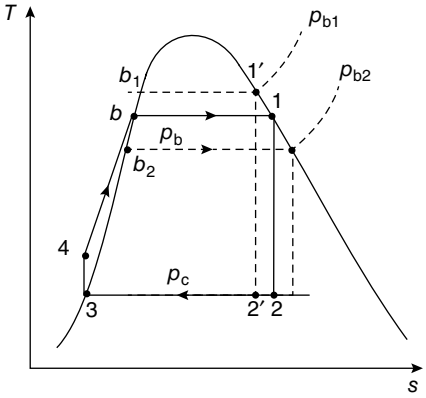


Fig. 4.10 *Increasing the Boiler Pressure*

2. *Increasing the boiler pressure*

Increasing the boiler pressure to p_{b1} at constant condenser pressure p_2 again improves the turbine work $(h_1 - h_2)$. In addition, specific steam consumption also decreases, improving the cycle efficiency as shown in Figure 4.10. However, higher pressure demands for stronger boiler joints and sealings.

The variation of Rankine cycle efficiency with boiler pressure is shown in Figure 4.11.

3. *Superheating the steam to high temperatures*

The effect of superheating on the performance of Rankine cycle is shown on a *T–s* diagram as shown in Figure 4.12.

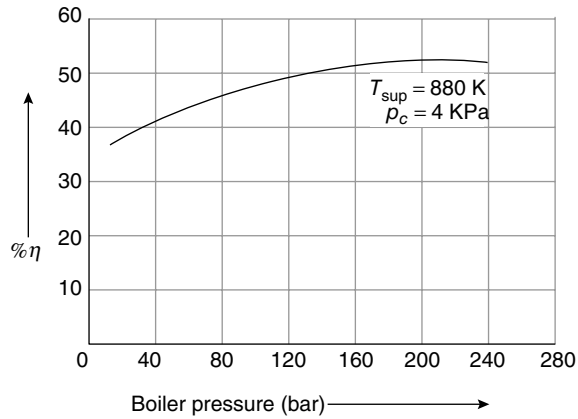


Fig. 4.11 *Rankine Cycle: Efficiency versus Boiler Pressure*

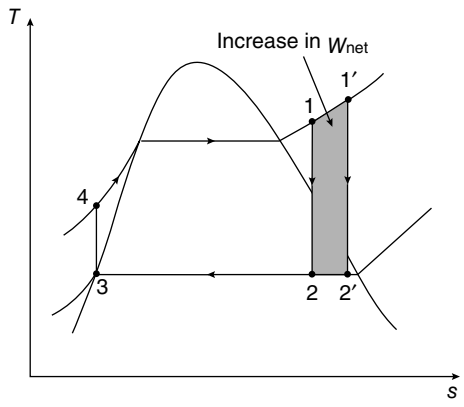


Fig. 4.12 *Superheating the Steam to High Temperatures*

It is evident from the diagram that superheating the steam to higher temperatures decreases the moisture content of the steam at the turbine exit. However, there is limit to this temperature considering the melting temperature of the boiler vessel.

In Figure 4.12, for the same boiler pressure p_1 and constant condenser pressure p_2, the steam is superheated from temperature T_1 to temperature $T_{1'}$. This, as evident from the figure, increases the turbine work $(h_1 - h_2)$ to $(h_1 - h_{2'})$. Thus, cycle efficiency increases with increased temperature. However, higher temperature demands special metals, namely alloy steels that can withstand high temperature and pressure.

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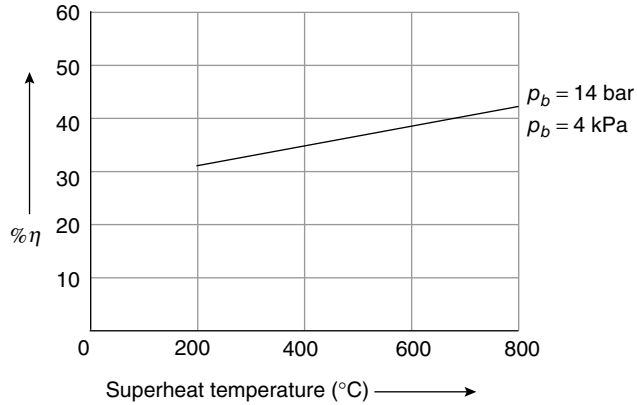


Fig. 4.13 *Efficiency versus Superheat Temperature*

The variation of Rankine cycle efficiency with superheat temperature is shown in Figure 4.13.

4.3.2.3 Comparison between Rankine and Carnot Cycle

Figure 4.14 shows Rankine and Carnot cycles on a *T–s* diagram.

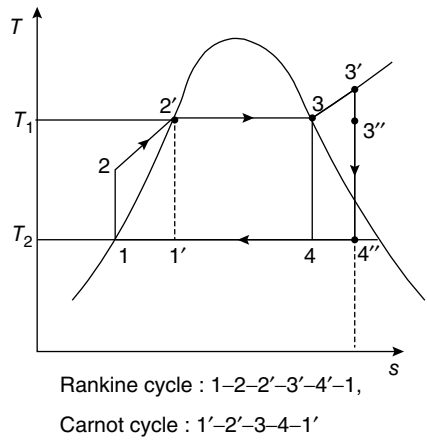


Fig. 4.14 *Rankine and Carnot Cycles on a T–s Diagram*

1. For the same maximum and minimum temperatures, Rankine cycle has lower efficiency than that of Carnot cycle. This is because the heat supplied at isothermal temperature T_1 is less compared with that during evaporation. By reducing the superheat temperature rise, Rankine cycle efficiency increases and approaches to that of Carnot cycle efficiency.
2. For the same maximum and minimum temperatures, Rankine cycle has higher specific work output than that of Carnot cycle. This is because $W_T = (h_3 - h_4) < h_{3'} - h_{4'}$ as evident from the figure. Thus, specific steam consumption of Rankine cycle is less than that of Carnot cycle. This means, for a given specific output, Rankine cycle has smaller sized plant.
3. Compression of wet vapour during process 1′–2′ is difficult and very large in case of Carnot cycle when compared to pumping feed water to the boiler during process 1–2 in the case of Rankine cycle.

Figure 4.15 shows comparison of Carnot and Rankine cycles with respect to efficiency and specific steam consumption against boiler pressure.

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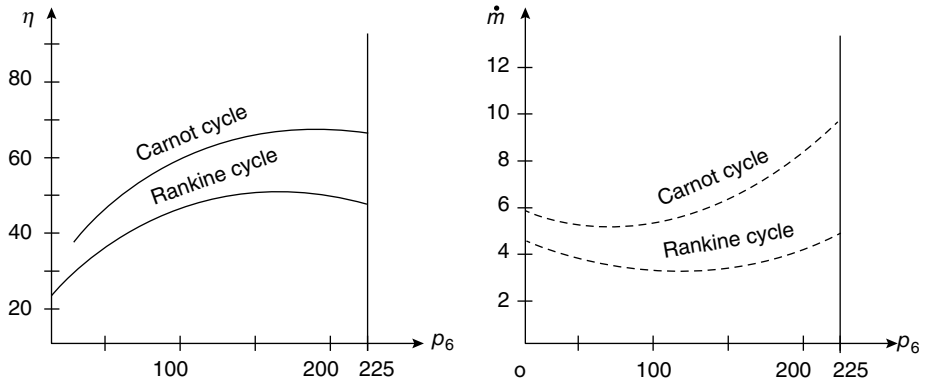


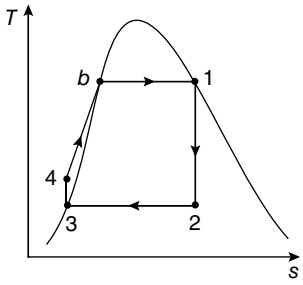
Fig. 4.15 *Efficiency and Specific Steam Consumption versus Boiler Pressure*

Example 4.1

A simple Rankine cycle works between the boiler pressure of 3 MPa and condenser pressure of 4 kPa. The steam is dry saturated before the throttling in the turbine. Determine

(i) Rankine cycle efficiency
(ii) work ratio
(iii) specific steam consumption

Solution: From the steam tables at 3 MPa (30 bar) and saturated vapour condition



$$h_1 = 2802.3 \text{ kJ/kg}$$

$$h_2 = h_{f2} + x\, h_{fg2}$$

$$= 121.4 + x_2 \times 2433.1$$

But, process 1–2 is isentropic,

$$\therefore\ s_1 = 6.1838 = s_2 = (s_{f2} + x_2\, s_{fg2})_{4\text{ kPa}}$$

$$\text{i.e. } 6.1838 = 0.4225 + x_2 \times 8.0530$$

$$x_2 = 0.7154$$

$$\text{Hence, } h_2 = 121.4 + 0.7154 \times 2433.1$$

$$= 1862.04 \text{ kJ/kg}$$

(i) Rankine cycle efficiency,

$$\text{Pump work} = v_3\,(p_4 - p_3)$$

$$w_p = 0.001\,(30 - 0.04)100$$

$$= 3 \text{ kJ/kg}$$

$$h_{f4} = h_{f3} + w_p$$

$$= 121.4 + 3$$

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$$= 124.4 \text{ kJ/kg}$$

$$\eta = \frac{(h_1 - h_2) - w_p}{(h_1 - h_{f4})}$$

$$= \frac{(2802.3 - 1862.04) - 3}{(2802.3 - 124.4)}$$

$$= \frac{937.26}{2677.9}$$

$$= 0.35 = 35\%$$

(ii) Work ratio

$$W_R = \frac{(h_1 - h_2) - w_p}{(h_1 - h_2)}$$

$$= \frac{(2802.3 - 1862.04) - 3}{(2802.3 - 1862.04)}$$

$$= \frac{937.26}{940.26} = 0.997$$

(iii) Specific steam consumption

$$m = \frac{1}{\text{Turbine work}}$$

$$= \frac{1}{(h_1 - h_2)} = \frac{1 \times 3600}{(2802.3 - 1862.04)}$$

$$= \frac{3600}{940.26}$$

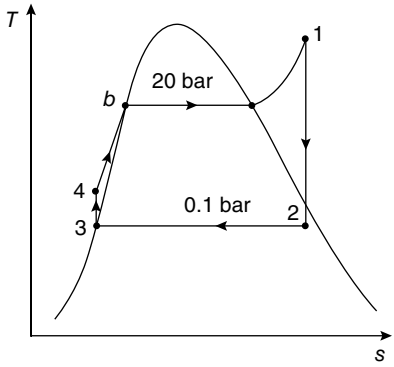
$$= 3.83 \text{ kg/kWh}$$

Example 4.2

In a steam power plant operating on ideal Rankine cycle, steam enters the turbine at 20 bar with an enthalpy of 3248 kJ/kg and an entropy of 7.127 kJ/kg K. The condenser pressure is 0.1 bar. Find the cycle efficiency and specific steam consumption in kg/kWh. Do not neglect pump work. You may make use of the extract of steam table given below.

p	t	h_f	h_g	s_f	s_g
(bar)	(°C)	kJ/kg		kJ/kg	
20.0	212.4	908.8	1890.7	2.447	6.331
0.1	45.81	191.83	2584.7	0.6493	8.1502

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Solution:

At 20 bar,

$$h_1 = 3248 \text{ kJ/kg}$$

$$s_1 = 7.127 \text{ kJ/kg K}$$

As process 1–2 is isentropic,

$$s_1 = s_2$$

$$= s_{f2} + x_2 \times s_{fg2}$$

i.e. $7.127 = 0.6493 + x_2 \times (8.1502 - 0.6493)$

$$\therefore x_2 = 0.864$$

At 0.1 bar,

$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$= 191.83 + 0.864 \times (2584.7 - 191.83)$$

$$= 191.83 + 0.864 \times 2392.87$$

$$= 2259.27 \text{ kJ/kg}$$

Pump work, $W_p = v_3 (p_4 - p_3)$

But, $v_3 = v_f = 0.001 \text{ m}^3/\text{kg}$ at 0.1 bar from steam tables*

$$\therefore W_p = 0.001 (20 - 0.1)100$$

$$= 1.99 \text{ kJ/kg}$$

Now, $W_p = h_{f4} - h_{f3}$

$$\therefore h_{f4} = W_p + h_{f3} \text{ at 0.1 bar}$$

$$= 1.99 + 191.83$$

$$= 193.82 \text{ kJ/kg}$$

Rankine cycle efficiency

$$\eta = \frac{(h_1 - h_2) - w_p}{(h_1 - h_{f4})}$$

$$= \frac{(3248 - 2259.27) - 1.99}{(3248 - 193.82)}$$

$$= \frac{986.74}{3054.18}$$

$$= 32.31\%$$

Specific steam consumption

$$m = \frac{3600}{\text{Turbine work}}$$

$$= \frac{1 \times 3600}{(h_1 - h_2)} = \frac{3600}{3248 - 2259.27}$$

$$= 3.64 \text{ kg/kWh}$$

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Example 4.3

A steam turbine receives steam at 15 bar and 300°C and leaves the turbine at 0.1 bar and 4% moisture. Determine

(i) Rankine efficiency

(ii) steam consumption per kW per hour if the efficiency ratio is 0.70

(iii) Carnot cycle efficiency for the given temperature limits

(iv) change in Rankine efficiency and specific consumption if the condenser pressure is reduced to 0.04 bar

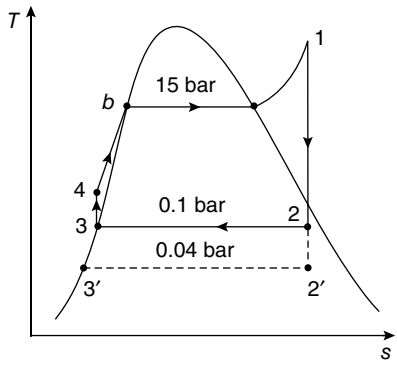
Solution:

From the steam tables,

at 15 bar 300°C,

$$h_1 = 3038.9 \text{ kJ/kg}$$

$$s_1 = 6.9207 \text{ kJ/kg}$$



At 0.1 bar and 4% dry,

$$h_2 = h_{f2} + x_2 h_{fg2}$$
$$= 191.8 + 2392.9 \times 0.96$$
$$= 2488.98 \text{ kJ/kg}$$

At 0.04 bar

$$s_1 = s_2' = s_{f2'} + x_2'.\, s_{fg2'}$$
$$6.9207 = 0.4225 + x_2' \times 8.0530$$
$$x_2' = 0.81$$

Hence,
$$h_{2'} = h_{f2'} + x_{2'} h_{fg2'}$$
$$= 121.4 + 0.81 \times 2433.1$$
$$= 2092.21 \text{ kJ/kg}$$

(i) Rankine efficiency

$$\text{pump work} = W_p = v_3 (p_4 - p_3)$$

but, $v_3 = v_f$ at 0.1 bar

$$= 0.001 \text{ m}^3/\text{kg}$$

$$\therefore W_p = 0.001 \times (15 - 0.1) \times 100$$
$$= 1.49 \text{ kJ/kg}$$

Again,
$$W_p = h_{f4} - h_{f3}$$
$$h_{f4} = 1.49 + 191.8$$
$$= 193.29 \text{ kJ/kg}$$

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Hence, Rankine efficiency

$$\eta = \frac{(h_1 - h_2) - w_p}{(h_1 - h_{f4})}$$

$$= \frac{(3038.9 - 2488.98) - 1.49}{(3038.9 - 193.29)}$$

$$= \frac{548.43}{2845.6} = 0.193$$

$$= 19.30\%$$

(ii) Specific fuel consumption

Now, efficiency ratio $= \dfrac{\text{Indicated thermal efficiency}}{\text{Rankine efficiency}}$

Indicated thermal efficiency $= 0.7 \times 0.193 = 0.135$

But, indicated thermal efficiency,

$$0.135 = \frac{3600}{\text{Turbine work}}$$

$$= \frac{1 \times 3600}{m(h_1 - h_2)}$$

$\therefore$ specific fuel consumption

$$m = \frac{3600}{0.135(3038.9 - 2488.98)}$$

$$= 48.49 \text{ kg/kWh}$$

(iii) Carnot efficiency

$$\eta = \frac{T_{max} - T_{min}}{T_{max}}$$

but, $T_{max} = 300°C$

$T_{min} = 45.83°C$ at 0.1 bar

Hence, $\eta = \dfrac{300 - 45.83}{300 + 273} = 0.4436$

$$= 44.36\%$$

(iv) Change in Rankine efficiency and specific fuel consumption at 0.04 bar condenser pressure

At 0.04 bar, condenser pressure

$$V_{3'} = V_{f3'} = 0.001 \text{ m}^3/\text{kg}$$

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pump work, $w_p = h_{f4'} - h_{f3'} = v_{3'}(p_4 - p_{3'})$

$= 0.001\ (15 - 0.04)\ 100$

$= 1.496$ kJ/kg

$h_{f4'} = 1.496 + 121.4 = 122.896$ kJ/kg

Rankine efficiency,

$$\eta = \frac{(h_1 - h_2) - w_p}{(h_1 - h_{f4})}$$

$$= \frac{(3038.9 - 2092.21) - 1.496}{(3038.9 - 122.896)}$$

$$= \frac{9451.94}{2916.0} = 0.324$$

$= 32.4\%$

It is evident that Rankine cycle efficiency increases when condenser pressure is reduced.

Indicated thermal efficiency = Rankine efficiency × Efficiency ratio

$= 0.324 \times 0.70$

$= 0.2268$

Hence, specific fuel consumption, $m = \frac{1 \times 3600}{m(h_1 - h_2)}$

$$= \frac{3600}{0.2268(3038.9 - 2092.21)}$$

$= 16.77$ kg/kWh

Steam consumption decreases by decreasing the condenser back pressure.

4.3.3 Modified Rankine Cycle

All steam engines work based on modified Rankine cycle. A modified Rankine cycle is shown in Figure 4.16. In this cycle, steam expands in the cylinder only up to state point 5 instead of expanding upto state point 3. This is because if steam is allowed to expand completely in the engine cylinder, then the stroke length will be too long. In addition, the work obtained is so small at the end of the stroke that it may hardly be sufficient to overcome friction. In Figure 4.16(b), loss due to termination of the stroke at point 5 instead of at point 3 results in loss of work corresponding to area 5–6–3–5 in the *p–h* and *T–s* diagram. Thus, pressure in the cylinder drops from p_2 to back pressure p_b at constant volume along the line 5–6. However, the loss of work due to this is comparatively small.

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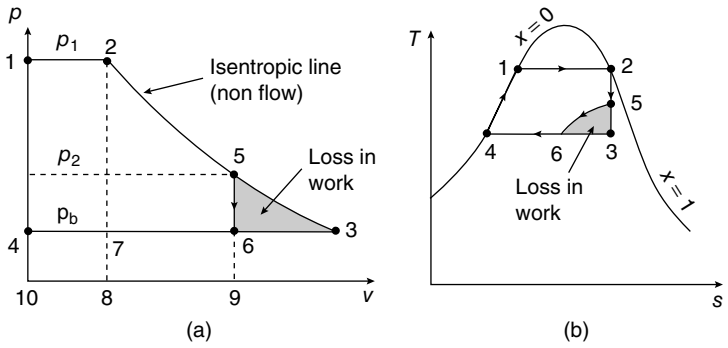


Fig. 4.16 *Modified Rankine Cycle*

In Figure 4.16(a), work done/kg of steam,

$$W = \text{Area } 1\text{–}2\text{–}5\text{–}6\text{–}4\text{–}1$$

$$= \text{Area } 1\text{–}2\text{–}8\text{–}10\text{–}1 + \text{Area } 2\text{–}5\text{–}9\text{–}8\text{–}2 - \text{Area } 4\text{–}6\text{–}9\text{–}10\text{–}4$$

$$= p_1 v_1 + (u_2 - u_5) - p_b v_5$$

$$= (u_2 - u_5) + p_1 v_1 - p_b v_5$$

$$= (h_2 - p_1 v_1) - (h_5 - p_2 v_5) + p_1 v_1 - p_b v_5$$

$$W = (h_2 - h_5) + v_5(p_2 - p_b)$$

where v_5 = Specific volume of steam at the end of isentropic expansion.

4.3.4 Reheat Cycle

As evident from our earlier discussion, efficiency of the simple Rankine cycle increases by increasing the pressure and the temperature of steam entering the turbine. If the steam pressure is increased at the turbine inlet, expansion ratio of the turbine also increases and hence steam becomes very wet at the end of expansion. This has an ill effect of erosion and corrosion of turbine blades, maximizing the losses, resulting in drop in efficiency of nozz le and blades. This difficulty is overcome by extracting steam at a suitable point in the turbine and reheating it again by some means.

A schematic diagram of a reheat cycle is shown in Figure 4.17.

In this cycle, the steam at point 1 is expanded in the turbine and a part of the steam is extracted at point 2 and returned back to the steam generator. Hence, the extracted steam is again heated at constant pressure (ideal case) to a temperature at 3, but nearer to temperature at 1. The reheated steam expands in the LPT to the condenser pressure.

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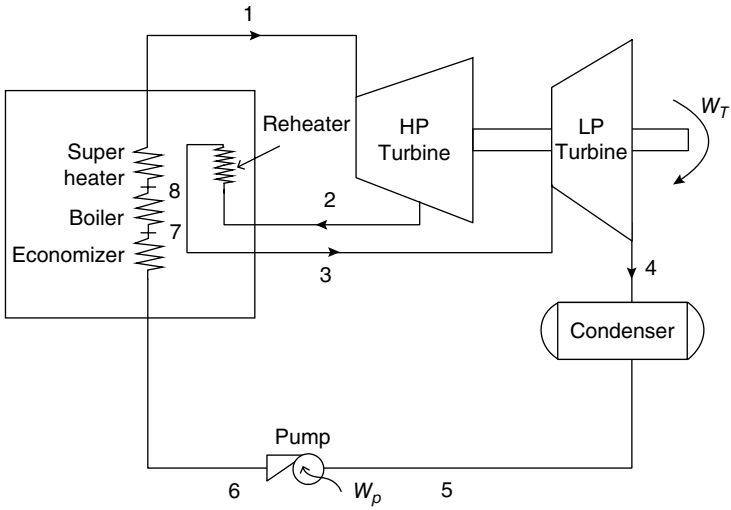


Fig. 4.17 *Reheat Cycle*

Advantages of reheat cycle are as follows:

(i) Dryness fraction of steam coming out of the turbine increases resulting in reduced erosion and corrosion of blades. Dryness fraction is maintained well above 90 per cent.

(ii) At higher operating pressure (p_1), thermal efficiency of the cycle increases provided the ratio of reheat pressure p_2 to the initial pressure p_1 is limited between 20 and 25 per cent.

(iii) Specific fuel consumption decreases.

In most of the power plants, reheating is limited to a maximum of two stages and higher operating pressures due to cycle complication and increased capital costs.

An ideal reheat cycle is shown in Figure 4.18 on the *T–s* diagram and *h–s* diagram.

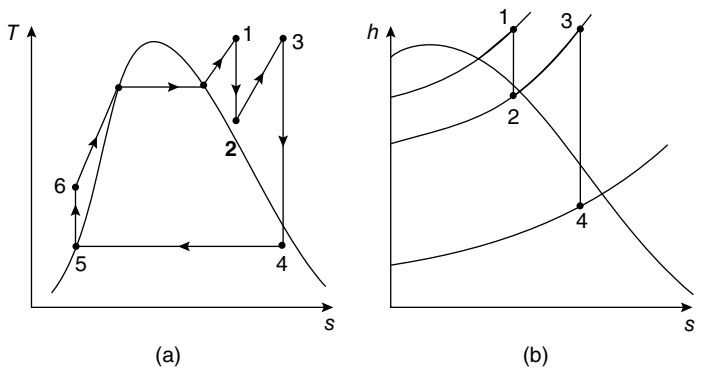


Fig. 4.18 *Reheat Cycle T–s Diagram and h–s Diagram*

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(iv) Heat added during constant pressure process 6–1 and 2–3,

$$q_A = (h_1 - h_6) + (h_3 - h_2)$$

$$q_A = (h_1 - h_{f6}) + (h_3 - h_2) \text{ kJ/kg}$$

(v) Heat rejected at constant pressure process 4–5,

$$q_R = (h_4 - h_5)$$

$$q_R = (h_4 - h_{f5}) \text{ kJ/kg}$$

(vi) Turbine work during the process 1–2 and 3–4

$$w_T = (h_1 - h_2) + (h_3 - h_4) \text{ kJ/kg}$$

(vii) Pump work during the process 5–6

$$w_p = h_6 - h_5$$

$$w_p = h_{f6} - h_{f5} \text{ kJ/kg}$$

Net work done/kg of steam

= Heat added – Heat rejected

$$= (h_1 - h_{f6}) + (h_3 - h_2) - (h_4 - h_{f5})$$

$$= (h_1 - h_2) + (h_3 - h_4) - (h_{f6} - h_{f5})$$

$$= (h_1 - h_2) + (h_3 - h_4) - w_p \text{ kJ/kg}$$

Thermal efficiency
Thermal efficiency is given by

$$\eta = \frac{q_w}{q_A} = \frac{(h_1 - h_2) + (h_3 - h_4) - w_p}{(h_1 - h_{f6}) + (h_3 - h_2)}$$

Example 4.4

A steam power plant incorporates an ideal reheat cycle to improve the existing efficiency. Steam at 30 bar and 250°C is supplied at the HPT inlet and expands till it is dry saturated at 3 bar. Now the steam is taken to a reheater and its temperature is again increased to 250°C at constant

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pressure reheating process. The reheated steam expands in the LPT to a condenser pressure of 0.04 bar. Determine the cycle efficiency.

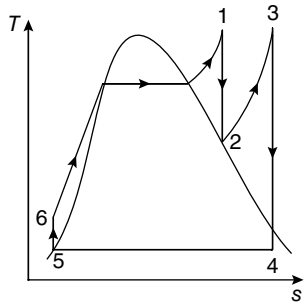
Solution: From the steam tables,

At 30 bar and 250°C

$$h_1 = 2854.8 \text{ kJ/kg}$$

At 3 bar and dry saturated condition,

$$h_2 = 2724.7 \text{ kJ/kg}$$



At 3 bar and 250°C

$$h_3 = 2967.9 \text{ kJ/kg}$$

$$s_3 = 7.5176 \text{ kJ/kg}$$

At 0.04 bar,

$$v_{f5} = v_5 = 0.001 \text{ m}^3/\text{kg}$$

$$h_{f4} = 121.4 \text{ kJ/kg}; \; h_{fg4} = 2433.1 \text{ kJ/kg}$$

$$s_{f4} = 0.4225 \text{ kJ/kg K}; \; s_{fg4} = 8.053 \text{ kJ/kg K}$$

equating $s_3 = s_4$

$$7.5176 = 0.4225 + x_4 \times 8.0530$$

$$x_4 = 0.881$$

$$\therefore \; h_4 = h_{f4} + x_4 \times h_{fg4}$$

$$= 121.4 + 0.881 \times 2433.1$$

$$= 2264.96 \text{ kJ/kg}$$

$$\text{pump work} = w_p = v_5 (p_6 - p_5)$$

$$\text{i.e., } h_{f6} - h_{f5} = 0.001 \; (30 - 0.04)100$$

$$= 3 \text{ kJ/kg}$$

$$h_{f6} = 3 + h_{f5} = 3 + h_{f4}$$

$$= 3 + 121.4 = 124.4 \text{ kJ/kg}$$

Thermal efficiency of the cycle,

$$h = \frac{q_w}{q_A} = \frac{(h_1 - h_2) + (h_3 - h_4) - w_p}{(h_1 - h_{f6}) + (h_3 - h_2)}$$

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$$= \frac{(2854.8 - 2724.7) + (2967.9 - 2264.96) - 3}{(2854.8 - 124.4) + (2967.9 - 2264.96)}$$

$$= \frac{830.04}{2973.6} = 0.279$$

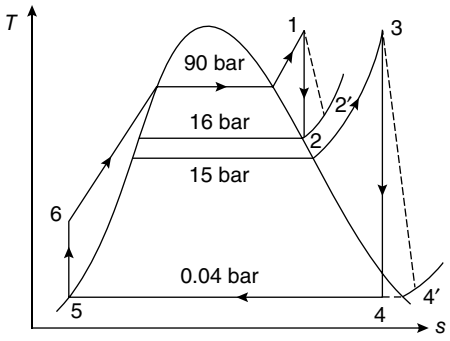
$$= 27.9\%$$

Example 4.5

An HP turbine receives steam from a boiler at 90 bar and 500°C and expands to a pressure of 16 bar and enters a reheater at 16 bar and 270°C. Pressure drops to 15 bar in the reheater and reheated steam at 500°C enters the LP turbine and finally gets exhausted to a condenser pressure of 0.04 bar. Assuming steam consumption of 3 kg/kWh, generator efficiency of 94 per cent and heat lost through the turbine casing as 1 per cent of throttle enthalpy, determine the efficiency of the turbine.

Solution: From the steam tables,

At 90 bar, 500°C;



$h_1 = 3386.1$ kJ/kg

$s_1 = 6.6576$ kJ/kg K

At 16 bar, $T_S = 474.4$ K

Now, $s_2 = s_1 = 6.6576$

$$= s_g + C_{ps} \ln \frac{T_{\text{sup}}}{T_s} \text{ at 16 bar}$$

$$= 6.4176 + 2.093 \ln \frac{T_{\text{sup}}}{474.4}$$

$$T_{\text{sup}} = 474.4 \times e^{0.115}$$

i.e., $T_{2'} = 532.2$ K

$p_{2'} = 16$ bar

$T_3 = 773$ K and $p_3 = 15$ bar

At 16 bar,

$$h_2 = h_{g2} + C_{ps} (T_{2'} - T_2)$$

$$= 2791.7 + 2.093 (532.2 - 474.4)$$

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$$= 2912.68 \text{ kJ/kg}$$

$$h_{2'} = 2919.2 + \frac{(280-250)}{(300-250)}(3034.8 - 2919.2)$$

$$= 2988.56 \text{ kJ/kg}$$

At 15 bar and 500°C,

$$h_3 = 3472.8 \text{ kJ/kg}$$

Considering the heat balance of the turbine as a system,

Energy entering the turbine = $h_1 + h_3$

Energy leaving the turbine = $h_{2'} + h_{4'} + \oint W$ + heat loss

But, heat loss = 0.01 h_1 = 0.01 × 3386.1 = 33.86 kJ/kg

$$\text{Net work, } = \frac{3600}{3 \times 0.94} = 1276.6 \text{ kJ/kg}$$

For energy balance,

Energy in = Energy out

i.e., $h_1 + h_3 = h_{2'} + h_{4'} + \oint W$ + heat loss

$$3386.1 + 3472.8 = 2988.56 + h_{4'} + 1276.6 + 33.86$$

$$\therefore h_{4'} = 2559.88 \text{ kJ/kg}$$

From the steam tables corresponding to 0.04 bar, and $h_{4'}$ values, $t_{4'} \approx 30°C$.
As saturation temperature is 28.98°C, the exhaust steam is superheated at point 4′.
Efficiency of the turbine,

$$\eta = \frac{h_1 - h_2}{h_1 - h_2} = \frac{3386.1 - 2988.56}{33861 - 2912.68} = \frac{397.54}{473.42} = 0.8397$$

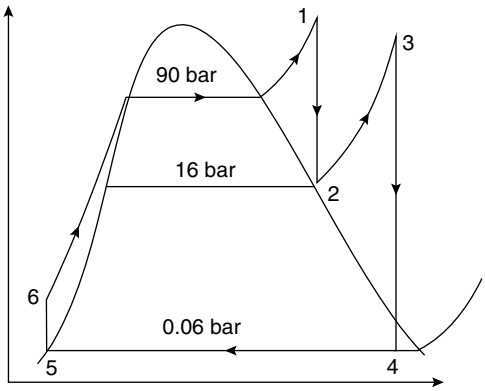
$$= 83.97\%$$

Example 4.6

An ideal reheat cycle has pressure at HP turbine inlet equal to 9 MPa, reheat pressure equal to 1.6 MPa and exhaust pressure equal to 7 kPa. The useful work developed by the turbine is 1400 kJ/kg. Determine the temperature of steam leaving the reheater, if thermal efficiency of the cycle is 38 per cent. Temperature at turbine inlet is 500°C and steam expands to dry saturated state before entering the reheater at 1.6 MPa.

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Solution: From the steam tables,



At 9 MPa, 500°C;

$$h_1 = 3386.1 \text{ kJ/kg}$$

At 1.6 MPa, dry saturated condition;

$$h_2 = 2791.7 \text{ kJ/kg}$$

At 0.06 bar

$$h_{f5} = 151.5 \text{ kJ/kg}$$

Now,

Thermal efficiency,

$$\eta = \frac{\oint W}{q_A}$$

$$\therefore q_A = \text{heat added}$$

$$= (h_1 - h_{f5}) + (h_3 - h_2)$$

$$= \frac{\oint W}{\eta} = \frac{1400}{0.38} = 3684.2 \text{ kJ/kg}$$

$$\therefore (h_3 - h_2) = 3684.2 - (h_1 - h_{f5})$$

$$h_3 = 3684.2 - (h_1 - h_{f5}) + h_2$$

$$= 3684.2 - (3386.1 - 151.5) + 2791.7$$

$$= 3241.3 \text{ kJ/kg}$$

From the steam tables at 16 bar and 3241.3 kJ/kg

$$t_3 = 350 + \frac{(3241.3 - 3145.4)}{(3254.2 - 3145.4)}(400 - 350)$$

$$= 350 + 44.07$$

$$= 394.07°C$$

Hence temperature of steam leaving the reheater is 394.07°C

Example 4.7

In a power plant working on a reheat cycle, the turbine develops 650 kJ/kg of work between the inlet and reheater and develops 760 kJ/kg between the reheater and condenser. After reheating steam enters the LPT at a pressure of 2.0 MPa and 500°C and expands down to condenser pressure of 8 kPa. Determine the thermal efficiency of the turbine (cycle efficiency) ignoring pump work.

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Solution: From the steam tables,

At 2 MPa and 500°C

$$h_3 = 3467.3 \text{ kJ/kg}$$

At 8 kPa,

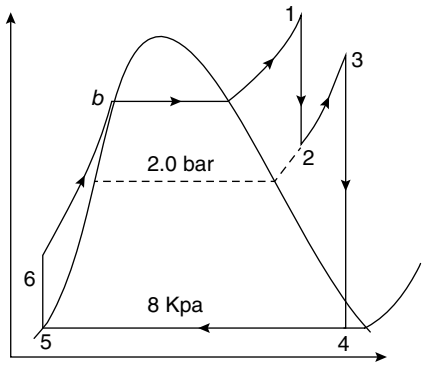
$$h_{f5} = 173.9 \text{ kJ/kg}$$

Thermal efficiency of the turbine,

Now,

$$h_1 - h_2 = 650 \text{ kJ/kg}$$

$$h_3 - h_4 = 760 \text{ kJ/kg}$$



Net work done by the turbine

$$W_T = (h_1 - h_2) + (h_3 - h_4)$$

$$= 650 + 760 = 1410 \text{ kJ/kg}$$

Heat supplied during the process,

$$q_A = (h_1 - h_{f5}) + (h_3 - h_2)$$

$$= (h_1 - h_2) + (h_3 - h_{f5})$$

$$= 650 + (3467.3 - 173.9)$$

$$= 3943.4 \text{ kJ/kg}$$

$$\eta = \frac{W_T}{q_A} = \frac{1410}{3943.4} = 0.3576$$

$$= 35.76\%$$

4.3.5 Regenerative Cycle

As it is known, reheating has limited ability to improve the thermodynamic efficiency of the cycle but is quite useful in the reduction of moisture in the turbine. However, it is observed that the largest single loss of energy in a power plant occurs at the condenser in which heat is rejected to the coolant. Hence, reducing this rejected heat drastically improves cycle efficiency.

In both ideal Rankine and reheat cycles, the condensate is returned to the boiler at the lowest temperature of the cycle. The fluid is heated to saturation by direct mixing in the steam drum of the boiler, by furnace radiation in boiler tubes or by gas convection heating by the flue gases in the economizer. All these methods involve large temperature differences and are inherently irreversible. Instead of resorting to such procedure, a method of feed water heating is considered.

An ideal regenerative cycle is shown in Figure 4.19. The steam expands in a reversible manner such that area under the curve 2–3 (2–3–4–*B*–2) would be exactly

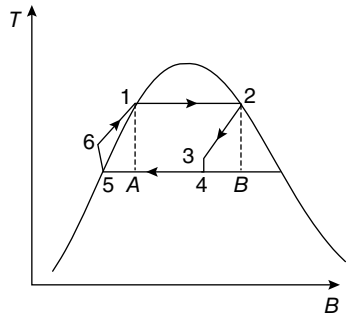


Fig. 4.19 *Ideal Regenerative T-s Diagram*

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equal to the area under the curve 6–1 (6–1–*A*–5–6). That means the increase in entropy during heating equals to the decrease during the expansion and cooling of vapour. Thus, the cycle is equivalent to a Carnot cycle operating between maximum and minimum temperatures.

In practice, the ideal cycle is approached by allowing the condensate from the feed pump to be heated in a separate heater or heaters by steam extracted from the turbine after it was partly expanded and work done. The extracted steam either mixes directly with the condensate, as in an open heater, or exchanges heat indirectly and condense as in a closed heater. The schematic of an ideal regenerative cycle power plant is shown in Figure 4.20.

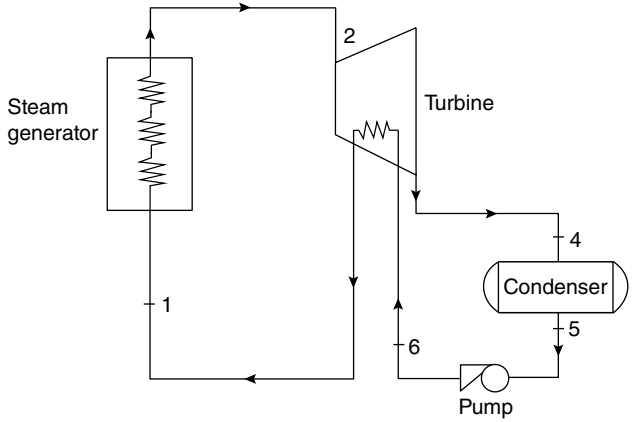


Fig. 4.20 *Ideal Regenerative Cycle Power Plant*

An infinite number of heat exchangers, known as feed water heaters, are essential in order to carryout ideal regeneration process. These feed water heaters are used to preheat the condensate with steam extracted from the steam turbine. This is practically not feasible. In addition, the thermal efficiency gain due to the addition of heaters drops as the number of heaters increases. A point is reached so that further addition of heaters is no longer economically justified due to the increased capital cost. In practice, maximum six to seven heaters are employed in very large power plants (refer Figure 4.21).

Work done by the turbine during process 3–4, 4–5, 5–6 and 6–7

$$w_T = (h_3 - h_4) + (1 - m_1)(h_4 - h_5) + (1 - m_1 - m_2)(h_5 - h_6)$$

$$+ (1 - m_1 - m_2 - m_3)(h_6 - h_7)$$

1. *Net work done by the cycle*

$$= \text{Heat added} - \text{Heat rejected} = q_A - q_R$$

$$= (h_3 - h_{f1}) - (1 - m_1 - m_2 - m_3)(h_7 - h_8)$$

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2. *Thermal efficiency of the cycle*
   Thermal efficiency is given by

$$\eta = \frac{q_w}{q_A} = \frac{(h_3 - h_1) - (1 - m_1 - m_2 - m_3)(h_7 - h_8)}{(h_3 - h_1)}$$

Note that the temperature rise of the liquid due to compression 14–1 is small and hence $h_1 \approx h_{14}$.

3. *Specific steam consumption*

$$m_S = \frac{3600}{q_w} \text{ kg/kWh}$$

4. *Quantities of extraction*
   For finding the various quantities of extraction, it is assumed that energy given by extraction steam is completely absorbed by the feed water.

   Energy balance around feed water heater 1

$$m_1 (h_4 - h_{13}) = (h_{14} - h_{13})$$

$$m_1 = \frac{(h_{14} - h_{13})}{(h_4 - h_{13})}$$

   Energy balance around feed water heater 2

$$m_2 (h_5 - h_{11}) = (1 - m_1)(h_{12} - h_{11})$$

As $h_{11} \approx h_{10}$,

$$m_2 = \frac{(1 - m_1)(h_{12} - h_{10})}{(h_5 - h_{10})}$$

   Energy balance around feed water heater 3

$$m_3 (h_6 - h_9) = (1 - m_1 - m_2)(h_{10} - h_9)$$

As $h_9 \approx h_8$

$$m_3 = \frac{(1 - m_1 - m_2)(h_{10} - h_8)}{(h_6 - h_8)}$$

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4.3.5.1 Types of Feed Water Heaters

Feed water heaters may be generally classified into two types, namely open and closed.

1. *Open type or direct contact feed water heaters*

In open-type feed water heater, the extraction steam is directly mixed with the incoming sub-cooled feed water to produce the saturated water at the extraction steam pressure. Figure 4.21 shows a Rankine cycle using three feed water heaters extracting m_1, m_2 and m_3 kg of steam at different extraction pressures. The condensate from the condensate extraction pump is pumped from heater in successive steps, which in turn mixes with extracted steam in heaters 3, 2 and 1 by direct mixing. Temperature of the condensate is thus increased before it enters the boiler at condition 1.

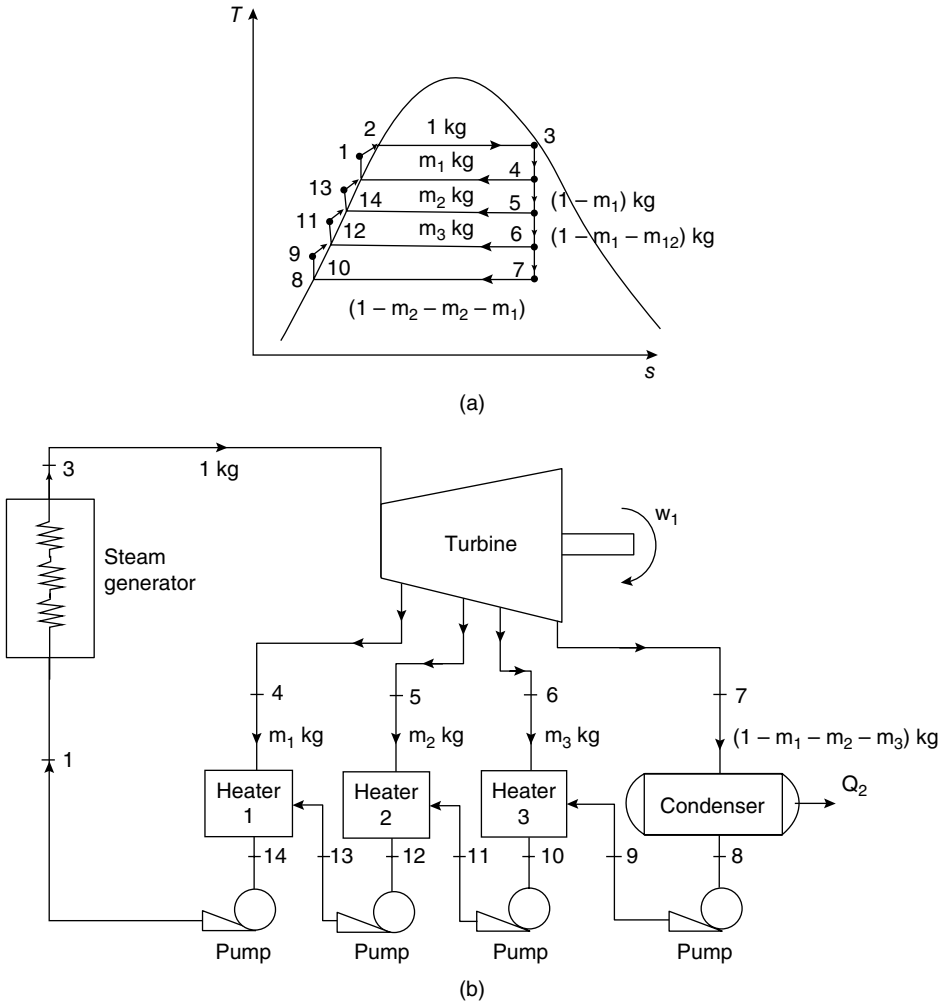


Fig. 4.21 *Open-Type or Direct Contact Feed Water Heaters*

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2. *Closed-type or indirect contact feed water heaters*

In closed-type feed water heater, feed water passes through the tubes, and the bled steam, on the shell side, transfers its energy to it and condenses. Figure 4.22 shows a Rankine cycle plant with closed-type feed water heaters. The condensate pump pumps condensate at pressure 6 to the feed water heaters. Heat transfer between the bled steam at extraction points 2 and 3 in the feed water and the condensate takes place resulting in a high-temperature feed water at condition 8 entering the steam generator.

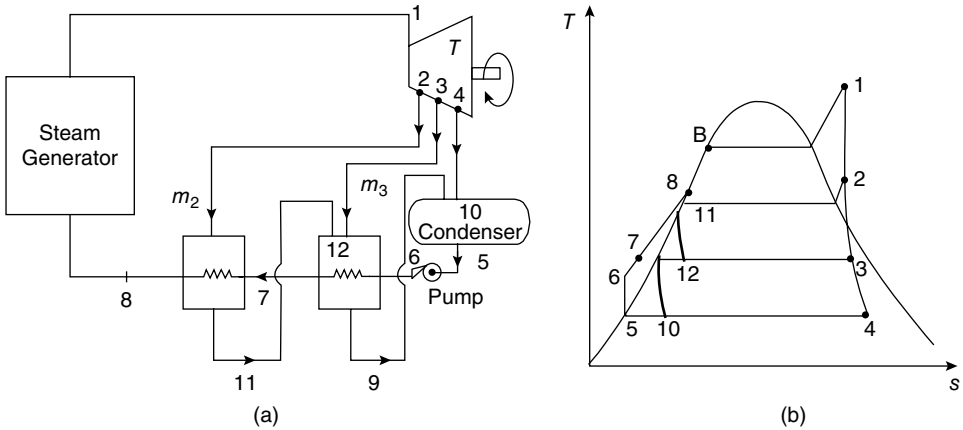


Fig. 4.22 *Closed-Type or Indirect Contact Feed Water Heaters: (a) Closed Feed Water Heater Plant; (b) Cycle on T-s Diagram*

Open-type feed water heaters are less expensive and possess better heat-transfer characteristics compared with closed-type heaters. However, they require a separate pump to handle the feed water between each heater. Chances of corrosion are more in the open-type heaters.

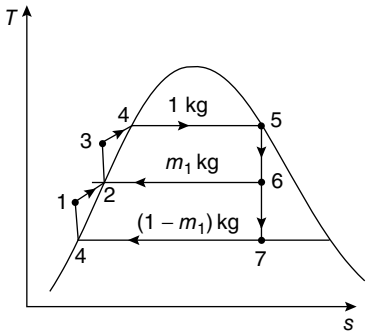
Example 4.8

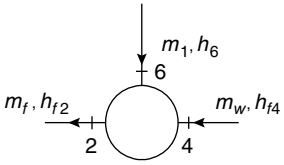
A power plant turbine receives steam at an enthalpy of 4000 kJ/kg. The enthalpy of extracted steam to the condenser is 2000 kJ/kg. The turbine bleeds steam for heating feed water in a regenerative feed water heater about 4 kg of steam per sec at a pressure of 4 bar. The quality of steam is 90 per cent dry. The condensate coming from the condenser is fed to the heater by means of a pump. The condensate has an enthalpy of 150 kJ/kg before entering the heater and becomes saturated while leaving the heater at 4 bar. Determine the power developed by the turbine.

Solution: From steam tables at 4 bar, 0.9 dry.

$$h_6 = h_{f\,6} + x h_{f\,g\,6}$$
$$= 604.7 + 0.9 \times 2132.9$$
$$= 2524.31 \text{ kJ/kg}$$
$$h_{g\,6} = 2737.6 \text{ kJ/kg}$$
$$h_5 = 4000 \text{ kJ/kg}$$

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$h_7 = 2000$ kJ/kg

$h_{f4} = 150$ kJ/kg

$h_{f2} = 604.7$ kJ/kg at 4 bar (saturated)

Writing energy balance for the feed water,
if m_f = steam entering the turbine, kg/s
then, $m_f \times h_{f2} = m \times h_6 + m_w \times h_{f4}$

$m_f \times 604.7 = 4 \times 2524.31 + (m_f - 4) \times 150$; $m + m_w = m_f$

$$m_f = \frac{9497.24}{454.7}$$

$$= 20.89 \text{ kg/s}$$

Hence, expressing mass of steam bled/kg of steam entering the turbine,

$$m = \frac{4}{20.89} = 0.191$$

Net work done by the turbine,

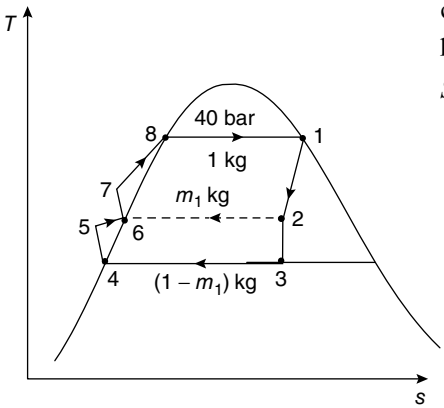
$$W_T = (h_5 - h_6) + (1 - m)(h_6 - h_7)$$
$$= (4000 - 2524.1) + (1 - 0.191)(2524.1 - 2000)$$
$$= 1475.9 + 424$$
$$= 1899.9 \text{ kJ/kg}$$

Power developed by the turbine,

$$P = m_f \times W_T$$
$$= 20.89 \times 1899.9$$
$$= 39.69 \text{ MW}$$

Example 4.9

A regenerative cycle has turbine inlet pressure of 40 bar and dry saturated. Steam expands in the condenser to a pressure of 0.04 bar. Steam is bled at optimum pressure from the turbine to heat the condensate water in the feed water heater. Neglecting pump work determines the cycle efficiency.



Solution: From steam tables at 4 bar, 0.9 dry.

Dry saturated condition,

$t_S = 250.33°C$

$h_1 = 2800.3$ kJ/kg

$S_1 = 6.0685$ kJ/kg K

At 0.04 bar,

$t_S = 28.98°C$

$s_{f3} = 0.4225$ kJ/kg K

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$$s_{fg3} = 8.4755$$

equating $s_1 = s_3$ (ideal case)

$$6.0685 = 0.4225 + x_3 \times 8.4765$$

$$x_3 = 0.67$$

Hence, $h_3 = (h_f + x_{hfg})$ at 0.04 bar

$$= 121.4 + 0.67 \times 2433.1$$

$$= 1751.6 \text{ kJ/kg}$$

$$h_{f3} = h_{f4} = 121.4 \text{ kJ/kg}$$

At optimum pressure of bleeding, the temperature (regenerative) can be approximated as

$$t_2 = \frac{t_1 + t_3}{2} = \frac{250.33 + 28.98}{2} = 139.66°\text{C}$$

Corresponding to this temperature, the pressure can be found by interpolation.

$$p_2 = 35 + \frac{(139.66 - 138.88)}{(143.63 - 138.88)}(40 - 35) = 3.58 \text{ bar}$$

Corresponding to this pressure, equating s_1 and s_2,

$$6.0685 = 1.7390 + x_2 \times 5.1894$$

$$x_2 = 0.834$$

$$\therefore h_2 = 589.10 + 0.834 \times 2144$$

$$= 2377.2 \text{ kJ/kg}$$

$$h_{f6} = 589.10 \text{ kJ/kg}$$

Considering the energy balance around the heater,

$$m_1 (h_2 - h_{f5}) = (1 - m_1)(h_{f6} - h_{f4}); \; h_{f5} \approx h_{f4}$$

i.e. $m_1(2377.2 - 121.4) = (1 - m_1)(589.10 - 121.4)$

$$2255.8\, m_1 = 467.7 - 467.7\, m_1$$

$$\therefore m_1 = \frac{467.7}{2723.5} = 0.172 \text{ kg}$$

Net work done by the turbine,

$$W_T = (h_1 - h_2) + (h_2 - h_3)(1 - m_1)$$

$$= (2800.3 - 2377.2) + (2377.2 - 1751.6)(1 - 0.172)$$

$$= 423.1 + 518 = 941.1 \text{ kJ/kg}$$

Heat added, $q_A = (h_1 - h_{f6})$

$$= (2800.3 - 589.10) = 2211.2 \text{ kJ/kg}$$

Hence, cycle efficiency,

$$\eta = \frac{\oint W}{q_A} = \frac{941.1}{2211.2} = 0.426$$

$$= 42.6\%$$

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Example 4.10

A steam turbine gets steam at 60 bar and 450°C from a steam generator that expands to a condenser pressure of 0.07 bar. Some amount of steam is bled from the turbine at 3 bar to heat the feed water from the condenser. Steam turbine generates 30 MW through a directly coupled generator with 95 per cent efficiency. Assuming turbine efficiency of 90 per cent determines

(i) the amount of steam bled/kg of steam entering the turbine.

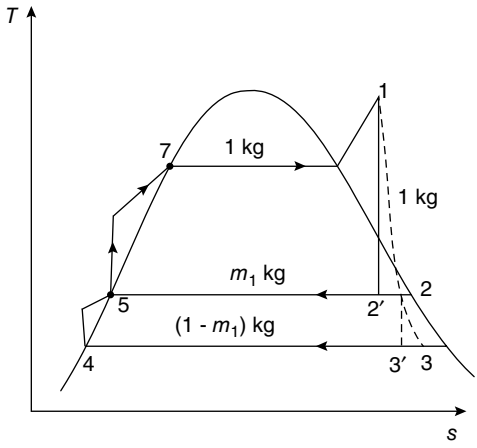
(ii) the steam generation per hour.

(iii) the overall efficiency of the plant if boiler efficiency is 92 per cent and mechanical efficiency is 98 per cent.

Assume the pump work is negligible and 15 per cent of generated power is used to run auxiliaries.

Solution: The actual expansion in the turbine is shown by dotted lines on the *T*–*s* diagram.

From steam tables at 60 bar, 450°C



$$h_1 = 3301.8 \text{ kJ/kg}$$

$$s_1 = 6.7193 \text{ kJ/kg K}$$

At 3 bar, for process 1–2′

$$s_1 = s_{2'}$$

$$6.7193 = 1.6717 + x_{2'} \times 5.3192$$

$$x_{2'} = 0.95$$

$$h_{2'} = 561.4 + 0.95 \times 2163.2$$

$$= 2616.44 \text{ kJ/kg}$$

$$\therefore (h_1 - h_2) = \text{turbine efficiency} \times (h_1 - h_{2'})$$

$$= 0.90 \times (3301.8 - 2616.44)$$

$$= 616.82 \text{ kJ/kg}$$

also, $h_2 = h_1 - 616.82$

$$= 3301.8 - 616.82$$

$$= 2684.98 \text{ kJ/kg}$$

$$= 561.4 + x_2 \times 2163.2 \text{ at 3 bar}$$

$$x_2 = 0.982$$

$$\therefore s_2 = 1.6717 + 0.982 \times 5.3192$$

$$= 6.895 \text{ kJ/kg K}$$

Considering the process 2–3′ (isentropic)

$$s_2 = s_{3'} \text{ at 0.07 bar}$$

$$6.895 = 0.5591 + x_{3'} \times 7.7176$$

$$x_{3'} = 0.821$$

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$$\therefore h_{3'} = 163.4 + 0.821 \times 2409.2 \text{ at } 0.07 \text{ bar}$$
$$= 2141.35 \text{ kJ/kg}$$
$$\therefore (h_2 - h_3) = 0.9 \times (h_2 - h_{3'})$$
$$= 0.9 \times (2684.98 - 2141.35)$$
$$= 489.27 \text{ kJ/kg}$$

also,
$$h_3 = h_2 - 489.27$$
$$= 2684.98 - 489.27$$
$$= 2195.71 \text{ kJ/kg}$$
$$h_{f4} = 163.4 \text{ kJ/kg at } 0.07 \text{ bar}$$
$$h_{f5} = 561.4 \text{ kJ/kg at } 3 \text{ bar}$$

(i) The amount of steam bled/kg of steam entering the turbine

Making energy balance,

$$m\,(h_2 - h_{f4}) = (h_{f5} - h_{f4})$$
$$m = \frac{(h_{f5} - h_{f4})}{(h_2 - h_{f4})}$$
$$= \frac{(561.6 - 163.4)}{(2684.98 - 163.4)} = \frac{398}{2521.58} = 0.158 \text{ kg}$$

(ii) The steam generation per hour,

Net work done by the turbine,

$$W_T = (h_1 - h_2) + (1 - m)\,(h_2 - h_3)$$
$$= 616.82 + (1 - 0.158) \times 489.27$$
$$= 616.82 + 411.965$$
$$= 1028.785 \text{ kJ/kg}$$

Actual turbine work
$$= \frac{30}{0.95 \times 0.98} = 32.22 \text{ MW}$$

Steam generated in kg/s
$$m_S = \frac{W_{act}}{W_T} = \frac{32.22 \times 10^3 \text{ kJ/s}}{1028.785 \text{ kJ/kg}} = 31.32 \text{ kg/s}$$
$$= 112.752 \text{ tonnes/hour}$$

(iii) Heat supplied to the boiler
$$q_A = \frac{m_s(h_1 - h_{f5})}{0.92}$$
$$= \frac{31.32(3301.8 - 561.4)}{0.92}$$
$$= 93.293 \text{ MW}$$

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As 15 per cent of turbine work is utilized for running auxiliaries, actual generated power available is

$$= 30 \times 0.85$$

$$= 25.5 \text{ MW}$$

Overall plant efficiency,

$$\eta_0 = \frac{\text{Net available power}}{\text{Heat supplied}}$$

$$= \frac{25.5}{93.293} = 0.2733$$

$$= 27.33\%$$

4.4 WORKING OF DIFFERENT CIRCUITS

Figure 4.23 shows a general layout of coal-fired thermal power station. The layout consists of the following four important systems:

1. coal- and ash-handling system
2. air and flue gas system
3. cooling water system
4. feed water and steam generation system

A steam power plant works on *Rankine cycle*. Steam generated in the boiler expands in the *steam turbine*. Low-pressure exhaust steam from the turbine is condensed in the *condenser*. The condensate is fed back to the boiler along with feed water.

A thermal power plant mainly uses hot products of combustion known as *flue gas* to generate steam in a steam generator. Heat is generated by burning coal, oil or gas. For this purpose, boiler furnace is designed for firing coal, air or gas.

High-pressure steam generated in the steam boiler is passed through steam turbine where steam expands in the nozzles. Heat energy of steam is converted into mechanical energy. Mechanical energy is utilized to run the generator and hence to generate electric power. The low-pressure steam after expanding in the turbine enters the condenser. Here, steam condenses by rejecting heat to the cooling water. The condensed steam is supplied back to the boiler using condensate extraction pump. The overall efficiency of the plant is increased by using economizer, air preheater, super heater and feed water heaters.

A brief explanation of different flow paths of a power plant is given below:

1. *Coal- and ash-handling system*

In this system, coal from the coal yard is crushed to the required size in coal crusher. The *crushed* coal is supplied to the coal bunker in the boiler house. This *coal* is fed to the furnace through coal *hoppers* where it is burnt. The products of combustion, namely ash and unburnt coal, are disposed to the ash storage site.

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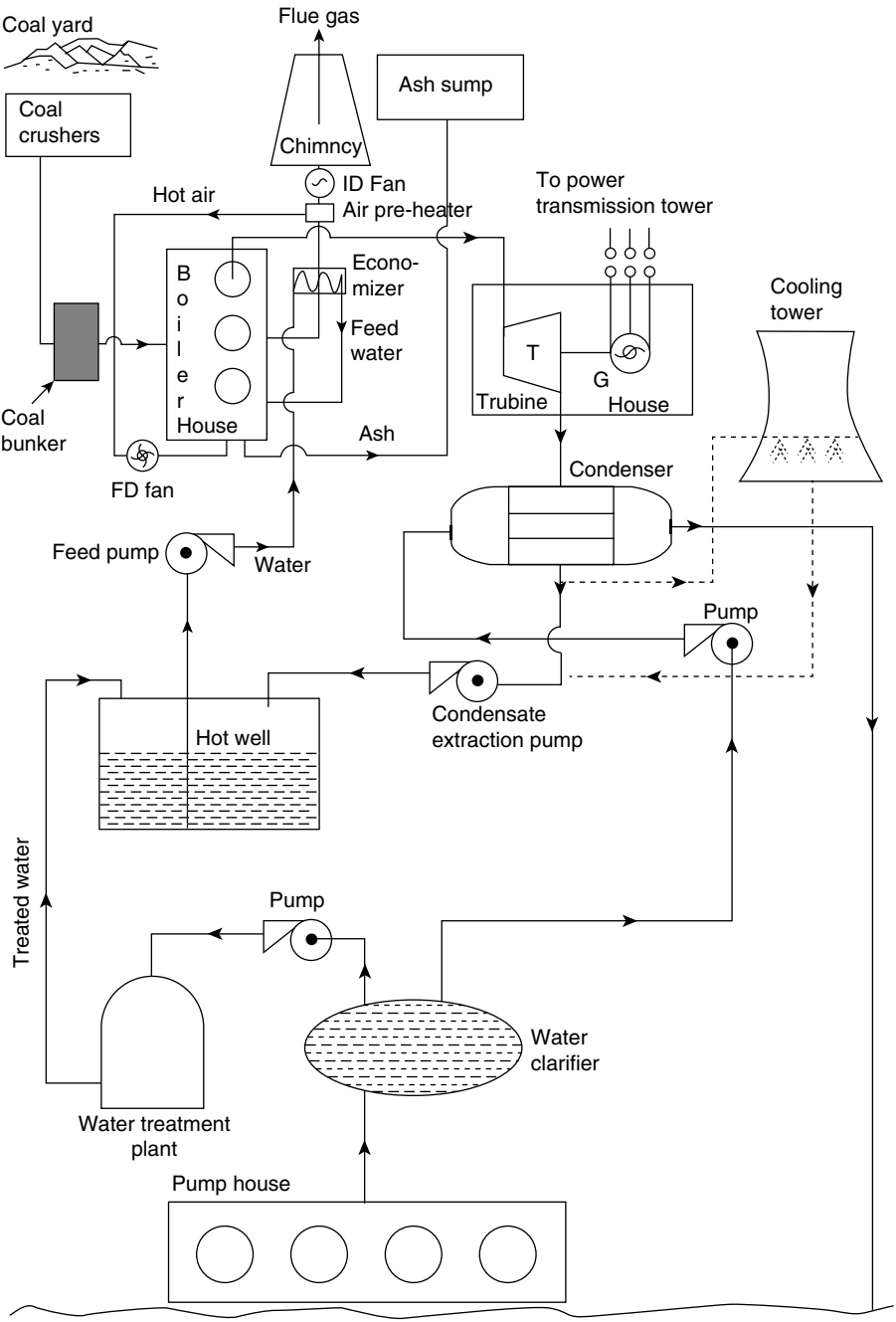


Fig. 4.23 *Coal-Fired Thermal Power Station Showing All Working Systems*

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2. *Air and flue gas system*

In this system, air from the air heater is supplied to the boiler furnace using *forced draught* fan. The products of combustion in the form of flue gases are circulated in the furnace to heat the water in the tubes. The heat in the exhaust flue gases is utilized by passing it first over *economizer* tubes and then over air *preheater* tubes. After extracting maximum possible heat, the flue gases are discharged through the chimney using an *induced draught* fan.

3. *Cooling water system*

This system mainly consists of a *condenser, condensate extraction pump, cooling tower* and *cooling water circulation pump*. When plenty of water is not available, a cooling tower is preferred. Water pumped from the water clarifier plant is circulated through the condenser tubes in order to condense the exhaust steam coming from the turbine. Water leaving the condenser is discharged to the downward side of the river. Such a system is known as open system, which is very useful when abundant water supply is available. When there is scarcity of water, a cooling tower is installed. This system is also known as a *closed system*. In this system, the condensate water is cooled by passing it through the cooling tower. The cold water is pumped to the hot well. The loss of water due to evaporation is compensated by supplying water directly from the river.

4. *Feed water and steam generation system*

This system mainly consists of *feed water pump, steam generator, condenser* and *hot well*. Feed water from the hot well is supplied to the steam generator by using the feed pump. Steam generated in the boiler is superheated in super heater tubes. Superheated steam is supplied to the prime mover to develop power. The steam exhausted in the prime mover is condensed in the condenser. Any water that is lost in the system is compensated by supplying feed water from an external source. Such feed water should be treated in a feed water treatment plant before sending it to the hot well.

4.5 SELECTION OF SITE FOR STEAM POWER PLANTS

Selecting a proper site for a thermal power plant is vital for its long-term efficiency. Many factors have to be considered regarding suitable location of a plant. As it may not be possible to satisfy all the requirements in a single location, an ideal location should contain an optimum mix of the feasible requirements, for long-term economic justification of the plant. The factors to be considered for site selection of steam power plants are but not limited to nearness to load centre, supply of water, availability of coal, availability of land at a reasonable price, transportation facilities, availability of labours and distance from populated area.

1. *Availability of coal*

As the power plant consumes large amount of coal, enough quantity must be available either in the vicinity or it should be easily and economically viable to transport the coal to the power plant site. Proximity to sea route and rail transport are the major criteria.

2. *Transportation facilities*

A typical power plant with 1000 MW capacity approximately consumes more than 10,000 tons of coal per day. Hence, the necessity for continuous supply and storage capability of coal in the

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power station by rail, by sea and by road must be ensured. It is also essential to have easy and enough access to transportation network during power plant construction and operation periods.

3. *Availability of land*

Ash is the main by-product of combustion and as the amount of coal used is large, proportionately, thousands of tons of ash are generated per day. Ash as it comes out from the boiler is hot and is very corrosive in nature. Disposing of such huge quantities of ash requires a large amount of empty space where it can be safely dumped.

In addition, enough space must be available for the storage of coal, disposal of ash, building of the power plant, township or residential colony of workers, markets, etc. Normally, for every MW of power generated, at least 3–4 acres of land is available for this purpose.

4. *Supply of water*

A large amount of water is required for cooling and ash-handling systems in the power plant. Hence, it is essential to have water resources in the form of rivers, lake, sea, etc., nearby. During construction and operation stages also, different quantities of water are required. However, care should be taken to ensure that the site is at a reasonable distance from permanent and seasonal rivers and floodways.

5. *Availability of labours*

Apart from these major requirements, availability of skilled labour is also essential in the plant locality, as the semi- and unskilled labour requirements are very high. Proper transportation facility is very vital for the arrangement of labours.

6. *Geology and soil type*

The power plant should be built on a land that is not prone to vibrations generated by rotating equipments. Proper inspection and testing of soil and rock layers is essential. Also, the site should be away enough from the faults and earthquake-prone areas as weak and small earthquakes can damage many parts of a power plant intensively.

7. *Distance from populated area*

Location of the plant must be so chosen that it has minimum or no ill impact on the surrounding environment. Plant location should be away from national parks, wildlife, protected areas, etc. As nearby forests, ecosystem, etc., are very sensitive to the pollutions caused by a power plant, ideal location should be away from them. Similarly, for the same reasons, the site should be away from population centres.

8. *Nearness to load centre*

In general, the site should be near to the load centres to avoid transmission and distribution losses. It should have the capacity to decrease the amount of power loss and transmission expenses. It should be near to the local electric grid.

9. *Climatic conditions*

As climatic conditions such as temperature, humidity, wind direction and speed affect the productivity of a power plant, proper study must be conducted before installing the major components such as chimneys and cooling towers.

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4.6 QUESTIONS

4.6.1 Objective Questions

1. Carnot cycle comprises
   (a) two isentropic processes and two constant volume processes
   (b) two isentropic processes and two constant pressure processes
   (e) two isothermal processes and three constant pressure processes
   (d) none of the above
2. In Rankine cycle, the work output from the turbine is given by
   (a) change of internal energy between inlet and outlet
   (b) change of enthalpy between inlet and outlet
   (c) change of entropy between inlet and outlet
   (d) change of temperature between inlet and outlet
3. The overall efficiency of thermal power plant is equal to
   (a) Rankine cycle efficiency
   (b) Carnot cycle efficiency
   (c) Regenerative cycle efficiency
   (d) Boiler efficiency × turbine efficiency × generator efficiency
4. Rankine cycle efficiency of a good steam power plant may be in the range
   (a) 15–20 per cent
   (b) 35–45 per cent
   (c) 70–80 per cent
   (d) 90–95 per cent
5. Rankine cycle operating on low-pressure limit of p_1 and high-pressure limit of p_2
   (a) has lower thermal efficiency than Carnot cycle operating between same pressure limit
   (b) has same thermal efficiency as Carnot cycle operating between same pressure limits
   (c) may be more or less depending on the magnitude of p_1 and p_2
   (d) has higher thermal efficiency than the Carnot cycle operating between same pressure limits
6. Rankine efficiency of a steam power plant
   (a) improves in summer compared with that in winter
   (b) improves in winter compared with that in summer
   (c) is unaffected by climatic conditions
   (d) none of the above
7. Regenerative cycle thermal efficiency
   (a) is always greater than simple Rankine thermal efficiency
   (b) is greater than simple Rankine cycle thermal efficiency only when steam is bled at particular pressure
   (c) is same as simple Rankine cycle thermal efficiency
   (d) is always less than simple Rankine cycle thermal efficiency

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8. In a regenerative feed heating cycle, the optimum value of the fraction of steam extracted for feed heating
   (a) is unaffected by increase in Rankine cycle efficiency
   (b) decreases with increase in Rankine cycle efficiency
   (c) increases with increase in Rankine cycle efficiency
   (d) none of the above
9. In a regenerative feed heating cycle, the greatest economy is affected
   (a) when steam is extracted only from the first stage of steam turbine
   (b) when steam is extracted from several places in different stages of steam turbine
   (c) when steam is extracted only from the last stage of steam turbine
   (d) when steam is extracted from only one suitable point of steam turbine
10. The maximum percentage gain in regenerative feed heating cycle thermal efficiency
   (a) remains same unaffected by number of feed heaters
   (b) increases with number of feed heaters increasing
   (c) decreases with number of feed heaters increasing
   (d) none of the above
11. In regenerative cycle feed water is heated by
   (a) exhaust gases (b) heaters
   (c) draining steam from the turbine (d) all of the above
12. Reheat cycle in steam power plant
   (a) utilizes heat of flue gases (b) increases thermal efficiency
   (c) improves condenser performance (d) reduces loss of heat
13. Steam engines work based on
   (a) Rankine cycle (b) modified Rankine cycle
   (c) Carnot cycle (d) Otto cycle
14. A steam power plant works based on
   (a) Rankine cycle (b) modified Rankine cycle
   (c) Otto cycle (d) Carnot cycle
15. A Rankine cycle does not have the following process:
   (a) reversible adiabatic (b) isobaric
   (c) isochoric (d) isothermal
16. Thermal efficiency of the Rankine cycle with pump work is

   (a) $\eta = \dfrac{(h_1 - h_2) - W_p}{h_1 - h_{f4}}$ (b) $\eta = \dfrac{(h_1 + h_2) - W_p}{h_1 - h_{f4}}$

   (c) $\eta = \dfrac{(h_1 - h_2) + W_p}{h_1 - h_{f4}}$ (d) $\eta = \dfrac{(h_1 - h_2)}{(h_1 - h_{f4})}$

   where h_1 and h_2 are inlet and outlet enthalpies, and h_{f4} is the enthalpy of water at condenser pressure

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17. The efficiency of Rankine cycle lies in the range
    (a) 10–20 per cent (b) 20–30 per cent
    (c) 40–60 per cent (d) 30–40 per cent
18. Lowering the condenser pressure in a Rankine cycle
    (a) decreases thermal efficiency (b) increases thermal efficiency
    (c) does not affect thermal efficiency (d) none of these
19. Pick the wrong statement with reference to a Rankine cycle.
    (a) Raising boiler pressure and temperature raises thermal efficiency
    (b) Lowering the condenser pressure raises the thermal efficiency
    (c) Raising the steam temperature raises the thermal efficiency
    (d) Raising the condenser pressure raises the thermal efficiency

Answers

1. b 2. b 3. c 4. b 5. d 6. b 7. a 8. c 9. b 10. b 11. c 12. b
13. a 14. b 15. d 16. a 17. c 18. b 19. d

4.6.2 Review Questions

1. Draw a neat diagram of thermal power plant using coal as fuel and explain briefly its working.
2. Draw a general layout of a steam power plant showing the different circuits and systems and explain them.
3. With the help of *T–s* diagram deduce an expression for Rankine cycle efficiency.
4. Discuss the effect of variation of pressure and superheat on Rankine cycle efficiency with the help of a *T–s* diagram.
5. Explain the process of reheating and its effect on cycle efficiency.
6. Draw a neat sketch of a regenerative power cycle and with the help of a *T–s* diagram analyse the cycle if three feed water regenerators are used.
7. In a Rankine cycle, the turbine inlet pressure is 6 MPa at 500°C. The expansion takes place to 10 kPa with efficiency of expansion 0.8. Determine the cycle efficiency and dryness fraction after expansion. Compare these with the ideal cycle. If the steam is dry saturated at turbine inlet at the same pressure. What is the change in efficiency?
8. (a) Explain working of Rankine cycle and compare it with Carnot cycle.
   (b) Discuss reheat and regenerative cycles.
9. (a) Explain working of Rankine cycle. What is the modified Rankine cycle.
   (b) Explain the advantages of the regenerative cycle over the Rankine cycle.
   (c) Briefly explain the Reheat cycle.
10. (a) Explain why Carnot cycle is not used as a reference cycle for a steam power plant.
    (b) With the help of corresponding flow and *T–s* diagrams explain briefly the working of a practical regenerative Rankine cycle with one open feed water heater.
11. Show a simple Rankine cycle on *p–V* and *T–s* plane using dry saturated steam at the beginning of expansion. Write the expression for its efficiency with and without pump work. Also, draw the schematic diagram of the steam plant for the above and name all the components.

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Steam Generator 5

Contents

5.1 Fossil fuel steam generators
5.2 Classification of boilers
5.3 Circulation in water tube boilers
5.4 Modern high-pressure water tube boilers
5.5 Boiler performance calculations
5.6 Accessories for the steam generator
5.7 Boiler mountings
5.8 Questions

5.1 FOSSIL FUEL STEAM GENERATORS

Steam generator is a closed vessel that is used to generate steam at constant pressure as per the process requirement. The steam generated may be wet, dry saturated or superheated in state. In modern power plants, it is very common to use one boiler (single unit) per turbine, which leads to simpler piping systems and relatively easier boiler and turbine control. These boilers are usually designed to operate either at *critical pressure* (221.2 bar) or above or below the critical pressure. If the steam generators are designed to operate above the critical pressure, then they are known as *supercritical* boilers or *once-through* boilers. Some of the steam generators are designed to operate below the critical pressure, and they are known as *sub-critical* or drum boilers. In a steam generator or boiler, constant pressure is maintained by balancing the rate of steam generated with the rate of steam consumed. In a thermal power station, coal is the main source of combustion. The heat generated by burning coal is utilized to generate steam, which in turn runs the turbo-generator.

5.2 CLASSIFICATION OF BOILERS

5.2.1 Fire Tube Boilers and Water Tube Boilers

Based on the contents of the tubular heating surface, boilers are primarily classified as fire tube boilers and water tube boilers.

1. *Fire tube boilers*

Fire tube boilers are those in which the products of combustion pass through the tubes and water lies around the outside of tubes as shown in Figure 5.1. Examples include Cochran,

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Lancashire, Cornish, locomotive and Scotch marine boilers. These boilers operate at moderate pressure (16–20 bar) and are suitable for generating 3–8 tons of steam per hour, which is used in process heating.

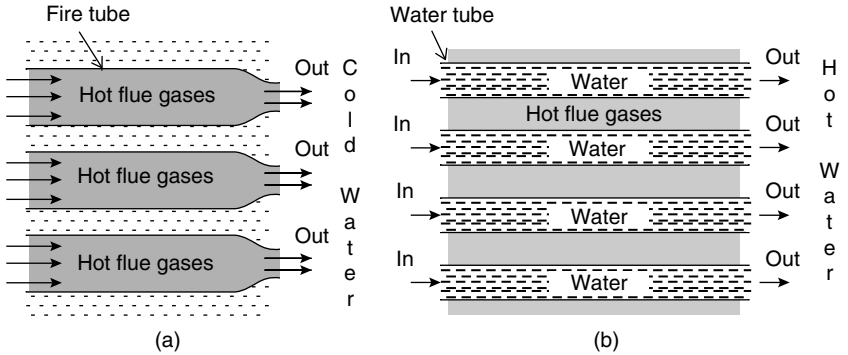


Fig. 5.1 *Comparison of Fire Tubes and Water Tubes: (a) Fire Tube Boiler Tubes; (b) Water Tube Boiler Tubes*

2. *Water tube boilers*

Water tube boilers are those in which the products of combustion (hot flue gases) surround the water tubes from outside. Cold water enters the tubes and leaves hot as shown in Figure 5.2. Water boilers operate at very high pressures and are used for power generation.

Examples: Babcock–Wilcox boiler and *Sterling boiler*

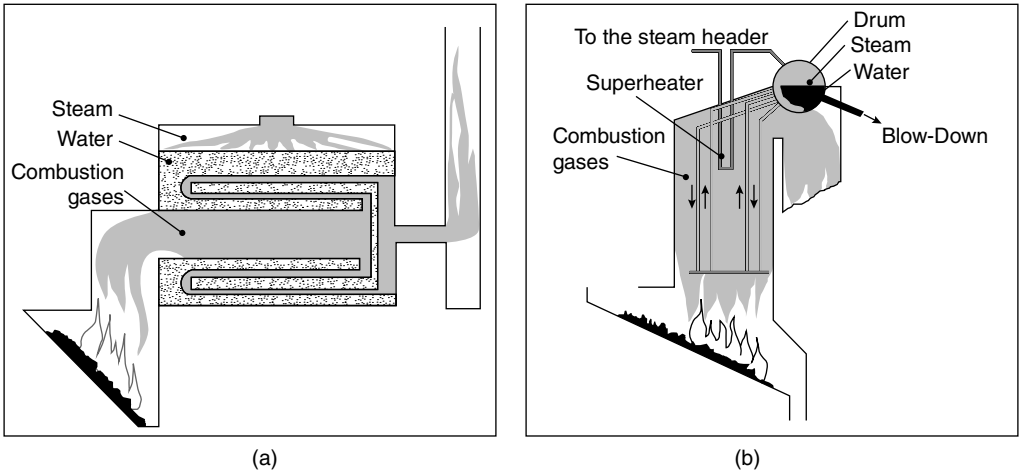


Fig. 5.2 *Principle of Operation of Fire Tube and Water Tube Boilers: (a) Principle of Fire Tube Boiler; (b) Principle of Water Tube Boiler*

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5.2.2 Stationary and Mobile Boilers

If the boilers are used at one place only they are termed stationary boilers. These boilers are used either for process heating in industries or for power generation in steam power plants.

Examples: Babcock–Wilcox boiler and *fluidized bed combustion (FBC) boiler*.

Mobile boilers are portable and are used in locomotives and ships.

Examples: Locomotive boiler and *marine boiler*

5.2.3 Internally Fired and Externally Fired Boilers

If the furnace is placed in the region of boiling water, then the boiler is termed internally fired boiler.

Example: Lancashire boiler

If the furnace is placed outside the boiling water region, then the boiler is known as externally fired boiler.

Example: Babcock–Wilcox boiler

5.2.4 Horizontal, Vertical and Inclined Tube Boilers

If the heating tubes are horizontal, then the boilers are called horizontal tube boilers.

Example: Lancashire boiler

If the heating tubes are vertical, the boilers are known as vertical tube boilers.

Example: Cochran boiler

If the heating tubes are inclined to the horizontal, the boilers are known as inclined tube boilers.

Example: Babcock–Wilcox boiler

5.2.5 Based on Heat Sources

Boilers may be classified based on the fuel used for combustion or heat generation source. Various heat sources used may be the following:

(i) Heat generated by the combustion of fuel in solid, liquid or gaseous forms
(ii) Heat generated by hot waste gases as byproducts of other chemical processes
(iii) Heat generated by electrical energy
(iv) Heat generated by nuclear energy

5.2.6 Natural Circulation and Forced Circulation Boilers

If the circulation of water is by natural convection currents produced by the application of heat, the boilers are known as natural circulation boilers.

Example: Babcock–Wilcox boiler

If the circulation of water is by external means using a pump, the boilers are known as forced circulation boilers. In these types, the fluid is forced *once through* or controlled by partial recirculation. Such boilers are also known as positive forced circulation boilers. These boilers work at very high pressures.

Examples: La-Mont boiler, Velox boiler, Benson boiler, etc

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5.2.7 Comparison of Water Tube Boilers and Fire Tube Boilers

Water tube boilers	Fire tube boilers
Advantages	
The rate of evaporation is more as the area of heating surface is more due to the presence of large number of small diameter tubes.	The rate of evaporation (steam generation rate) is less as the area of heating surface is less due to the presence of small number (usually 2) of large diameter tubes.
Due to smaller ratio of water to steam space (W/S), steam can be generated at a faster rate even if the boiler is cold.	Due to larger ratio of water to steam space (W/S), steam generation is not so fast.
As the rate of evaporation is more, it is preferred in steam power plants for power generation.	As the rate of evaporation is less, it is preferred in process industries.
Working pressure of the boiler is high as the shell can withstand high temperature or thermal stresses.	Working pressure is low as the flue tubes are subject to more thermal stresses for the same thickness of the water tube boiler tubes.
Due to higher working pressure and positive circulation of water, there are fewer chances of sediments getting deposited inside the water tubes in the shell.	Due to lower working pressure, there are more chances of sediments getting deposited.
The rate of heat transfer from the flue gases to the water tubes is more as the water tubes are positioned to obstruct the flow of flue gases (cross flow).	The rate of heat transfer is less as the direction of flow of the flue gases and the water flow is either parallel or counter flow.
Furnace can be altered to suit the type of fuel used for combustion, and hence the boiler can adopt flexible firing methods.	Furnace cannot be altered and is used for firing one fuel only.
Boiler need not be shut down immediately just in case water tube bursts, and hence is not fatal.	Bursting of flue tube results in serious explosion and may be fatal.
Cleaning, repairing and inspection are easy as all parts are accessible.	Cleaning, inspection and repairing are not easy due to inaccessible parts.
Water circulates inside the tubes and hot flue gases surround them.	Hot flue gases circulate inside the tubes and water surrounds the boiler shell.
Boiler furnace is placed outside the boiler, furnace alteration is easy.	Boiler furnace is placed inside the boiler, hence furnace alteration is difficult.
Water circulation is cyclic, that is, from boiler drum to water tubes and again to boiler drum.	Water circulation is limited inside the boiler shell.
Disadvantages	
Water should be treated in a systematic manner to remove impurities present in it. A water treatment plant is used to avoid silica deposits.	Water need not be that much pure and needs minimum treatment such as passing through a pressure filter.
It has high initial cost.	It has low initial cost.
Not used for mobile purpose.	Used as mobile boiler.

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5.3 CIRCULATION IN WATER TUBE BOILERS

If the circulation of water is by natural convection currents produced by the application of heat, the boilers are known as natural circulation boilers.

Example: Babcock–Wilcox boiler

If the circulation of water is by external means using a pump, the boilers are known as forced circulation boilers. In these types, the fluid is forced 'once through' or controlled by partial recirculation. These boilers are also known as positive forced circulation boilers. These boilers work at very high pressures.

Examples: La-Mont boiler, Velox boiler, Benson boiler, etc.

Modern-day power plants use forced circulation boilers that have the high steam-raising capacity. In the sections that follow, few such boilers are discussed.

5.4 MODERN HIGH-PRESSURE WATER TUBE BOILERS

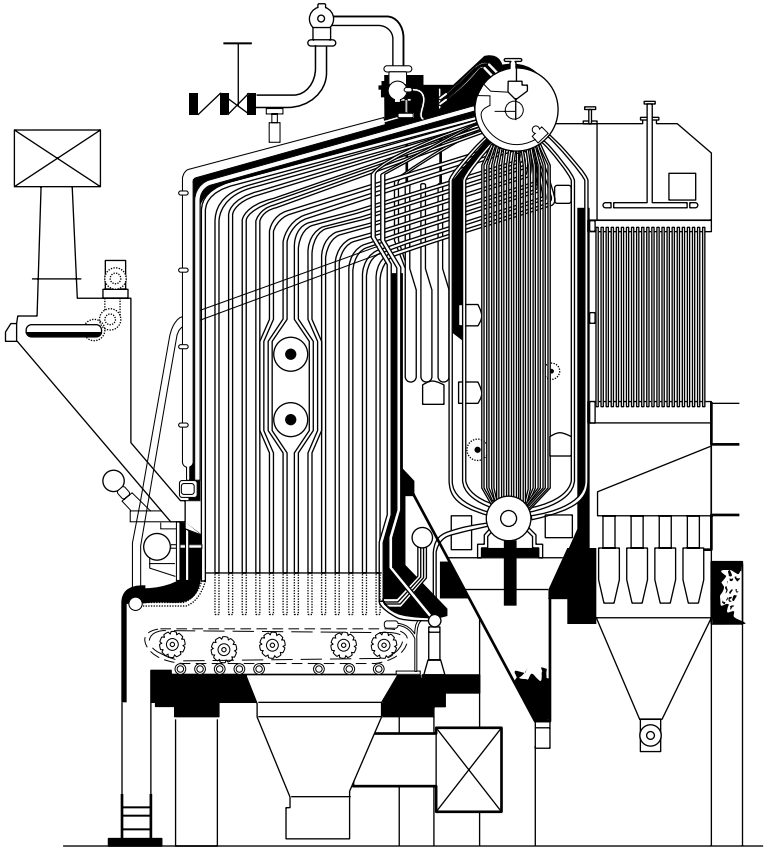


Fig. 5.3 *A Typical Modern High-Pressure Boiler*

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In modern power plants, it is very common to use one boiler (single unit) per turbine, which leads to simpler piping systems and relatively easier boiler and turbine control. These boilers are generally designed to operate at *critical pressure* (221.2 bar) or above or below the critical pressure. If the boilers are designed to operate above the critical pressure, then they are known as *supercritical* boilers or *once-through* boilers. If the boilers are designed to operate below the critical pressure, then they are known as *sub-critical* or drum boilers (Figure 5.3).

5.4.1 Generation of Steam Using Forced Circulation, High and Supercritical Pressures

High-pressure boilers may further be classified into natural circulation, forced circulation and once-through boilers.

5.4.1.1 Natural Circulation Boilers

A typical flow pattern of a natural circulation boiler is shown in Figure 5.4. Here, water is circulated purely by density difference with most of the heat from the fuel flame being radiated to the water walls directly. The steam pressure of such boilers is limited to about 180 bar, with water and steam being separated in the boiler drum.

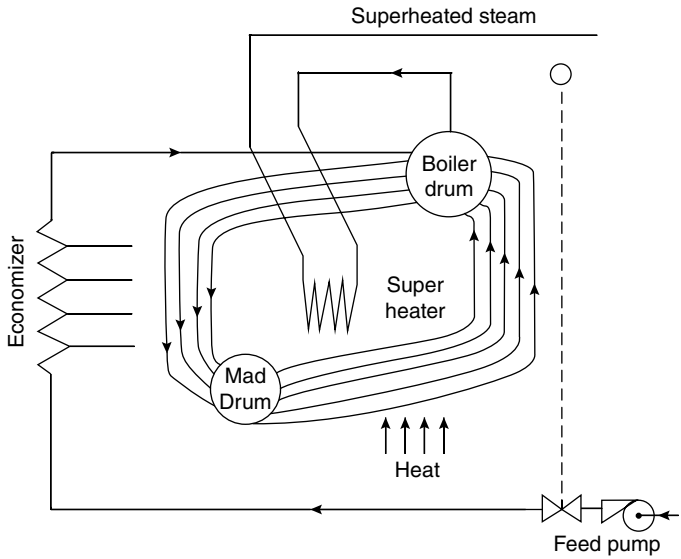


Fig. 5.4 *A Typical Natural Circulation Boiler*

5.4.1.2 Forced Circulation Boilers

Figure 5.5 shows the flow pattern of a forced circulation boiler. In these boilers water is circulated by using an additional pump. These boilers often use orifices, which control the flow circulation. Orifices are located at the bottom of the tubes that ensure even distribution of flow through water wall tubes. These boilers can generate steam up to about 200 bar.

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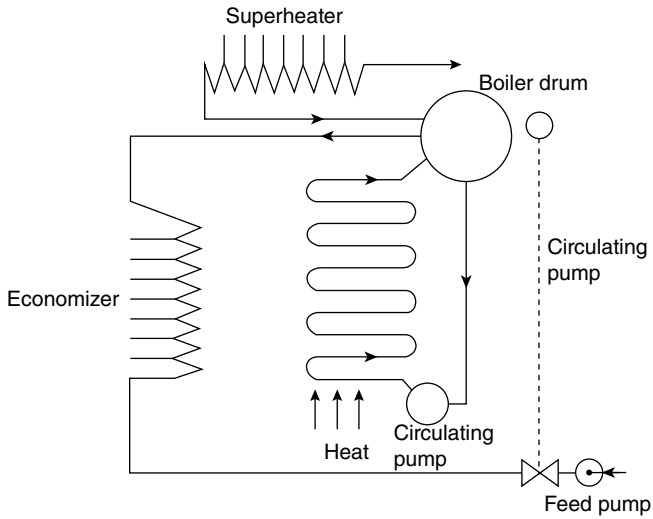


Fig. 5.5 *A Typical Forced Circulation Boiler*

5.4.1.3 Once-Through Boilers

Figure 5.6 shows a flow diagram of a *once-through boiler*. These boilers operate above critical pressure, that is, above 221.2 bar. As the density of water and steam is same above critical pressure, there will be no recirculation. In these boilers, water enters the bottom of the tubes and completely transforms into steam as it passes through the tubes and reaches at the top. Thus, these boilers do not need a steam drum, and are hence often referred to as *drumless boilers.*

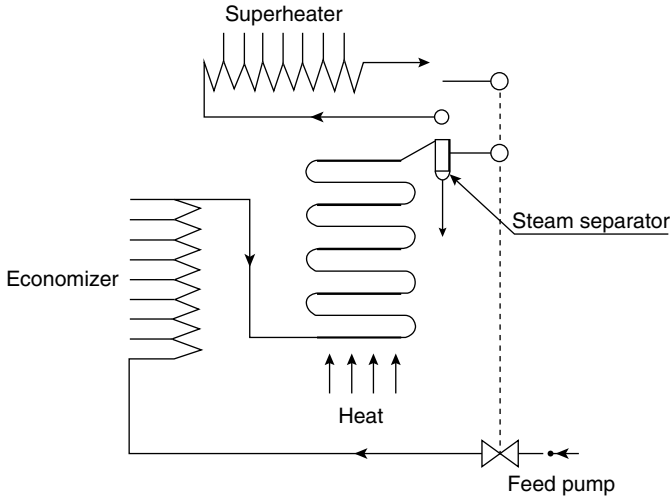


Fig. 5.6 *Flow Diagram of a Once-Through Boiler*

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5.4.2 A Brief Account of Modern Steam Generators

In the section that follow, a brief explanation of few high-pressure boilers is given.

5.4.2.1 La-Mont Boiler

Figure 5.7 shows a modern forced circulation type La-Mont boiler.

Feed water from the hot well is passed through the economizer tubes before entering the steam and water drum. During its flow through the economizer tubes, water gains maximum sensible heat from the flue gases escaping to the exhaust. As the boiler operates at pressures ranging between 120 bar and 160 bar, natural circulation is limited due to reduced density difference between the liquid and vapour. Hence, an external centrifugal pump is used to assist the circulation. This pump delivers large amount of water up to 10 times the rate of evaporation from the steam drum. Water from the *circulating pump* is forced through the *evaporator* tubes placed in the *radiant section* and *convective section* of the furnace simultaneously. This is done by the *distributing header* that distributes water from the circulating pump to the evaporator tubes through *nozzles*. The steam generated in the steam drum is then passed through the superheater tubes before being fed to the turbo generator. This boiler is useful to supply steam between 130 and 3000 tons/h.

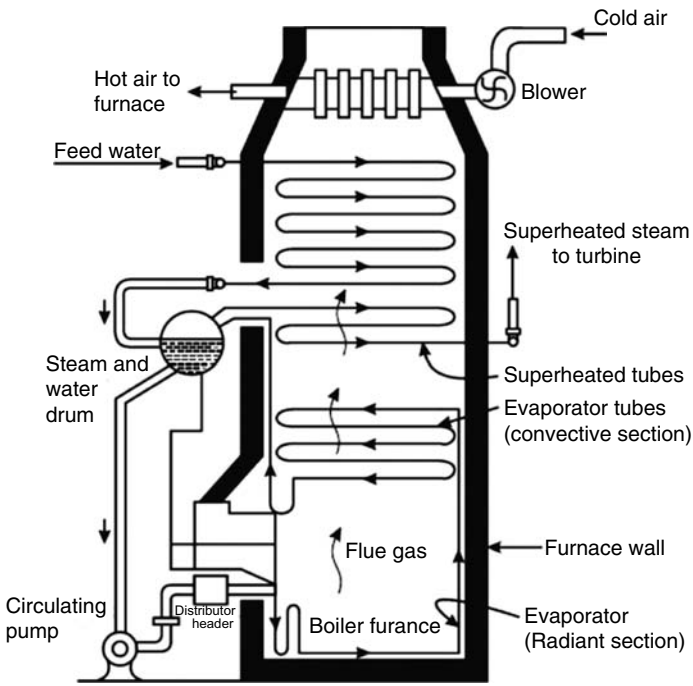


Fig. 5.7 *A Modern Forced Circulation Type La Mont Boiler*

The main disadvantage of La-Mont type boiler is the formation of bubbles. These bubbles come in contact with the inner surface of the heating tubes and are subsequently attached to it. The bubbles attached to the tube surface have higher thermal resistance, and consequently reduce the heat flow and steam generation.

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5.4.2.2 Benson Boiler

Figure 5.8 shows a typical Benson boiler. It is a supercritical boiler with pressure ranging from 225 bar to 500 bar. The difficulty of bubble formation is completely eliminated due to the operation of the boiler above the critical pressure.

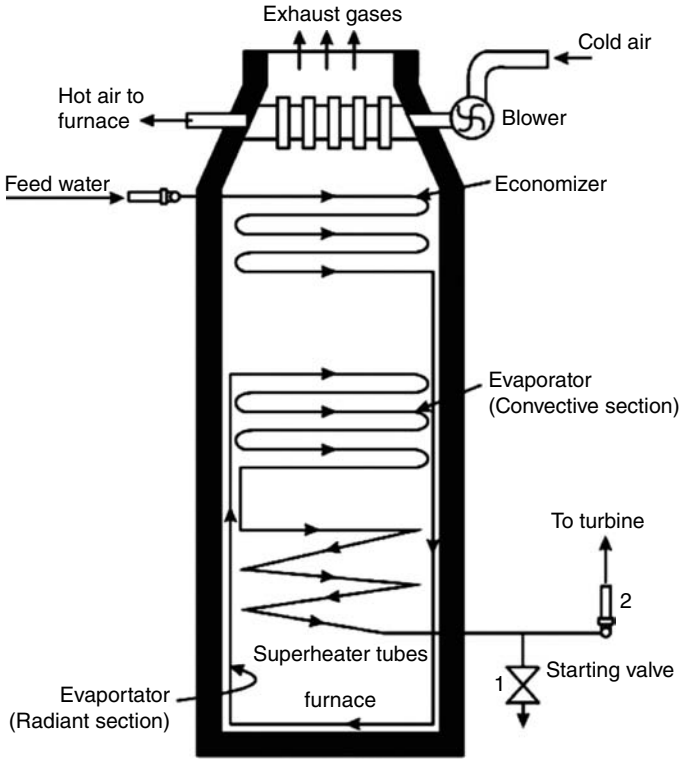


Fig. 5.8 *Benson Boiler*

Feed water from the hot well is forced through the economizer tubes. The hot water enters the *radiant section* of the evaporator tubes and then passes through *convective section* of the evaporator tubes. Since the boiler has no drum, the steam generated in the evaporator tubes enters the superheater tubes where it is heated above the critical pressure. Steam-raising capacity of the boiler is around 150 tonnes/h and above.

The major disadvantage of the boiler is that when water transforms into steam in the convective section of the evaporator, salts get deposited in the transformation zone. Hence, periodic flashing of evaporator tubes in the convective section becomes necessary.

The boiler is started from cold by circulating the feed water from the hot well by operating the *starting valve*. During this period, the valve (valve 2) that supplies superheated steam to the turbine is closed. Thus, water circulates through economizer, evaporative tubes, superheater and back to the feed water circuit via the starting valve. While taking the boiler on range, starting valve is closed and valve 2 is opened. This method avoids excessive heating of the tubes. The advantages of Benson boiler are the following:

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1. It is compact is size.
2. It is lesser in weight due to the absence of drum.
3. Boiler erection is easy as all parts are welded at the plant site.
4. It is suitable for both partial loads and overloads.
5. Formation of bubbles is eliminated due to supercritical pressure.
6. It can withstand sudden load fluctuations; therefore, it is more suitable for power stations.
7. It has negligible blow-down losses compared with natural circulation boiler.
8. It is prone to lesser explosion hazards due to its small storage capacity.

5.4.2.3 Velox Boiler

Figure 5.9 shows a line diagram of fire tube Velox boiler. This is a fire tube boiler that uses oil or gaseous fuel. The combustion gases are circulated through the tubes with supersonic velocity to increase the heat transfer between the hot gases and the feed water. Air compressed to 2.5 bar by an air compressor is supplied to the combustion chamber. The combustion gases produced by the combustion of fuel pass through the tubes transferring heat to the water. The mixture of water and steam formed enters a steam separator with a spiral flow. The centrifugal force produced by this effect causes the heavier water particles to be thrown outword to the walls. The steam thus separated is passed to superheater where it is superheated by the gases coming from the combustion chamber and finally passed to prime mover. The water is again sent to the combustion chamber through a circulating pump. The combustion gases coming out of the superheater are used to run a gas turbine that runs the air compressor. The exhaust gases coming out of the gas turbine is passed through the economizer to heat the feed water. The feed water is circulated through the tubes with the help of a pump.

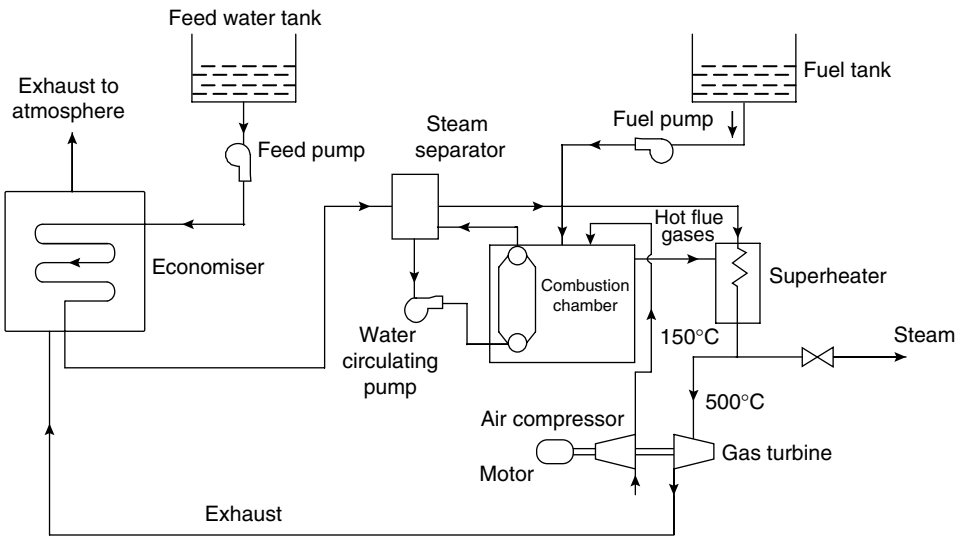


Fig. 5.9 *Velox Boiler*

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5.4.2.4 Schmidt–Hartmann Boiler

Figure 5.10 shows a Schmidt–Hartman boiler. There are two circuits in this boiler. In the primary circuit, distilled water from the water passes through the primary evaporator located in the combustion chamber of the boiler. The steam produced in the evaporator is passed through a submerged heating coil located in the evaporator drum. The high-pressure condensate formed in the submerged heating coil is circulated through a low-pressure feed heater where its natural circulation is maintained in the primary circuit. In the secondary circuit, feed pump supplies impure feed water from hot well to the evaporator drum through feed preheater. The saturated water that comes from the feed preheater is heated by the high-pressure steam in the heating coil. The steam thus produced in the evaporator drum from impure water is further passed through the superheater and then supplied to prime mover.

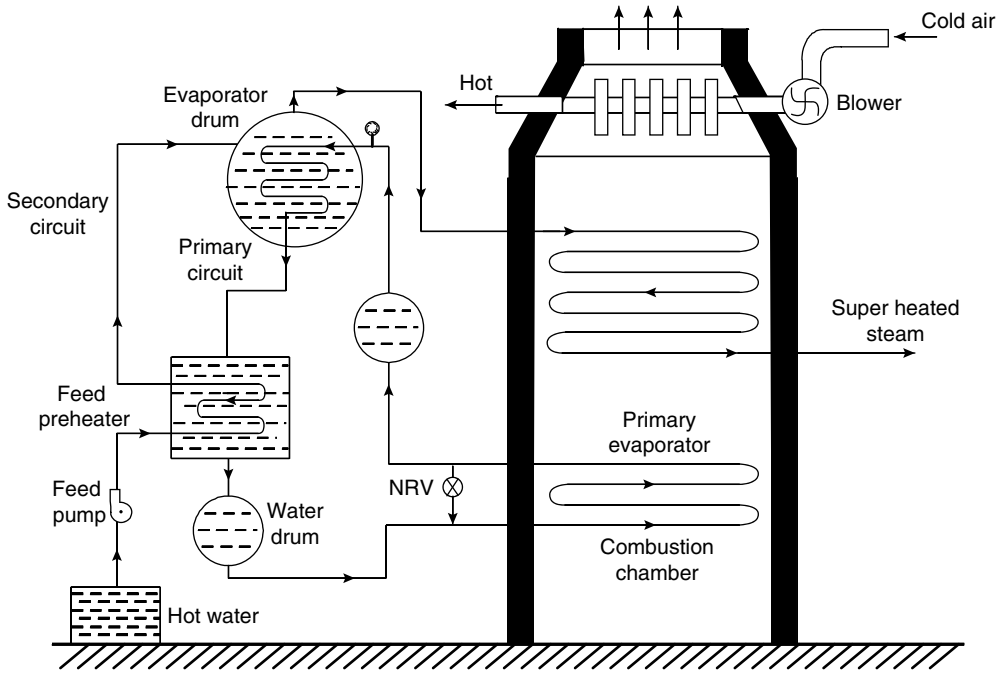


Fig. 5.10 *Schmidt–Hartman Boiler*

5.4.2.5 Loeffer Boiler

Figure 5.11 shows a Loeffer boiler. In this boiler, the feed water is evaporated by superheated steam. The high-pressure feed pump forces the water through the economizer and delivers it to the evaporator drum. The steam-circulating pump draws saturated steam from this drum and passes it through the superheater. About one-third of superheated steam passes to the turbine, and the remaining steam enters the evaporating drum where it evaporates the feed water.

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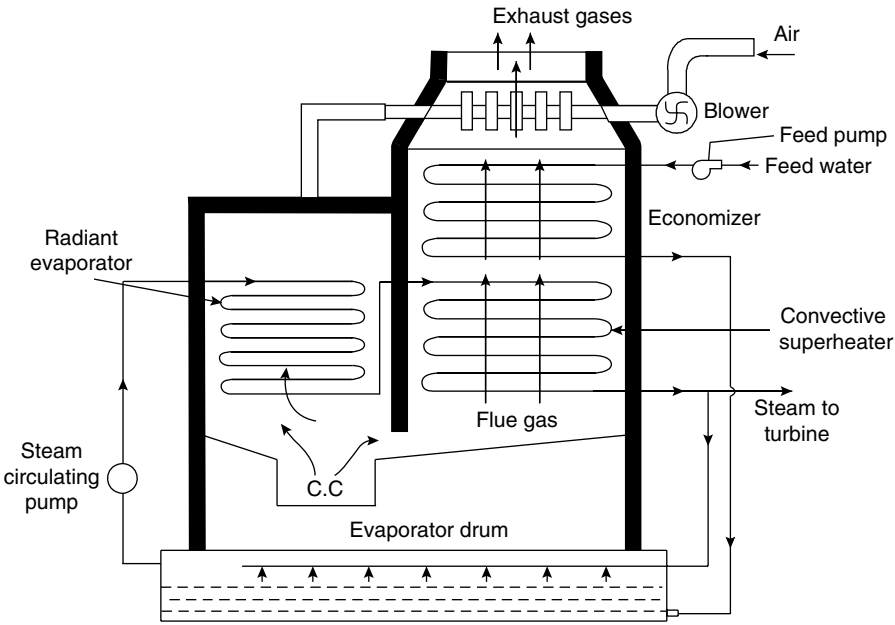


Fig. 5.11 *Loeffer Boiler*

5.4.2.6 Supercritical Boilers

The increasing fuel costs with decreasing fuel quality have constantly persuaded power engineers to search for more economical methods of power generation. The most recent method to produce economical thermal power is by the use of supercritical steam cycle. Between the working ranges of 125 bar and 510°C to 300 bar and 600°C, large number of steam-generating units are designed, which are basically characterized as sub-critical and supercritical. Usually, a sub-critical boiler consists of three distinct sections: preheater (economizer), evaporator and superheater. In the case of supercritical boiler, only preheater and superheater are required. The constructional layouts of both types of boilers are otherwise practically identical.

With the recent experiences gained in design and construction of supercritical boilers, it has become a rule to use supercritical boilers above 300 MW capacity units.

The advantages of supercritical boilers over critical type are listed below:

1. They offer higher heat transfer rates as compared with sub-critical boilers. In a subcritical boiler, the steam side heat transfer coefficient is 165 MJ/m_2h-°C at respective steam pressure and temperatures of 180 bar and 538°C, whereas the steam side heat transfer, coefficient for supercritical boiler is 220 MJ/m_2–h-°C when the steam is generated at 240°C.
2. The pressure level is more stable due to less heat capacity of the generator and therefore gives better response.
3. Higher thermal efficiency (40–42 per cent) of power station can be achieved with the use of supercritical steam.

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4. The problems of erosion and corrosion are minimized in supercritical boilers as two-phase mixture does not exist.
5. The turbo generators connected to supercritical boilers can generate peak loads by changing the pressure of operation.
6. There is a great case of operation, and their comparative simplicity and flexibility make them adaptable to load fluctuations.

Although thermodynamically steam temperature and pressure are always desirable, the trend is halted due to the availability of material and difficulties experienced in the turbine and condenser operations due to large volumes. Presently, 246 bar and 538°C are used for unit sizes above 500 MW capacity plants.

Boilers operating above critical pressure are known as supercritical boilers. They are used in power plants of capacity 300 MW and above. The constructional layout of supercritical boiler is practically identical to subcritical boiler. Usually, a sub-critical boiler consists of three distinct sections: economizer, evaporators and superheater; no distinction can be made between such components in supercritical boiler. Figure 5.12 shows Ramsin's once-through boiler.

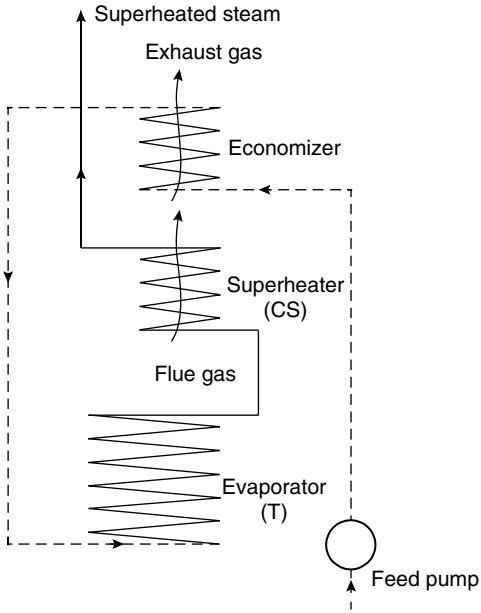


Fig. 5.12 *Ramsin Boiler*

The boiler consists of inclined evaporator coils (T) arranged in a spiral. Forty such coils are paralleled around the furnace. Steam generated in the evaporators flows into the headers and then to the convection superheater (CS). The superheated steam is utilized for power generation.

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Advantages

1. Heat transfer rates are considerably large.
2. Higher thermal efficiency of power plant.
3. Problems of erosion and corrosion are minimized as two-phase mixture does not exist.
4. Adaptable to load fluctuation.

Disadvantages

1. It is costly due to increased requirement of steel for heat transfer surface, pumps and feed water piping.

5.5 BOILER PERFORMANCE CALCULATIONS

Overall efficiency of the plant is defined by the ratio of plant output expressed as follows:

$$\eta_0 = \frac{\text{Output}}{\text{Input}} \times 100$$

$$= \frac{\text{Energy generated}}{\text{Heat supplied}}$$

$$\eta_0 = \frac{\text{Energy generated (KW)}}{m_f \times CV_f}$$

where m_f = mass of fuel burnt/s, kg/s, CV_f = calorific value of fuel, kJ/kg.
Considering the Rankine cycle, we can define both boiler and tubine efficiencies as follows:
Boiler efficiency,

$$\eta_b = \frac{\text{output}}{\text{Input}}$$

$$= \frac{\text{Heat supplied to steam in raising its temperature to superheat condition}}{\text{Heat input to boiler}}$$

$$= \frac{\text{Total heat of feed water}}{m_f \times CV_f} \times 100$$

$$\eta_b = \frac{h_1 - h_w}{m_f \times CV_f} \times 100$$

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where h_1 = enthalpy of superheated steam, kJ/kg
h_w = enthalpy of feed water, kJ/kg
m_f = mass of fuel burnt, kg
CV_f = calorific value of fuel, kJ/kg

5.5.1 Turbine Efficiency

Turbine efficiency is defined as the ratio of mechanical work output to isentropic heat drop in the turbine.

$$\eta_t = \frac{\text{Mechanical work output}}{\text{Isentropic heat drop}} \times 100$$

$$= \frac{\text{Mechanical work output}}{(h_1 - h_{2'})} \times 100$$

where $h_{2'}$ = enthalpy of exit steam, kJ/kg.

Due to heat rejection (45–49 per cent of total available heat energy) in the condenser, Rankine cycle efficiency decreases.

The following Figure 5.13 shows the heat balance diagram considering the losses.

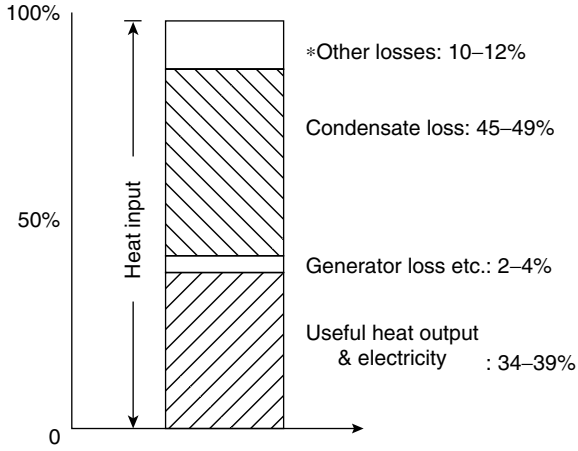


Fig. 5.13 *Heat Balance Diagram*

Note: Other losses include the following:
Moisture in fuel: 0.5–1.0%
Combustion of hydrogen: 3%
Dry chimney gasses: 4.5–5.5%
Combustible in ash: 1–2%
Radiation and unaccounted: 1–2%

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5.5.2 Rankine Cycle Efficiency (η_c)

Rankine cycle efficiency is given by the ratio of energy available for conversion in work to energy given as heat in boiler.

$$\therefore \eta_c = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

$$= \frac{h_1 - h_2}{h_1 - h_{2'}}$$

where h_2 = actual enthalpy of steam at turbine exit.

5.5.3 Generator Efficiency (η_g)

During conversion of mechanical energy into electrical energy, some losses occur due to friction and core losses (copper loss, iron loss).

Generator efficiency is given by the following equation:

$$\eta_g = \frac{\text{Electrical energy sent out, kwh}}{\text{Mechanical work (kw)} \times 3600} \times 100$$

Based on the above four efficiencies, the overall efficiency of the plant may be defined as follows:

$$\eta_0 = \eta_b \times \eta_t \times \eta_c \times \eta_g$$

A part from the above two more terms namely overall turbo-alternator efficiency and heat rate are quite useful for calculation purpose.

5.5.4 Overall Turbo-Alternator Efficiency (η_{ota})

Overall turbo-alternator efficiency is defined as the ratio of electrical energy sent out to the heat supplied in the boiler and is expressed as follows:

$$\eta_{ota} = \frac{\text{Electrical energy sent out}}{\text{Heat supplied to steam in boiler}} \times 100$$

$$= \eta_t \times \eta_c \times \eta_g$$

Hence, overall station efficiency is expressed as follow:

$$\eta_0 = \eta_b \times \eta_{ota}$$

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5.5.5 Heat Rate

Based on heat rate, it is sometimes convenient to express overall turbo-alternator efficiency.

Heat rate is expressed as follows:

$$\text{HR} = \frac{\text{Heat added to steam in boiler}}{\text{Electrical energy sent out}}$$

$$\therefore \eta_{\text{ota}} = \frac{1}{\text{HR}} \times 100$$

In addition, the overall station or plant efficiency is

$$\eta_0 = \frac{1}{\text{HR of plant}} \times 100$$

Note: When the plant has multiple turbines fed by a common header or bus from the boilers, the overall efficiency must be multiplied by a factor known as range factor, or range efficiency factor.

5.5.6 Boiler Performance

Boiler efficiency depends on the ability to burn the fuel and subsequently utilize the heat energy to transform water into steam; in general, a lager boiler has higher efficiency in comparison with a smaller one, due to its physical size and ability to operate at higher pressure and temperature.

Based on the energy consumption of auxiliaries and the type of the fuel chosen, boiler efficiency can be expressed as follows:

1. *Gross-on-gross efficiency*

Gross-on-gross efficiency is based on the gross calorific value or GCV (higher calorific value, HCV) of the fuel and the gross heat supplied to the working fluid.

2. *Net on gross efficiency*

Net on gross efficiency calculation is based on the GCV of the fuel and the net supplied to the working fluid after excluding heat equivalent energy consumption from boiler auxiliaries.

3. *Gross on net efficiency*

Gross on net efficiency calculation is based on net calorific value of the fuel (LCV) and gross heat supplied to the working fluid.

4. *Net-on-net efficiency*

Net-on-net efficiency calculations are based on the net calorific value (LCV) of the fuel and the net heat supplied to the fuel. Two different method are used for computing the boiler efficiency.

These are as follows:

(i) Direct method

(ii) Indirect or losses method

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(i) *Direct method*

In direct method, efficiency is calculated for a non- reheat (NRH) unit as follows:

$$h = m_s \frac{(\text{Enthalpy of steam} - \text{Enthalpy of feed water})}{\text{Heat supplied}}$$

$$\eta_b = m_p \times \left[\frac{C_{PS}(T_1 - T_S) + h_{fg} + C_{pw}(T_S - T_W)}{m_f \times CV_f} \right]$$

where

m_β = steam flow rate
$C_{P\beta}$ = specific heat of steam
T_1 = superheat steam temperature
T_S = saturation steam temperature
h_{fg} = latent heat of vapourization
T_w = inlet temperature of feed water
m_f = mass of fuel burnt
CV_f = calorific value of fuel, KJ/Kg

With the advances in measurement techniques, this method is gaining popularity over again.

(ii) *Indirect or losses method*

In the indirect method, by knowing the losses, one can estimate the efficiency.

Thus

Efficiency = 100% – Losses

The advantage of this method is that even if some errors are present in measurement, it will not alter the efficiency drastically.

5.6 ACCESSORIES FOR THE STEAM GENERATOR

Boiler accessories are the appliances that ensure the improved efficiency of a boiler. Boiler accessories may be installed either inside or outside a boiler.

Most commonly used boiler accessories are the following:

1. Economizer
2. Superheater
3. Air preheater
4. Feed water pump
5. Pressure-reducing valve
6. Steam trap
7. Steam separator

The function of all the accessories is briefly discussed below.

1. *Economizer*

The function of the economizer is to recover a portion of heat of the exhaust gases before the flue gases enter the chimney and discharged to the atmosphere. The economizer is placed in the

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path of the flue gases in between the boiler exit and entry to the chimney. Feed water coming from the feed pump when passed through the economizer tubes absorbs the heat in the exhaust gases. This increases the temperature of water entering the boiler. Due to the high temperature of feed water, fuel consumption reduces, and this increases the overall efficiency of the boiler.

2. *Superheater*

The function of the superheater is to increase the temperature of steam above its saturation temperature. As heat contained in unit mass of superheated steam is more than dry saturated or wet steam, it is used extensively in steam power plants. Steam from the boiler drum is passed through superheater tubes. Superheater tubes are placed in the furnace along the passage of flue gases. Temperature of steam is thus raised above the saturation temperature.

3. *Air preheater*

The function of the air heater is to recover the heat of a portion of exhaust flue gases before the flue gases enter the chimney. It is placed along the passage of the exhaust flue gases in between the economizer and the chimney.

Air from the forced draught fan is passed over the preheater tubes that contain flue gases. Temperature of air is increased, and this high-temperature air enters the furnace. Due to higher air temperature, combustion of the fuel becomes rapid and fuel consumption becomes less. This increases the overall efficiency of the boiler. Two types of air heaters are generally used.

(i) *Tubular or recuperative air preheater*

This type of air preheater is composed of steel tubes through which hot flue gases flow. Air is made to circulate over these steel tubes and thus gains heat.

(ii) *Regenerative air preheater*

This type of air preheater consists of a rotor that turns at about 2–3 rpm. The rotor is filled with thin corrugated metal elements. Hot gases pass through one-half of the heater and air through the other half. As the rotor turns, the heat storage elements transfer the heat absorbed from the hot gases to the incoming air.

4. *Feed water pump*

The function of the feed pump is to pump water at high pressure to the water space of the boiler drum. Many types of feed pumps are used.

(i) *Rotary pumps*

They are either driven by electric motors or small steam turbines. Water is pumped due to rotary action of the impeller.

(ii) *Reciprocating pumps*

They are continuously run by steam from the same boiler to which the water is fed. Water is pumped by reciprocating action.

5. *Pressure-reducing valve*

The function of the pressure-reducing valve is to maintain constant pressure on the delivery side of the valve with the fluctuating boiler pressure. Whenever the steam demand is fluctuating, it becomes very difficult to maintain uniform pressure. In such cases, a pressure-reducing valve is connected to the steam supply line.

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6. *Steam trap*

The function of steam trap is to drain off water resulting from the partial condensation of steam in the steam pipe lines and jackets without allowing the steam to escape through it. Water collected in the steam pipe lines results in hammering, thereby damaging the pipelines and joints. Two types of steam traps are generally used: (i) bucket type or float type and (ii) thermal expansion type.

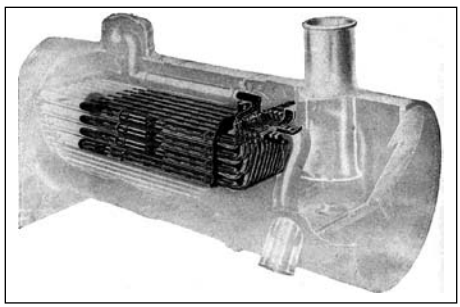
7. *Steam separator*

The function of the steam separator is to separate the water particles in suspension that are carried by the steam coming from the boiler. If suspended water particles enter the turbine or engine, they cause erosion and corrosion of blades and other parts. It is always installed as close to the engine or turbine as possible on the main pipeline.

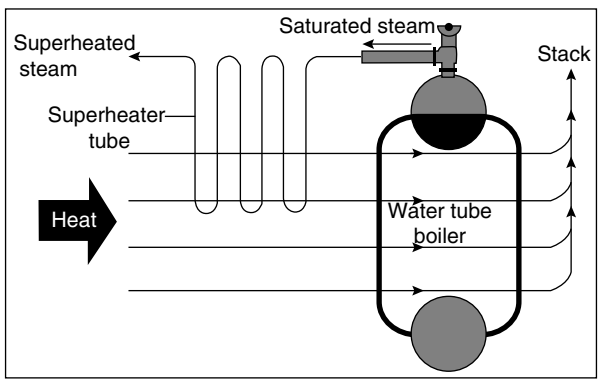
5.6.1 Superheaters Arrangement

5.6.1.1 Superheaters

The functions of the superheater are to remove the last traces of moisture from the saturated steam coming out of boiler and to increase its temperature above saturation temperature. Superheating increases overall cycle efficiency as well as avoids too much condensation in the last stages of the



Superheater arrangement



Superheater

Fig. 5.14 *Superheater Arrangement*

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turbine, which avoids the blade corrosion. Superheater is made of coils of tubes forming parallel tube circuits connected between heaters. The superheater tubes are made of high-temperature strength special alloy steels such of chromium molybdenum. The coils are heated by the heat of combustion gas during their passage from the furnaces to the chimney (Figure 5.14).

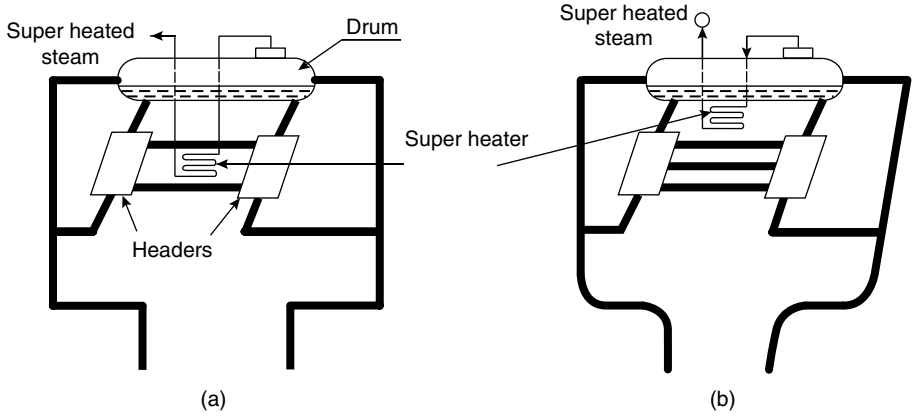


Fig. 5.15 *Superheater Pattern: (a) Interdeck Superheater; (b) Overdeck Superheater*

The superheaters are classified according to the system of heat transfer from the flue gas to the steam. When the superheater is located in the furnace between the furnace water walls, it absorbs heat from the burning fuel through radiation and is known as radiant superheater. A superheater that receives its heat entirely from the gases through convective heat transfer is called convective superheater. Since the radiant superheaters are liable to receive much heat from the furnace and thus get overheated, there is a necessity to design and construct them carefully. Convection superheater is more conveniently located. It may be set as interdeck form or overdeck form. The superheater is placed between the water tubes in interdeck arrangement, and is located above the water tubes in case of overdeck arrangement (Figure 5.15).

5.6.2 Control of Superheaters

Accurate steam temperature control is necessary for avoiding the overheating of superheater tubes and turbine front stages and to maintain overall efficiency as high as possible. This can be done by using a combined convective and radiation superheater. The radiant part of the superheater is kept in the furnace wall, and the convective port is located in the tube bank (Figure 5.16).

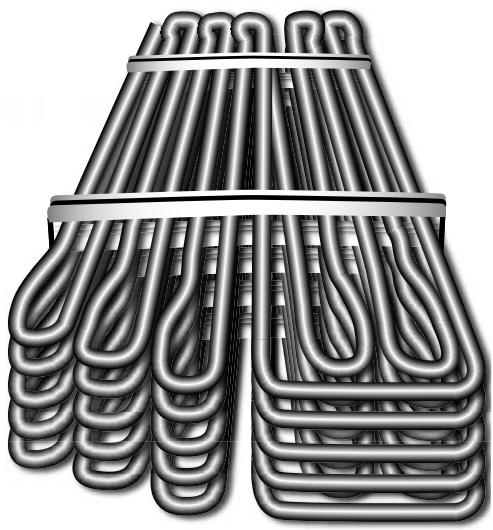


Fig. 5.16 *Photographic View of Superheater Tubes*

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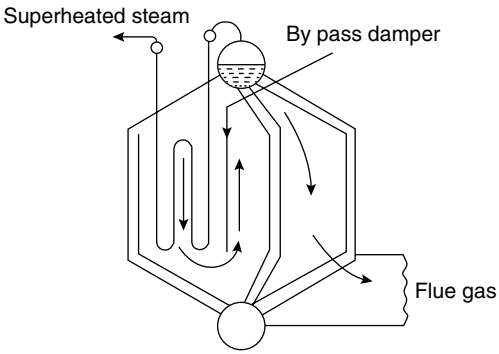


Fig. 5.17 *By Passing Furnace Gas*

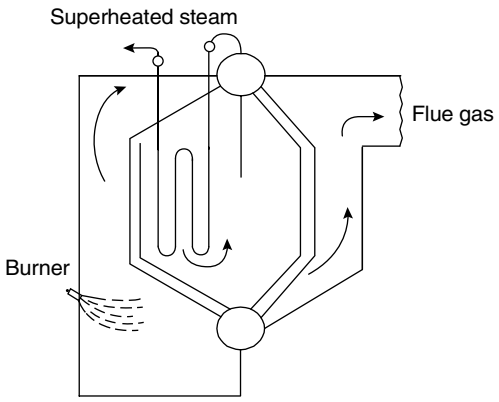


Fig. 5.18 *Tilting Burners in Furnace*

The common methods used for controlling the superheat temperature are given below.

5.6.2.1 Bypassing the Furnace Gases

Part of the furnace gases around the superheater is bypassed with the help of damper especially at lower loads. This method of control is limited because the material of damper cannot withstand stress and high temperature in the gas passage (Figure 5.17).

5.6.2.2 Tilting Burners in the Furnace

The burners are tilted up or down through a range of 30°. When burners are tilted downwards, much of heat is given to the water walls by the gas, and the gas entering the superheater region is relatively cool. When burners are tilted upwards, then heat given to boiler water wall is less, and hotter gas enters the superheater region to increase the steam temperature (Figure 5.18).

5.6.2.3 Auxiliary Burners

The gas temperature can be controlled by turning on and off the auxiliary burners provided in addition to the main burner as shown in Figure 5.19.

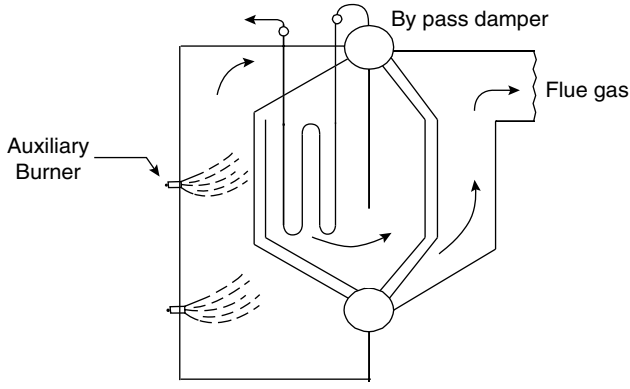


Fig. 5.19 *Auxiliary Burners*

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5.6.2.4 De-Superheater Using Water Spray

The temperature of the steam can be controlled by injecting the water either before the superheater or between sections of superheater. In Figure 5.20, superheated steam coming from the superheater is passed through de-superheater tubes, where water is injected through the nozzles. The de-superheated steam is further taken for power generation.

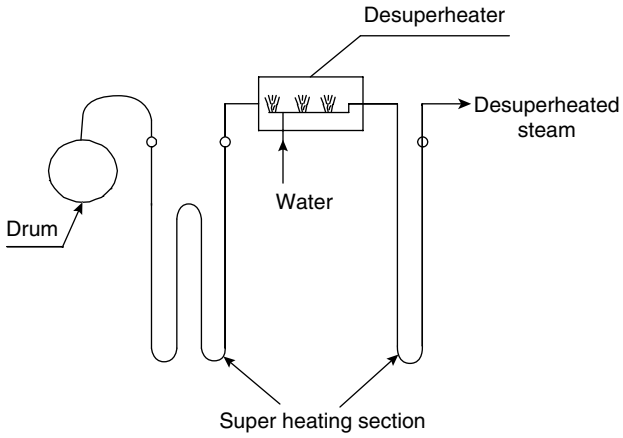


Fig. 5.20 *De-Superheater*

5.6.2.5 Precondensing Control

The temperature of the steam can be controlled by condensing the steam coming out of the boiler using a small condenser with the help of feed water. Here, feed regulation is done automatically. A portion of feed water from the feed pump is directed through the condenser tubes. Excess heat from the steam gets transferred to the incoming feed water, as shown in Figure 5.21.

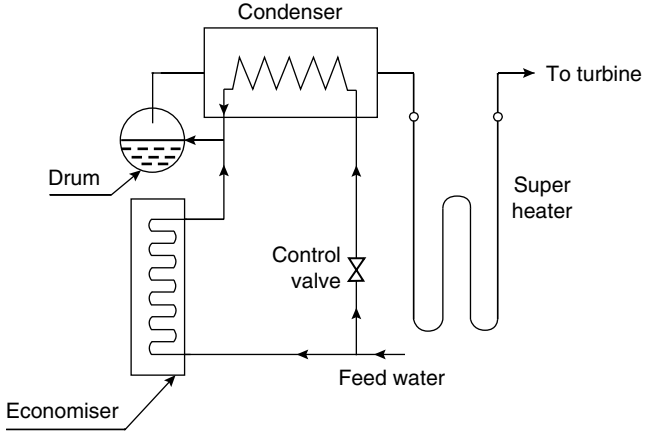


Fig. 5.21 *Pre Condensing Control*

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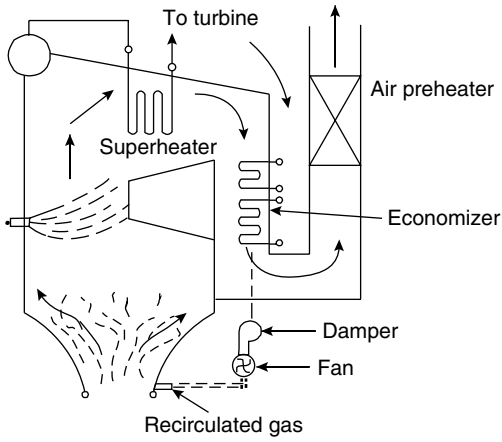


Fig. 5.22 *Gas Circulation*

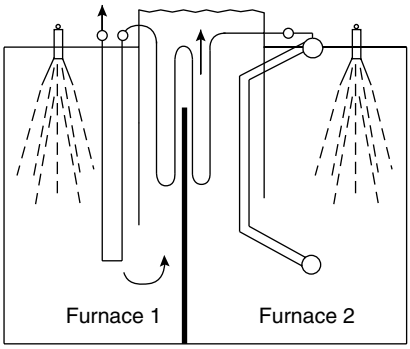


Fig. 5.23 *Twin Furnace Arrangement*

5.6.2.6 Gas Circulation

The gas coming out of economizer is partly re-circulated into the furnace with the help of a fan. This acts like excess air and shields the furnace wall reducing the heat absorption by water wall and increasing the heat absorption by the superheater (Figure 5.22).

5.6.2.7 Twin Furnace Arrangement

Varying the firing rates between furnaces controls the superheat temperature. Figure 5.23 shows a twin furnace arrangement in which firing rate is varied to control the superheat temperature. The degree of superheat is controlled by initiating additional firing in furnace 1, as steam passes inside the superheater tubes.

5.6.2.8 Coil Immersion in the Boiler Drum

A portion of the steam from low temperature section of the superheater is bypassed to a coil immersed in the lower drum of boiler under the control of a bypass valve. The control valve is activated by the final steam temperature. The de-superheated steam in the boiler drum is returned and mixed with the non-de-superheated steam in a junction header, and final superheating takes place in the second stages of superheater (Figure 5.24).

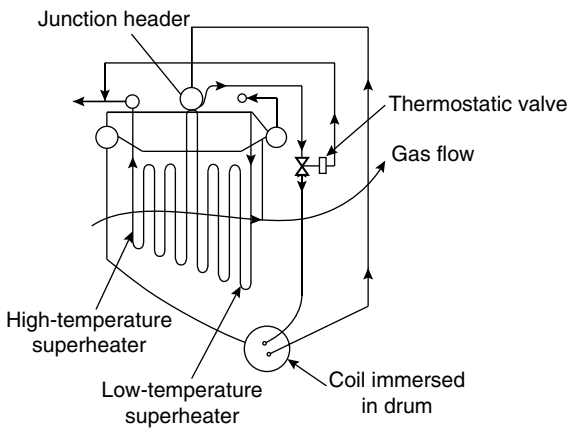


Fig. 5.24 *Coil Immersion in the Boiler Drum*

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5.6.3 Reheaters

The function of reheaters is to re-superheat the partly expanded steam from the turbine so that the steam remains dry as far as possible through the last stage of the turbine. Modern plants have reheaters as well as super-heaters in the same gas passage of the boiler. Reheaters can also be of combination type using both radiant and convective heating.

5.6.3.1 Methods of Reheating

Basically, there are three methods of reheating:

1. *Gas reheating*

In this method, the exhaust steam from the high-pressure turbine is reheated to the initial throttle temperature. However, long pipe connections are needed for this purpose, resulting in piping joints in addition to provision for making expansion and contraction of the pipe joint. After reheating, the steam is fed to the low-pressure turbine for further expansion and to do useful work (Figure 5.25).

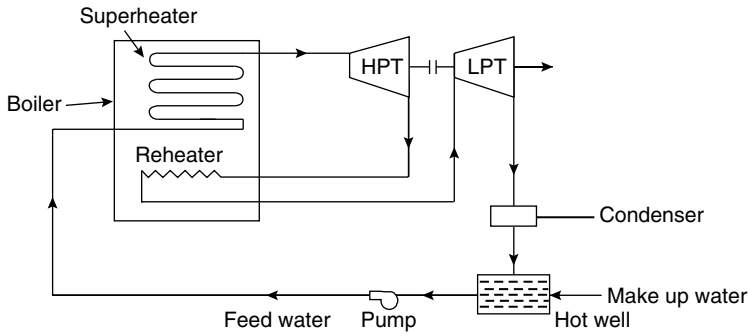


Fig. 5.25 *Gas Reheating*

2. *Live steam reheating*

In this method, HP turbine exhaust is reheated in a reheater by the live steam extracted from the superheater. Here, the use of long pipe can be avoided by placing the reheater near the turbine (Figure 5.26).

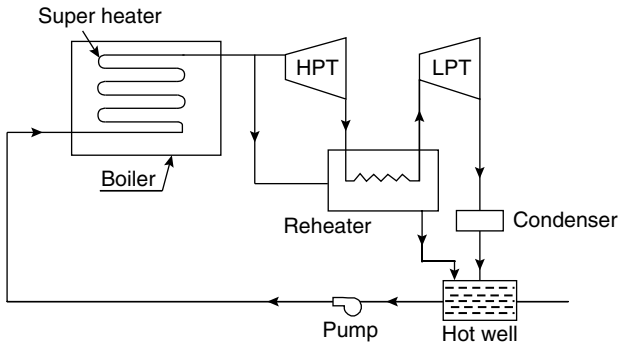


Fig. 5.26 *Live Steam Reheating*

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3. *Combination of gas reheating and live steam reheating*

The disadvantage of live steam reheating is that the steam cannot be reheated to its initial throttle temperature, therefore a combined system is used to have the advantages of the gas heating and live steam heating by placing these in series. The steam to be reheated is first passed through the live steam reheater and then through the gas reheater (Figure 5.27).

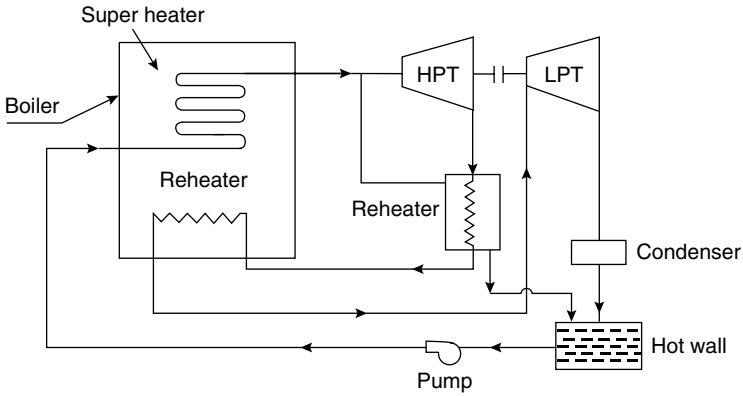


Fig. 5.27 *Combination of Gas Reheating and Live Steam Reheating*

5.6.4 Economizers

Economizer is a feed water heater that heats the feed water by flue gases. Figure 5.28 shows a typical return bend-type economizer. The construction of economizer employs a large member

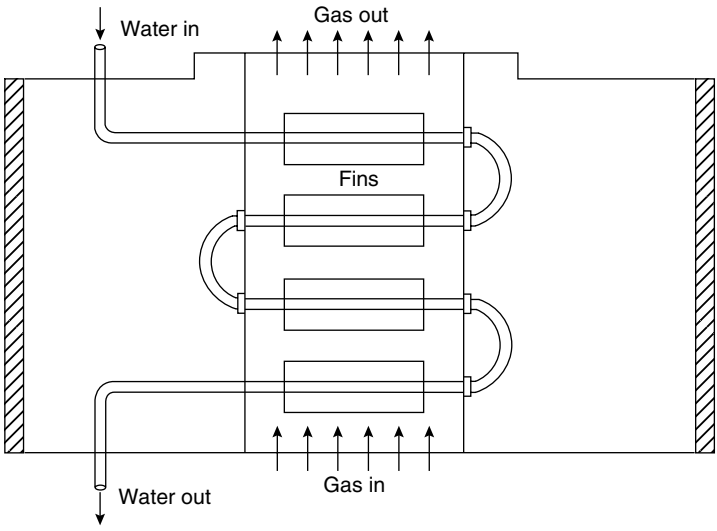


Fig. 5.28 *Return Bend Economizer*

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of closely spaced parallel tubes with thin walls and small diameters connected by headers of drums. The tubes are sometimes finned to increase the heat transfer surface. Water flows inside the tubes and flue gases flow outside usually in counter flow. The tubes are capable of withstanding high pressure of the boiler system. In addition, the feed water should have sufficient purity so that scaling and internal corrosion do not occur. Higher incoming water temperatures help in reducing external corrosion as gas temperatures are not lowered to such an extent that they are cooled to their dew point. A good economizer is designed to prevent accumulation of dust and soot and permit easy cleaning. The flue gases that act as heating medium should contact the entire surface of tubes and gas velocity through the economizer should be uniform without eddies being formed.

5.6.4.1 Types of Economizers

1. *Steaming*

In steaming economizers, a part of the feed water (5–7 per cent) is converted into steam. It is necessary that this steam should readily escape to the boiler.

2. *Non-steaming*

Non-steaming economizers heat the water to saturation temperature in the boiler.

3. *Independent*

Independent economizer is installed away from the main boiler setting connected to it through ducts. Such designs use tubes and are non-steaming.

4. *Integral*

Integral economizers use two drum headers connected to the bank of tubes water entering through one header and leaving through the other.

5.6.4.2 Other Advantages of the Economizer

There are several indirect advantages obtained by installing an economizer with a boiler plant as listed below:

1. The feeding of the boiler with water at a temperature near the boiling point reduces the temperature differences in the boiler, prevents the formation of stagnation pockets of the cold water and thus reduces greatly the thermal stress created in the pressure parts of the boiler and promotes better internal circulation.
2. When the feed water in not as pure as it should be, the temporary hardness is deposited on the inside of the economizer tubes, and although this necessitates internal cleaning of the economizer, the evil is not as great as internal cleaning of the boiler.
3. The temperature experienced in an economizer is not as high as in a boiler; the salts get deposited as a soft sludge and can be blown down through the economizer through a blow-off valve instead of forming a hard scale inside the boiler.
4. Even where the unwanted hardness is deposited as a hard scale on the inside of the economizer tubes, the economizer can be bypassed for internal cleaning without putting the boiler out of operation.

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5. Due to the reduction in the combustion rate of the furnace, the boiler will be more efficient, and the actual fuel saving will be greater than that theoretically calculated.
6. The flow of flue gases over the economizer tubes acts indirectly as a grit arrester, and a large portion of the soot and fly ash is deposited on the tubes and scraped off into the soot chamber. This reduces the emission of soot and fly ash.

5.6.5 Air Preheaters

The heat carried with the 'flue gases' coming out of economizer is further utilized for preheating the air before supplying to the combustion chamber. It has been found that an increase of 20°C in the air temperature increases the boiler efficiency by 1 per cent.

The air heater is not only considered in terms of boiler efficiency in modern power plants, but it is also considered as a necessary equipment for supply of hot air for drying the coal in pulverized fuel systems to facilitate grinding and satisfactory combustion of fuel in the furnace.

The use of preheater is much economical when used with pulverized fuel boilers because the temperature of flue gases going out is sufficiently large and high air temperature (250–350°C) is always desirable for better combustion.

Air heaters are usually installed in steam generators that burn solid fuels but rarely on gas or air fired units. By contrast, economizers are specified for most boilers burning liquid or gas or coal whether or not an air heater is provided.

The principal benefits of preheating the air are the following:

1. Improved combustion
2. Efficient use of low-grade fuel (high ash content)
3. Increased thermal efficiency
4. Saving in fuel consumption and
5. Increased steam generation capacity (kg/m^2-h) of the boiler.

The air heaters are not essential for the operation of steam generator, but they are used where a study of the costs indicates that some money can be saved or efficient combustion can be obtained by their use. The decision for its adoption can be made when the financial advantages are weighed against the capital and maintenance cost of the heater. The decision cannot be taken as the economic advantages of hot combustion extend to the size of the boiler, the efficiency of combustion, maintenance of furnaces and the saving of heat discharged to the chimney.

Since the heat of the exhaust gases cannot be fully extracted through the economizers, air preheaters are used to recover remaining heat escaping in these gases. If the gases can be cooled to 19°–22°C due to preheating of air, the plant efficiency increases by 2 per cent. A good air heater should have a high thermal efficiency, reliability and reasonable first cost. In addition, it should occupy small space, should be easily accessible and easily serviceable.

There are two types of air heaters: recuperative and regenerative type.

In recuperative type, the fluids are separated by a heat transfer surface and enclosed in a casing. One-third of the fluid flows continuously on one side, and the other third flows continuously on the other side. Recuperative heater may be either of tubular type or plant type. The tubular heater consists of a sheet of steel tubes spaced closely between the top and bottom sheets, the whole thing being enclosed in a casing. Flue gases may pass inside the tubes and air on their

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outside or vise versa, but the first arrangement in preferred as it is easier to clean the interior of the tubes than the exterior. The flow of the two fluids may be vertical or horizontal. But the disadvantage of these heaters are that they occupy a large space and their rate of heat transfer is low. It is difficult to remove nests of long tubes from vertical heaters and servicing becomes difficult (Figure 5.29).

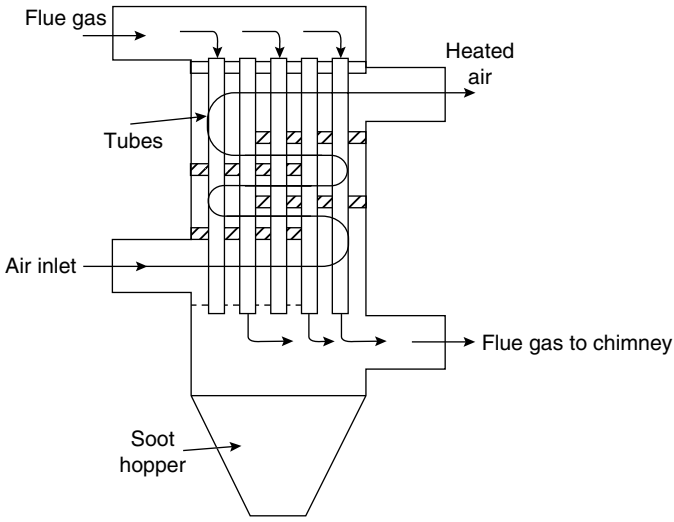


Fig. 5.29 *Tubular Air Preheater*

The plate-type air heater consists of a bunch of hollow leaves of sheet metal electrically welded and assembled in housing. The spacing between the adjacent leaves and the width of the opening of leaves are such that air and gas passages are of different widths so that greater volume of flue gases than of air to be preheated can flow through the heater. The two fluids flow in opposite directions – air usually flowing inside the hollow leaves and gases outside them. Initial cost and maintenance cost of plate heater is more than tubular heaters. The plates are likely to warp and bend, and it is always a problem to replace a spoiled plate (Figure 5.30).

In the regenerative heater, the same surface is alternately heated and cooled by the air and gas streams through respectively operating in cycles. It is often preferred for thermal stations due to its compactness, accessibility, reduced weight, high efficiency and less space requirement. It consists of a steel sheet box

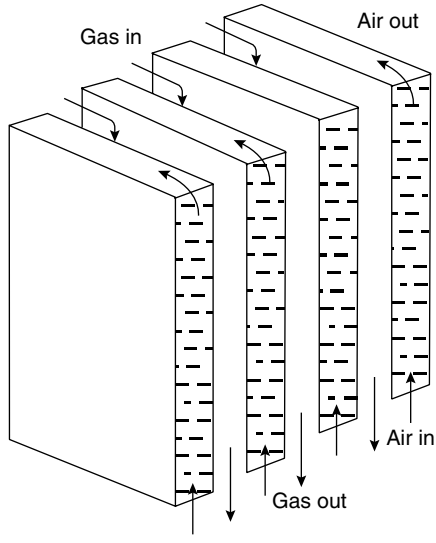


Fig. 5.30 *Plate Air Heater*

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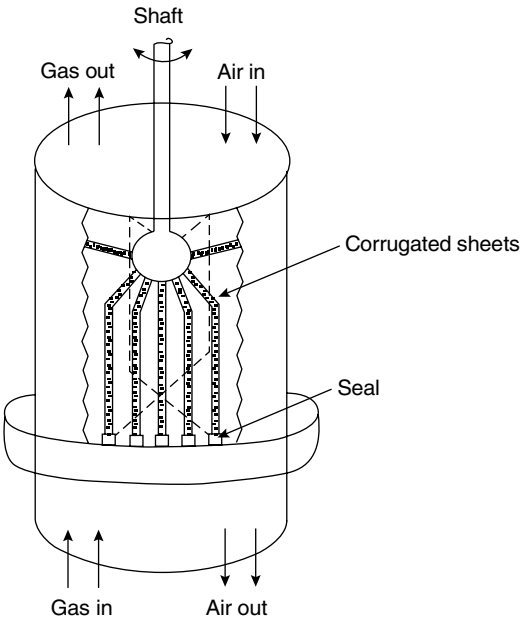


Fig. 5.31 *Plate Air Heater*

diametrically divided through elements of corrugated steel plates forming a cylindrical grid and spaced at about 6 mm from each other. This grid is fixed to a shaft that projects out of the box at the top and is driven by a motor at slow speed. The shaft has a ball bearing at the bottom and a thrust bearing at the top. At the top and bottom of the elements of the grid are fixed seals that wipe against top and bottom plates and prevent leakage. The box is diametrically divided into two halves by providing openings in half of the bottom cover plates. Air is passed through one half of the heater and gases through the other half in counter flow. The elements of the rotor falling in the gas portion are heated; and as they move on, they pass through the cool air portion and transfer heat to the air. Thus, alternately they pass from the gas side to the air side and the heat from gases is transferred to air (Figure 5.31).

5.7 BOILER MOUNTINGS

Boiler mountings are the external fittings that are required to ensure safe operation of the boiler.

These are necessary to regulate the steam flow, to measure certain parameters of water and steam, etc. The essential boiler mountings are as follows:

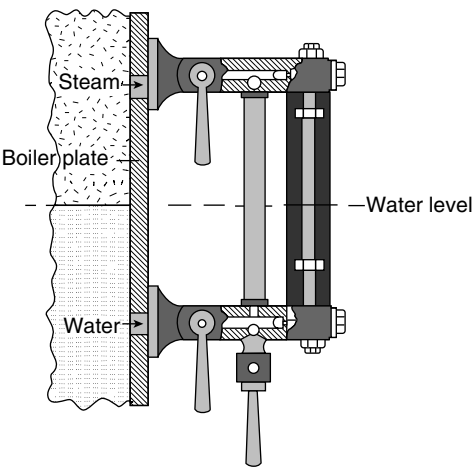


Fig. 5.32 *Water-Level Indicator*

1. Water-level indicators (two numbers)
2. Pressure gauge
3. Steam stop valve or junction valve
4. Feed check valve
5. Blow-down valve or blow-off cock
6. Fusible plug
7. Safety valves (two numbers)

The function of all these mountings are described below.

5.7.1 Water-Level Indicator

The function of the water-level indicator is to indicate the level of water inside the boiler drum at any given instant. Two water-level

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indicators (one serves as a standby) are fitted at the front of the boiler drum. The boiler operator keeps track of water level in the drum and operates the feed pump as per the requirement to maintain a constant level (about half) of water inside the drum (Figure 5.32).

5.7.2 Pressure Gauge

The function of the pressure gauge is to indicate the steam pressure inside the boiler drum in bar or in kgf/cm^2 or in kN/m^2 gauge pressure. If the boiler is fitted with a superheater, one more pressure gauge fitted to the superheater header indicates the superheated steam pressure at any given instant. As the gauge pressure is always above atmospheric pressure, its value should be added to the atmospheric pressure in order to know the absolute pressure (Figure 5.33).

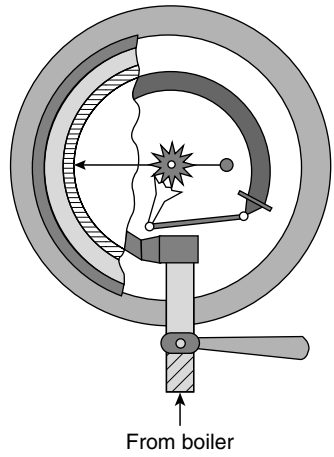


Fig. 5.33 *Pressure Guage*

5.7.3 Steam Stop Valve or Junction Valve

Steam stop valve or *junction valve* are essentially the same. Conventionally, stop valves are smaller in size, whereas junction valves are larger. When the valve is mounted on the topmost portion of the steam drum, normally the valve is called *junction valve*. If it is connected in the steam pipe to regulate the flow of steam, the valve is known as *stop valve*.

The function of the steam stop valve or junction valve is to shut off the steam flow or to regulate the steam flow as per the requirement or demand (Figure 5.34).

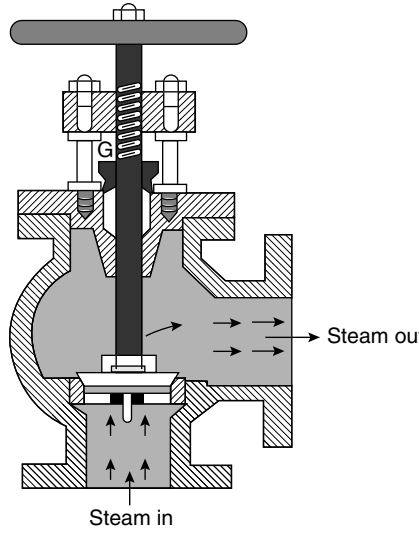


Fig. 5.34 *Steam Stop Valve*

5.7.4 Feed Check Valve

Feed check valve regulates the flow of feed water under pressure to the boiler drum. It is essentially a one-way valve and allows water to flow only in one direction, that is, towards the boiler drum. No water flows back from the boiler drum. It consists of two valves united together. One valve regulates the flow of water to the boiler drum or stops it completely, whereas another valve automatically prevents (operates by pneumatic pressure) water rushing back from the boiler (Figure 5.35).

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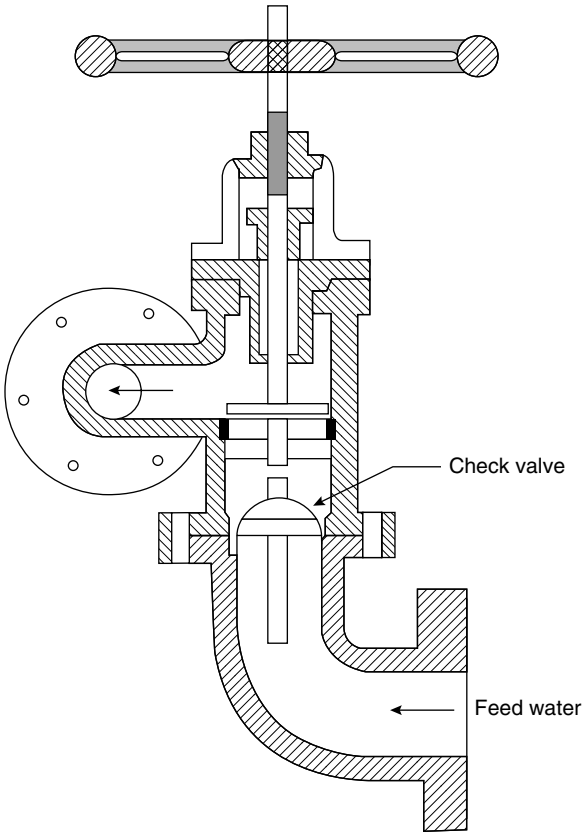


Fig. 5.35 *Feed Check Valve*

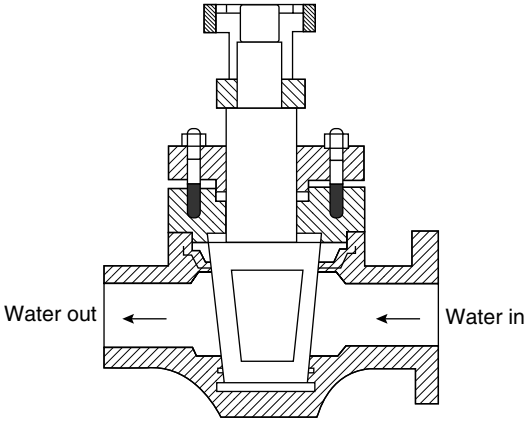


Fig. 5.36 *Blow-Down Valve*

5.7.5 Blow-Down Valve or Blow-Off-Cock

The function of the blow-down valve or blow-off cock is to remove the sludge or sediments collected at the bottom of the boiler drum from time to time. Whenever boiler cleaning and inspection are due water inside the tubes and in the boiler drum can be completely drained by operating the blow-down valve. Periodic blow-down is necessary to limit or to control the dissolved impurities in the feed water (Figure 5.36).

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5.7.6 Fusible Plug

The function of the fusible plug is to protect the firebox crown plate or fire tube from burning due to excessive heating. This usually happens when the water level inside the drum becomes too low and the shell and crown plate are directly exposed to steam space. The gun metal fusible plug fitted to the crown plate melts if the shell of the crown plate is overheated and hence allows remaining water inside the shell to fall in the furnace. This extinguishes the fire in the furnace, and hence prevents further damages (Figure 5.37).

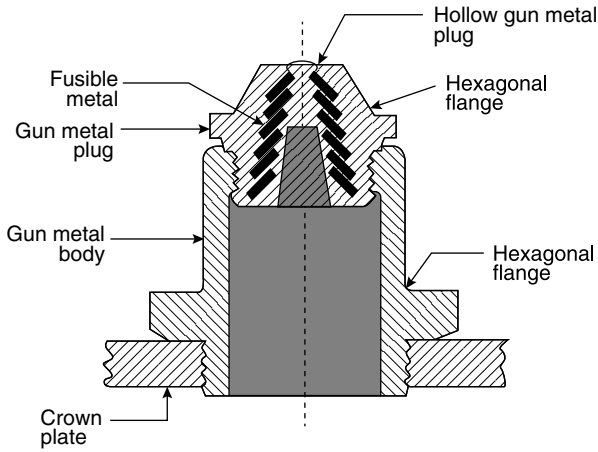


Fig. 5.37 *Fusible Plug*

5.7.7 Safety Valves

The function of the safety valve is to prevent the excessive steam pressure inside the boiler drum exceeding the design pressure. When the pressure inside the boiler drum exceeds the rated pressure, the safety valve automatically opens and discharges the steam to the atmosphere till normal working pressure is retained. This situation arises whenever furnace temperature increases causing excessive heat transfer or whenever steam demand suddenly drops.

Two types of safety valves are used in practice.

1. *Spring-loaded safety valve*

These valves are used in marine or locomotive boilers where there are chances of sudden jerk or vibration (Figure 5.38).

2. *Dead weight safety valve*

These valves are used only in stationary boilers such as Lancashire boiler or other low-capacity boilers. In this device, the upward steam pressure is balanced by the downward force of the dead weights acting on the valve.

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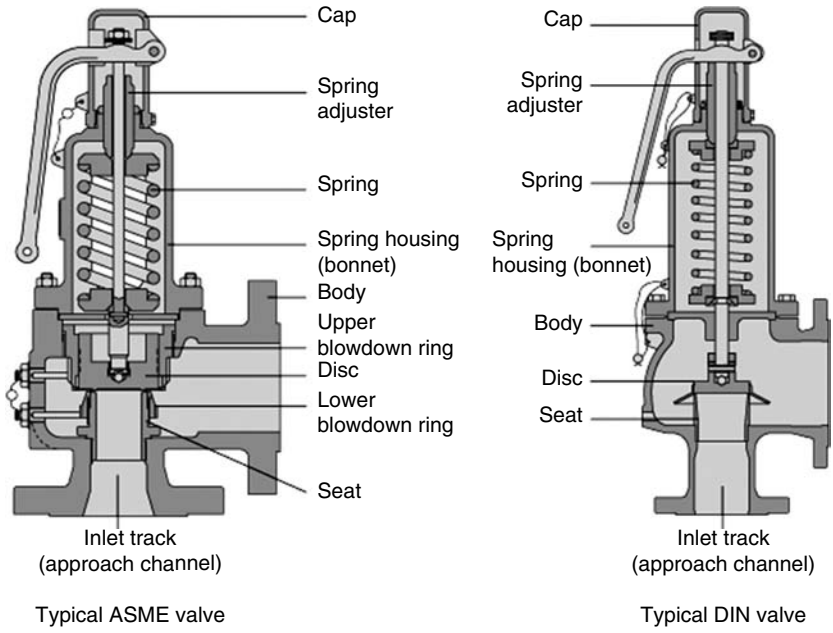
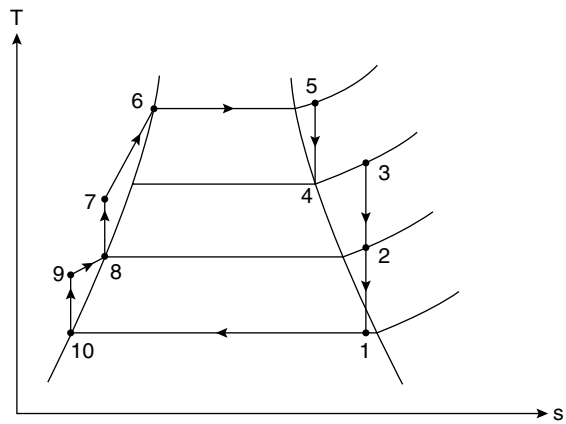


Fig. 5.38 *Spring-Loaded Safety Valve*

5.8 QUESTIONS

5.8.1 Objective Questions

1. The temperature – entropy diagram for a steam turbine power plant operating on the Rankine cycle with reheat and regenerative feed heating is shown in the figure given below. If m denotes the fraction of steam bled for feed heating, the work developed in the turbine per kilogram steam entering the turbine at state 5 is



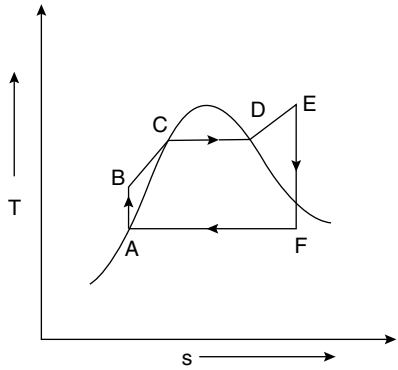
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(a) $(h_5 - h_4) + (1-m)(h_3 - h_1)$ (b) $(h_5 - h_4) + (h_3 - h_2) + (1-m)(h_2 - h_1)$

(c) $(2h_5 - h_4 - h_2) + (1-m)(h_2 - h_1)$ (d) $(h_5 - h_4) + (1-m)(h_3 - h_2)$

2. Assertion (A): For the same limits of boiler pressure and temperature, the specific stream consumption of ideal carnet cycle is less than that of ideal Rankine cycle.
Reason (R): For the same limit of boiler pressure and temperature, Carnot cycle is more efficient than Rankine cycle.

A superheat Rankine cycle is shown in the given T-s diagrams. Starting from the feed pump, the fluid flow up to the boiler exit is represented by state line.

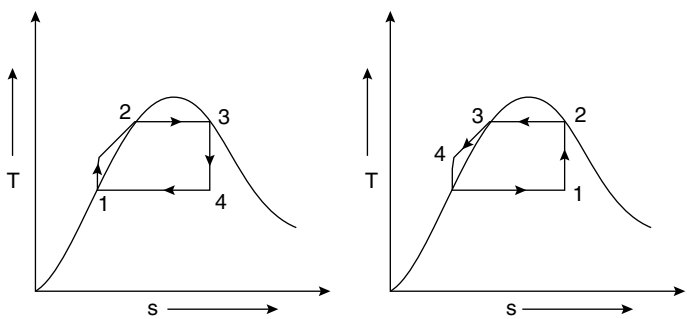


(a) ABCD
(b) BCDE
(c) ABDEFA
(d) ABCDE

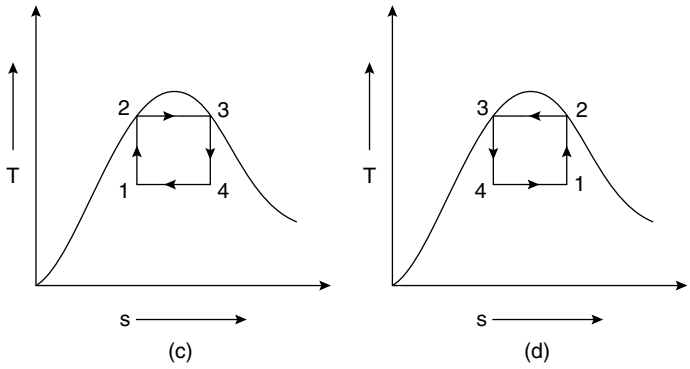
3. Blowing down of boiler water is the process of
(a) reducing the boiler pressure
(b) increasing the steam temperature
(c) controlling the solid concentration in the boiler water by removing some of the concentrated saline water
(d) None of the above

4. Consider the following statements regarding Rankine cycles:
(1) It reduces the specific steam consumption.
(2) It increases the dryness fraction of steam at the exhaust for the same value of condenser pressure.
(3) It reduces the cycle efficiency.
Of these statements
(a) 1 and 2 are correct (b) 2 and 3 are correct
(c) 1 and 3 are correct (d) 1, 2 and 3 are correct

5. The correct representation of a simple Rankine cycle on a T-s diagram is



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6. For a given set of operating pressure limits of a Rankine cycle, the highest efficiency occurs for
   (a) saturated cycle (b) superheated cycle
   (c) reheat cycle (d) regenerative cycle
7. Treatment of boiler feed water for the control of hardness is necessary to prevent
   (a) carryover (b) excessive feed water alkalinity
   (c) waterside scale deposits (d) foaming
8. In steam and other vapour cycles, the process of removing non-condensables is called
   (a) scavenging process (b) deaeration process
   (c) exhaust process (d) condensation process
9. Consider the following statements:
   (1) Boilers rated over 500 MW are not necessary super-critical boilers.
   (2) Power plant boilers are generally once-through boilers.
   (3) Blow down at regular intervals done to remove solids
   Of these statements,
   (a) 1, 2 and 3 are correct (b) 1 and 2 are correct
   (c) 2 and 3 are correct (d) 1 and 3 are correct
10. Boiler rating usually defined in terms of
    (a) maximum temperature of steam in K (b) heat transfer rate in kJ/h
    (c) heat transfer area in m^2 (d) steam output in kg/h
11. Constant pressure lines in superheated region of the Mollier diagram will have
    (a) a positive slope (b) a negative slope
    (c) zero slope (d) both positive and negative slopes
12. The equivalent evaporation (kg/h) of a boiler producing 2000 kg/h of steam with enthalpy content of 2426 kJ/kg from feed water at 40°C (liquid enthalpy = 168 kJ/kg) is
    (a) 2000 kg/h (b) 2149 kg/h
    (c) 1682 kg/h (d) 1649 kg/h
    Take enthalpy of vaporization of water at 100°C = 2258 kJ/h

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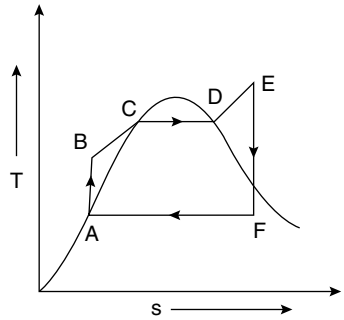
13. Match list I with list II (turbines) and select the correct answer using the codes gives below the lists:

List I	List II
(a) Two constant volumes and two adiabatic	(1) Ericsson
(b) Two constant pressures and two adiabatic	(2) Sterling
(c) Two constant volumes and two isothermals	(3) Joules
(d) Two constant pressure and two isothermals	(4) Otto

Codes:

	A	B	C	D
(a)	3	4	2	1
(b)	3	4	1	2
(c)	4	3	1	2
(d)	4	3	2	1

14. A superheat Rankine cycle is shown in the given T-s diagrams. Starting from the feed pump, the fluid flow up to the boiler exit is represented by state line.



(a) ABCD
(b) BCDE
(c) ABDEFA
(d) ABCDE

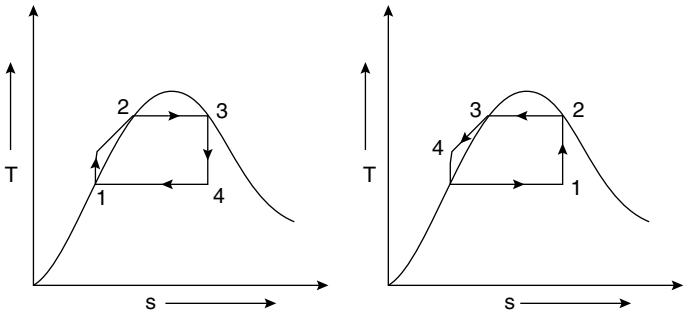
15. Consider the following statements regarding Rankine cycles.
(1) It reduces the specific steam consumption
(2) It increases the dryness fraction of steam at the exhaust for the same value of condenser pressure
(3) It reduces the cycle efficiency of these statements

(a) 1 and 2 are correct
(b) 2 and 3 are correct
(c) 1 and 3 are correct
(d) 1, 2 and 3 are correct

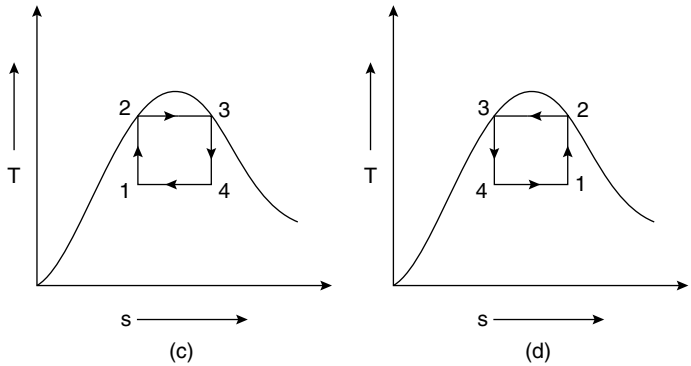
16. The control de-superheater of most boilers functions to control

(a) superheated steam flow
(b) superheated steam temperature
(c) de-superheated steam temperature
(d) superheater inlet temperature

17. The correct representation of a simple Rankine cycle on a T-s diagram is



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18. Under normal operating conditions, a drop in the steam temperature leaving an uncontrolled interdeck superheater could be caused by
    (a) decrease in steam velocity through the superheater
    (b) drop in the feed water temperature
    (c) badly fouled economizer
    (d) decrease in combustion gas velocity through the superheater

Answers

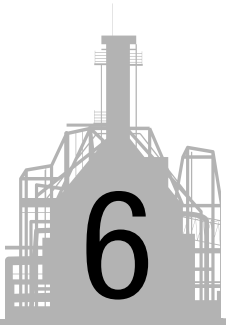
1. b 2. a 3. c 4. c 5. a 6. d 7. c 8. b 9. d 10. b 11. a 12. a
13. c 14. a 15. b 16. b 17. a 18. d

5.8.2 Review Questions

1. Explain the working of a Loeffler boiler with a neat sketch.
2. Describe various types of high-pressure boilers used in practice.
3. Explain the working of a 'Velox boiler' with a neat sketch.
4. Explain the working of a Benson boiler with a neat sketch.
5. Explain the meaning of boiler mountings and accessories. What is the need to have them in a power plant?
6. With a neat sketch, explain any one of the following:
   (i) Regenerative air heater
   (ii) Recuperative air heater
   (iii) Superheater
   (iv) Economizer
7. Why superheater temperature control is essential? Explain different means of achieving it.
8. What do you mean by reheating? Explain its significance.
9. Explain different methods of reheating.
10. Differentiate between the following:
    (i) Interdeck and overdeck arrangement
    (ii) Radiant and convective superheater
11. Explain different boiler mountings and their functions.
12. Explain the function of different boiler accessories.

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Fluidized Bed Combustion



6

Contents

6.1 Introduction to fluidized bed combustion
6.2 Regimes of combustion
6.3 Fluidized bed boilers – classification
6.4 Advantages of FBC system
6.5 Control of oxides of nitrogen
6.6 Desulphurization technology

6.1 INTRODUCTION TO FLUIDIZED BED COMBUSTION

Due to the availability of low calorific value, high ash content coal in India, it is essential to use technology-driven fluidized bed combustion (FBC) concept over the conventional stoker firing system. The benefits an FBC offers are as follows:

(a) Compact boiler design
(b) Flexibility in using fuel
(c) Higher combustion efficiency
(d) Reduced emission from pollutants such as NO_x and SO_x

6.2 REGIMES OF COMBUSTION

FBC works on the principle by passing an evenly distributing air or gas current upward through a finely divided bed of solid particles such as sand supported on a fine mesh. Referring to the Figure 6.1, when the air velocity is gradually increased, the individual particles are suspended in the air stream and the bed is called "fluidized". Further, increase in air velocity bubbles form with vigorous turbulence and rapid mixing. This results in the formation of dense bed surface containing solid particles that exhibit the properties similar to a boiling liquid. This stage is referred to as "bubbling fluidized bed". As velocity is increased further, bubbles disappear and particles are blown out of the bed, and hence some amounts of particles are re-circulated to maintain a stable system. This stage is referred to as "circulating fluidized bed". [1]

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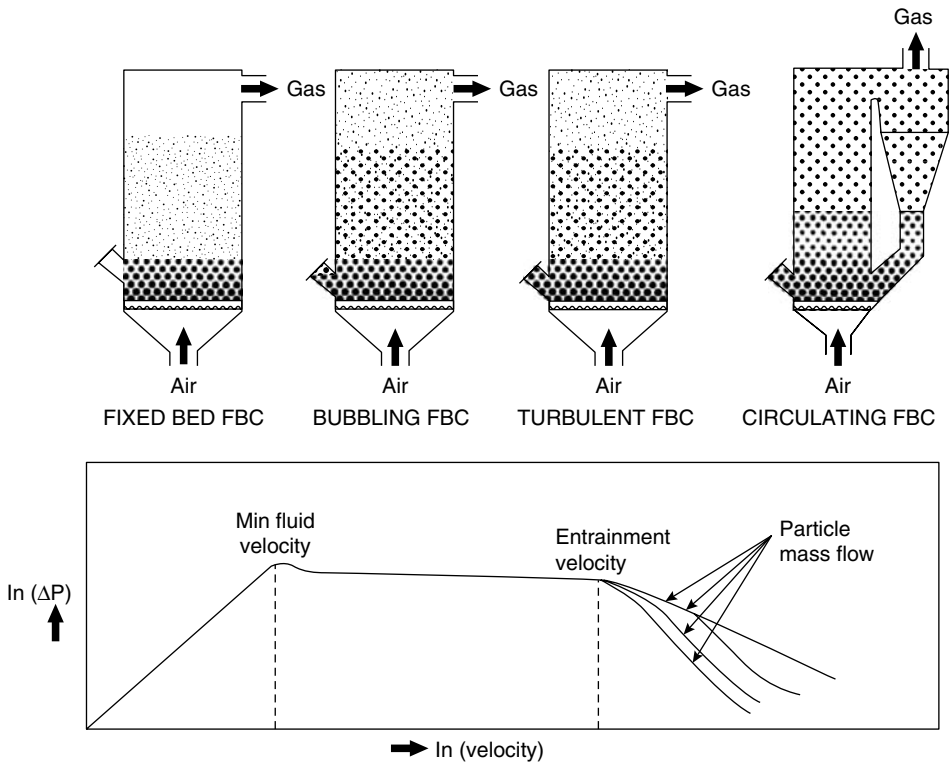


Fig. 6.1 *Principles of Fluidization*

Fluidization mainly depends on the particle size and the air velocity as shown in Figure 6.2. It can be observed from the figure that the mean solids velocity increases at a slower rate than the gas velocity, and this difference between the mean solid velocity and mean gas velocity is known as slip velocity. In order to have good heat transfer and intimate contact, it is essential to maintain maximum slip velocity between the solids and the gas. By heating the sand particles in fluidized state to the ignition temperatures of fuel, which could be rice husk, coal or bagasse, and by injecting the fuel continuously into the bed, a uniform temperature bed temperature is attained. Typically, FBC takes place at about 840°C to 950°C. The problems associated with the melting of ash are avoided in an FBC system because the bed temperature is much below the ash fusion temperature. Since mixing is rapid in a fluidized bed, the rate of heat transfer in bed tubes and walls of the bed is higher. In order to ensure a stable operation of the bed and avoid particle entrainment in the gas stream, it is always necessary that minimum fluidization velocity and particle entrainment velocity be maintained.

In an FBC, turbulence promoted by fluidization results in improved mixing and even distribution. In addition, due to higher particle resident time when compared to conventional stoker firing system, heat generation is rapid and more, even at lower temperature. If a fuel contains more sulphur content, a limestone particle bed is used to control SO_x and NO_x formation.

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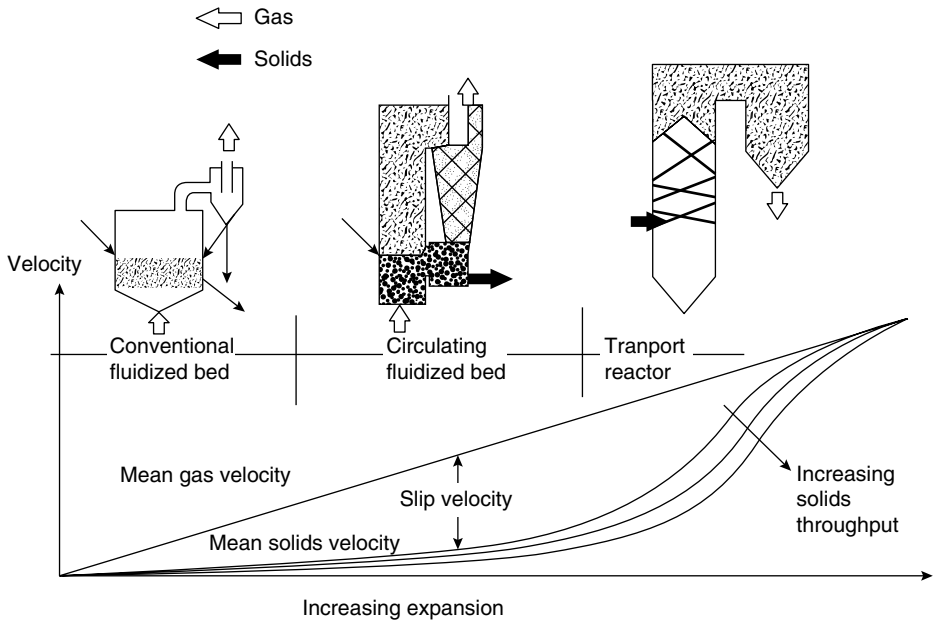


Fig. 6.2 *Variation of Gas Velocity with Solid Velocity*

6.3 FLUIDIZED BED BOILERS – CLASSIFICATION

FBC boilers are generally classified into the following three basic types: (1) atmospheric FBC (AFBC) system or bubbling FBC, (2) atmospheric circulating (fast) FBC (CFBC) system and (3) pressurized FBC (PFBC) system. [3]

6.3.1 Atmospheric FBC System or Bubbling FBC

Atmospheric FBC (AFBC) system is a versatile boiler used for burning various fuels including agricultural residues (rice husk or bagasse) and even low-quality coal. This type of boiler is used for combined heat (process industries) and power generation application.

In AFBC boilers, the fuel is sized based on the type of fuel and fuel feeding system before feeding into the combustion chamber. If coal is used as a fuel, it is crushed to a size of 1–10 mm before burning. The hot atmospheric air, which acts as both the fluidization air and combustion air, is delivered under pressure through the nozzles at a velocity of 1.2 to 3.7 m/sec.

Almost all AFBC/bubbling bed boilers use in-bed evaporator tubes in the bed of limestone, sand and fuel for extracting the heat from the bed to maintain the bed temperature. The bed depth is usually 0.9–1.5 m deep and the pressure drop averages about 1 inch of water per inch of bed depth. Since a small amount of solid particles leave the bubbling bed (about 2–4 kg of solids per ton of fuel burned), this is recycled. Figure 6.3 shows a cut section view of an AFBC boiler.

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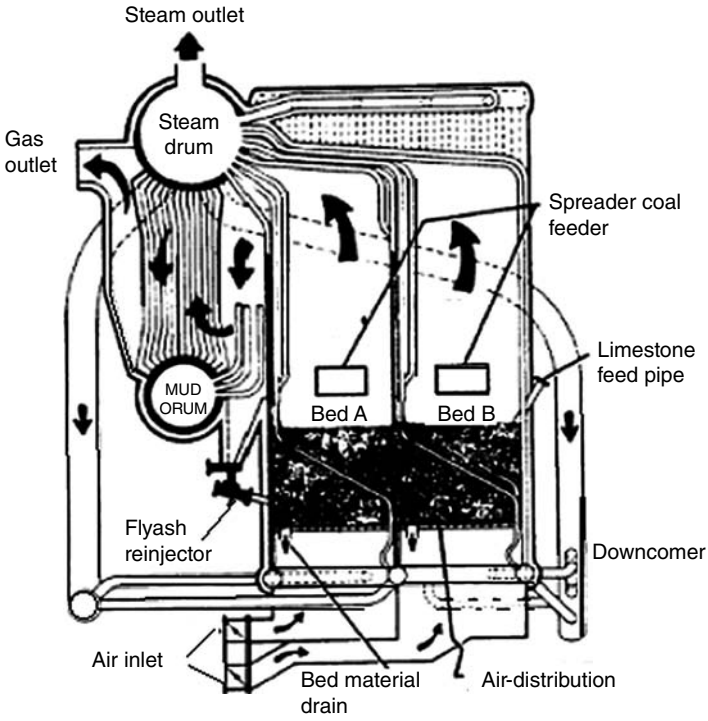


Fig. 6.3 *A Cut Section View of AFBC Boiler*

In the AFBC, the combustion gases pass over the super heater tube sections initially and then flow past the economizer, the dust collectors and the air pre-heaters before exiting through the chimney. However, the operating conditions of these boilers are restricted by a relatively narrow temperature range within which the bed must be operated. With coal, if the bed temperature exceeds 950°C chances of clinker formation are more and if the temperature falls below 800°C in the bed, the combustion efficiency drops. For efficient sulphur retention, and combustion efficiency, the ideal working temperature range is 800–850°C.

General Arrangements of AFBC Boiler

AFBC boilers comprise of following systems, and many of these systems are common to all types of FBC boilers:

(a) Fuel feeding system
(b) Air distributor
(c) Bed and in-bed heat transfer surface
(d) Ash-handling system

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(a) *Fuel feeding system*

This system is used to feed fuel and adsorbents such as limestone or dolomite, using any of the following two methods: under bed pneumatic feeding and over-bed feeding.

- *Under bed pneumatic feeding*

In under bed feeding, coal is crushed to 1–6 mm size and pneumatically conveyed from feed hopper to the combustor through a feed pipe piercing the distributor. For high capacity boilers, multiple such feeders are used at multi-points to ensure uniform distribution.

- *Over-bed feeding*

In over-bed feeding, crushed coal, 6–10 mm size is conveyed from coal bunker to a spreader by a screw conveyor. The spreader distributes the coal over the surface of the bed uniformly. Oversized coal particles can be used in this system. Over-bed feeding system is quite economical and ideal for rice husk and other agricultural residues.

(b) *Air distributor*

Air distributor plates are used to uniform distribute the fluidizing air across the bed cross section. This ensures constant motion of solid particles, preventing the formation of de-fluidization zones within the bed. The distributor plates (metallic) with a number of perforations (known as nozzles) are fixed to the furnace floor in a definite geometric pattern. The nozzles or nozzles with bubble caps prevent solid particles from flowing back into the space below the distributor. The distributor plate is protected from high temperature of the furnace by a refractory lining or by providing a static layer bed material or by means of water cooled tubes.

(c) *Bed and in-bed heat transfer surface*

As mentioned earlier, the bed material has an average size of about 1 mm and can be sand, ash, crushed refractory or limestone. Depending on the bed height either a shallow bed or a deep bed could be used. Even though the fluidizing velocity remains same, the two ends fluidize differently, with different heat transfer rates from the surfaces. The shallow bed system has distinct advantage of lower bed resistance, and hence a lower pressure drop and lower fan power requirements over a deep bed.

In-bed heat transfer surface is made of tube bundles, or coils with horizontal, vertical or inclined positions. However, the rate of heat transfer in the bed depends on parameters such as bed pressure and temperature, particle size, superficial gas velocity, heat exchanger and distributor plate design.

(d) *Ash-handling system*

In the FBC boilers, the bottom ash constitutes roughly 30–40 per cent of the total ash, the rest of it being the fly ash.

- *Bottom ash removal*

Bottom ash removal is essential in order to eliminate over-sized particles and to avoid accumulation and consequent de-fluidization. To maintain constant bed height, bed ash is removed continuously and bottom ash is removed periodically.

- *Fly ash removal*

Since the fly ash contains 60–70 per cent of total ash, due to elutriation of particles at high velocities, it is removed in a number of stages: initially in convective section, then from the bottom of

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air pre-heater/economizer section and finally in dust-collecting systems such as cyclone separators, bag house filters, electrostatic precipitators (ESPs) or a combination of all of these.

6.3.2 Circulating FBC

Some of the drawbacks associated with conventional bubbling bed combustion system are overcome by circulating FBC (CFBC). A circulating FBC (CFBC) system shown in Figure 6.4 is preferred when the boiler capacity is large or medium, inferior quality coal is handled and it is essential to contain pollution due to NO_x formation.

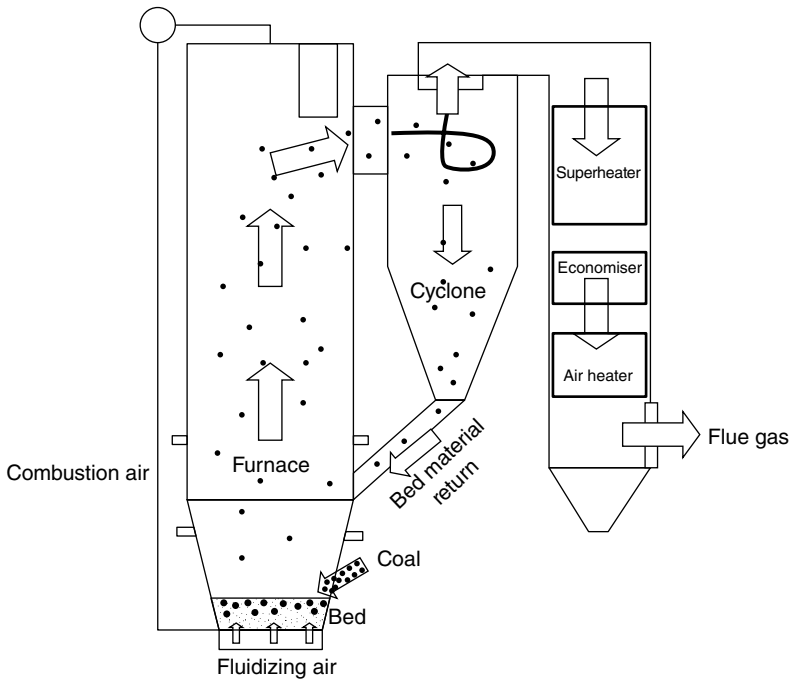


Fig. 6.4 *Circulating Fluidized Bed Combustion*

Advantages of a CFBC boiler are the following:

(a) Higher combustion efficiency and processing capacity

(b) Lower NO_x formation due to constant operating temperature of around 850°C

(c) Low-pressure combustion air, 10–14 kPa as against 20–35 kPa is used in an AFBC system

(d) Heat transfer surfaces are parallel to flow, which enables minimum erosion and corrosion of tubes. In an AFBC system, heat transfer surfaces are perpendicular to the flow

(e) Better turndown ratio as compared with a bubbling system

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In a CFBC system, crushed coal of 6–12 mm size and limestone is injected into the furnace combustor. About 60–70 per cent of total air moving in the upward direction through the nozzles of the distributor plates holds the particles in suspension. The air velocity is limited to 3.7–9 m/s here. The remaining air contributes as secondary air and is admitted from above the furnace bottom. Finer particles of < 450 μ are elutriated out of the furnace along with flue gases moving at 4–6 m/s. These particles are collected in solid cyclone separators (50 kg–100 kg/kg of fuel burnt) and circulated back into the furnace.

These boilers can handle 75–100 tph of steam, but they require big cyclone separators to handle large amount of bed material. The boiler is taller when compared with AFBC boilers. Recirculating the particles improves heat transfer efficiency and better calcium to sulphur utilization 1.5:1.0 as against 3.2:1, compared with AFBC boilers.

In a CFBC system, the bed is designed in such a way that most of the heat transfer takes place outside the combustion zone with no steam generation tubes present in the bed section. External heat exchangers are also used sometimes to achieve this.

6.3.3 Pressurized FBC System

Pressurized FBC (PFBC) system is preferred in handling large-scale coal-burning systems in a power plant at a bed operating pressure of around 16 bar.

Figure 6.5 shows a typical PFBC system [2] used for co-generation plant. The hot flue gases coming from the combustion chamber of the PFBC system is expanded in the gas turbine. Part of the power generated by the gas turbine is used to run the compressor that supplies air to the

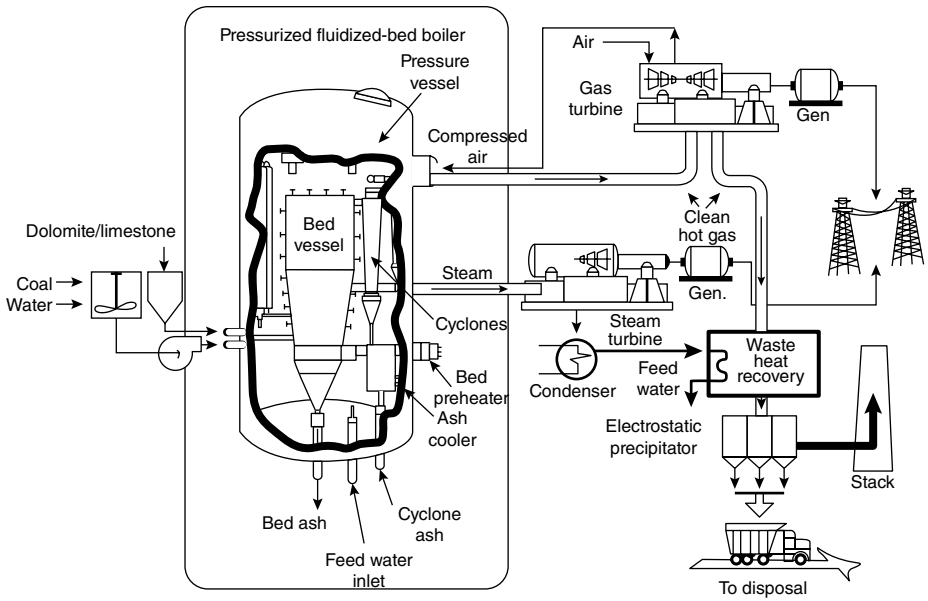


Fig. 6.5 *A PFBC System Used for Co-Generation (Source: Bureau of Energy Efficiency)* [4]

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combustion chamber. The expanded gases are then passed through a waste heat recovery system to heat the incoming condensate water from the steam turbine condenser. The steam generated in the PFBC system is expanded in a steam turbine to generate power. By the combined cycle operation, the overall efficiency of the system increases by around 5–8 per cent.

6.4 ADVANTAGES OF FBC SYSTEM

FBC boilers offer the following advantages:

(a) *High efficiency*

FBC boilers have high combustion efficiency of 92–95 per cent and overall efficiency of 82–86 per cent.

(b) *Fuel flexibility*

FBC boilers can burn variety of fuels from agricultural products to low-grade high ash content coal with high efficiency. Coal fines of less than 6 mm size can be easily burnt, which otherwise is difficult to burn in conventional boilers.

(c) *High heat transfer rate*

High rate of heat transfer is possible within a small area, reducing the boiler size compared with a conventional system.

(d) *Low erosion and corrosion*

Erosion and corrosion problems are minimized due to the lower operating temperature. In addition, chances of clinker formation are reduced drastically in these boilers.

(e) *Less pollution*

Due to the lower operating temperature, formation of NO_x is reduced. Also, addition of limestone or dolomite for coal containing high percent sulphur reduces the SO_2 formation.

(f) *Quick startup and fast response to load variations*

Boiler startup and shutdown time is less, as turbulence in the combustion zone is rapid. Response to varying load demand is very good.

(g) *Easy atomization*

Complete atomization of fuel and ash-handling systems is possible. Micro-processor-based automatic ignition systems give better combustion control and fuel economy.

(h) *High reliability and reduced maintenance*

As no moving parts are present in the combustion zone, system is highly reliable with reduced maintenance cost.

6.5 CONTROL OF OXIDES OF NITROGEN

Production of NO_x is an endothermic reaction and its concentration is temperature dependent. NO_x emissions can be reduced by lowering the combustion temperature and by eliminating hot spots in the furnace. It can also be inhibited by lowering the air–fuel ratio or by employing

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exhaust gas recirculation. Lowering the air–fuel ratio restricts the amount of oxygen available for combustion while exhaust gas recirculation reduces the combustion chamber temperature.

6.6 DESULPHURIZATION TECHNOLOGY

Flue gas desulphurization (FGD) is a technology that extracts sulphur dioxides from flue gases produced in coal-based thermal power plants, where sulphur content in coal is more than 0.5 per cent. The coal produced from Indian mines contains only 0.4 per cent sulphur contents; therefore in India, this technology was not required much. Now, a lot of companies in India are importing coal from other countries such as Indonesia and South Africa which contains sulphur contents 0.6–0.9 per cent. Thus, it is mandatory to install the FGD plant in order to maintain the ambient air-quality standards.

Sulphur dioxide is extracted from flue gases in wet scrubber, slurry of alkaline sorbent; lime stone reacts with the sulphur dioxide. More than 90 per cent of the limestone particles in the limestone powder (85% purity) are made to pass through a screen (325 mesh) to ensure that maximum amount of limestone particles come in contact with the sulfur dioxide molecules in the flue gas.

To have a chemical reaction, the limestone powder is mixed into 15–30 per cent slurry introduced into the FGD vessel, re-circulated, and sprayed into the flue gas stream. The following chemical reactions take place in the FGD system:

(a) Flue gases containing SO_2 enter the absorber and come in contact with limestone slurry ($CaCO_3$) in the wet scrubber thereby producing calcium sulfite.

$$CaCO_3 + SO_2 \rightarrow CaSO_3 + CO_2$$

(b) Further, $Ca(OH)_2$ in lime, when combined with SO_2 gas produces calcium sulfite and water

$$Ca(OH)_2 + SO_2 \rightarrow CaSO_3 + H_2O$$

(c) Calcium sulfite is further oxidized by forced oxidation utilizing blowers to produce marketable $CaSO_4$ $2H_2O$, popularly known as gypsum.

$$CaSO_3 + H_2O + ½O_2 \rightarrow CaSO_4.2H_2O$$

6.7 QUESTIONS

6.7.1 Objective Questions

1. Combustion temperature in FBC boilers is
   (a) 1,200°C
   (b) 900°C
   (c) 1,400°C
   (d) 1,800°C
2. Working pressure of an PFBC boilers
   (a) 12 bar
   (b) 14 bar
   (c) 16 bar
   (d) 18 bar

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3. The efficiency of a typical FBC boiler would be
   (a) 84%
   (b) 48%
   (c) 44%
   (d) 34%
4. The efficiency of a typical combined cycle system increases by
   (a) 4%–5%
   (b) 5%–8%
   (c) 14%–24%
   (d) 34%
5. Low combustion temperature in the furnace of FBC is responsible for
   (a) SO_x
   (b) NO_x
   (c) CO_2
   (d) CO
6. In AFBC boilers, coal size used is
   (a) 1–10 mm
   (b) 10–15 mm
   (c) 5 cm
   (d) 3 cm
7. The limestone or dolomite used in FBC plant removes
   (a) SO_x
   (b) NO_x
   (c) ash
   (d) sulphur

Answers

1. b 2. c 3. a 4. b 5. b 6. a 7. d

6.7.2 Review Questions

1. Explain the different regimes of combustion of an FBC boiler.
2. Explain the principle of working of an FBC.
3. How do you classify the FBC boilers? Explain.
4. With a neat sketch, explain the working of an AFBC boiler.
5. Sketch and explain the working of a CFBC boiler.
6. What are the benefits a CFBC boiler offers? Discuss.
7. List and discuss the advantages of an FBC boiler.
8. Write a short note on PFBC boiler.
9. Explain briefly the methods used to control SO_x and NO_x in FBC boilers.

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6.7.3 References

1. *Energy Technology Handbook*. Considine, D. M.; New York: McGraw-Hill, Inc., **1977**.
2. *Pressurized FBC Technology*. Podolski, W. F.; US: Noyes Data Corporation, **1983**.
3. Fluidised Bed Coal Fired Boilers – Department of Coal Publications, Government of India Fluidised Combustion of Coal – A National Coal Board Report, London.
4. Bureau of Energy Efficiency, Government of India.

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Draught System 7

Contents

7.1 *Introduction to draught system*
7.2 *Air and supply systems (natural mechanical draught systems)*
7.3 *Chimneys*
7.4 *Calculations involving height of chimney to produce a given draught*

7.1 INTRODUCTION TO DRAUGHT SYSTEM

Boiler draught is the pressure difference required to maintain constant flow of air into the furnace and to discharge the flue gases to the atmosphere through a chimney. Thus, boiler draught is one of the most essential systems of a thermal plant.

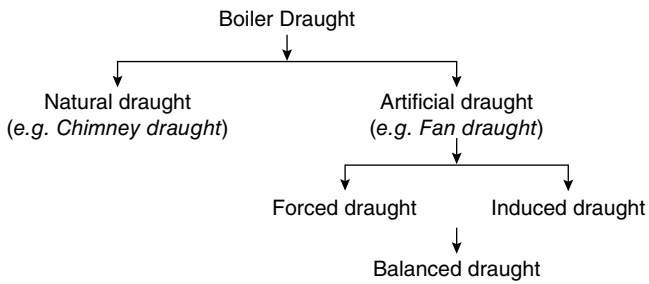
The total draught required to produce the current of air and to expel the flue gases is given as follows:

Total draught loss = Velocity head loss at the chimney exit + fuel bed resistance + head loss in equipments such as economizer, air heater, etc. + head loss in the chimney and ducts

i.e., $H_t = H_v + H_b + H_e + H_{cd}$

7.2 AIR AND SUPPLY SYSTEMS (NATURAL MECHANICAL DRAUGHT SYSTEMS)

Boiler draught is generated by using chimney, fan, steam jet or air jet, or a combination of both. Based on this, draught is classified as follows:



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1. *Natural draught*

Natural draught is produced by making use of a chimney. A chimney is a vertical tubular structure that is made up of brick, steel or reinforced concrete. The draught produced by the chimney is due to the difference in temperature of hot flue gases inside the chimney and the atmospheric air. In addition, the height of the chimney above the furnace grate, weather conditions and boiler-operating conditions also have considerable effect on the amount of natural draught (Figure 7.1).

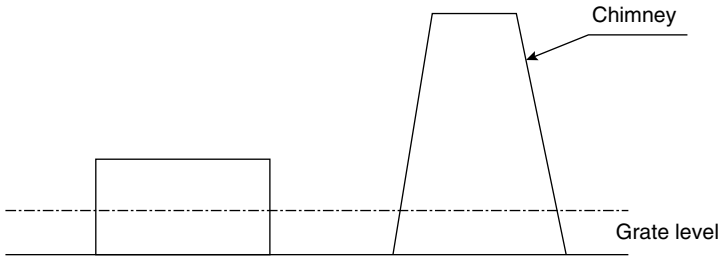


Fig. 7.1 *Natural Draught*

2. *Artificial or mechanical draught*

This type of draught is preferred when the draught produced by the chimney is not sufficient or when a certain draught is to be maintained irrespective of atmospheric temperature. In modern high-pressure boilers, the use of economizers, superheaters and air preheaters reduces the exit flue gas temperature considerably. However, this would necessitate an increased chimney height – the cost of which cannot be justified. By using artificial or mechanical draught, the height of the chimney can be considerably reduced. Mechanical draught is of two types: forced and induced.

(i) *Forced draught*

In the forced draught (FD) system (Figure 7.2), a blower is installed near the base of the boiler that blows air into the furnace. The pressure of air throughout its path is above atmospheric pressure. This system also uses a chimney to discharge hot flue gases into the atmosphere.

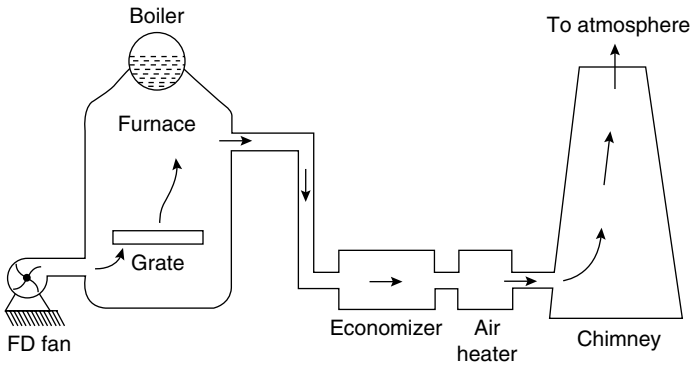


Fig. 7.2 *Forced Draught*

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(ii) *Induced draught*

In the induced draught (ID) system, the blower is installed near the base of the chimney so as to facilitate sucking of flue gases from the furnace. Thus, pressure inside the furnace is reduced below atmospheric pressure, inducing outside air into the furnace. The heat in exhaust gases can be recovered as much as possible by installing an air preheater and economizer along the gas flow path. As the draught produced in this system is independent of flue gas temperature, care should be taken such that the fan handles gas at its lowest possible temperature. *The total draught produced by the system is equal to the sum of the fan and chimney draughts.*

However, when the furnace doors are opened for firing purpose, the air from outside rushes into the furnace and reduce the draught.

(iii) *Balanced draught*

A balanced draught system (Figure 7.3) combines the features of FD and ID. In the case of FD, when the furnace doors are opened, high-pressure air rushes outside and even blows out the fire entirely. In the case of ID, when the furnace doors are opened, atmospheric air enters the furnace and causes an imbalance in the draught. This difficulty is overcome by using *balanced draught.*

In this system, an FD fan installed near the boiler helps in overcoming the resistance of fuel bed by supplying sufficient air for combustion. An ID fan installed near the chimney base removes the flue gases that come from the furnace, allowing the furnace pressure to be maintained slightly below atmospheric pressure.

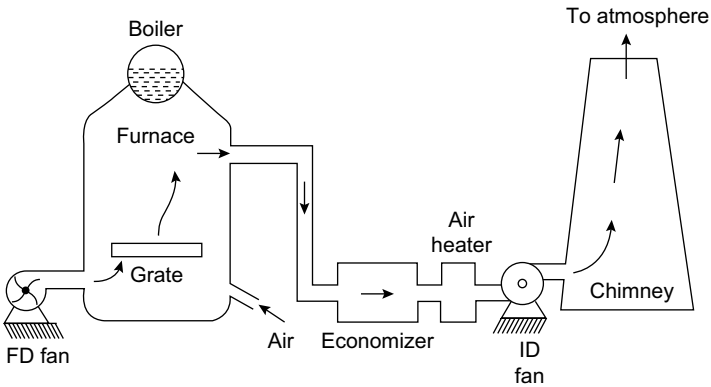


Fig. 7.3 *Balanced Draught*

3. *Steam jet draught*

Depending on the location of steam jet, draught produced may be either induced or forced.

(i) *Induced draught*

This type of draught is used in a boiler for a locomotive. When the locomotive is stationary, steam generated in the boiler is fed to the smoke box through the nozzles to create draught. When the locomotive moves, air enters through the dampers and makes its way through the grate and smoke box. In addition, exhaust steam from the engine cylinder is utilized to create the ID by passing it through the steam nozzle in the smoke box (Figure 7.4).

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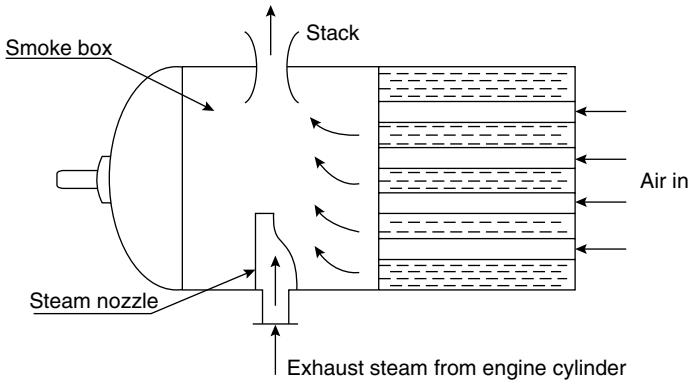


Fig. 7.4 *Induced Steam Jet Draught*

(ii) *Forced draught*

This type of draught is also known as *turbine draught* (Figure 7.5). In this case, steam from the boiler is throttled to a pressure of 1.5–2 bar and passed through a series of nozzles fitted to a diffuser pipe. The high-velocity steam that comes out from the nozzle drags the air column along with it, thereby creating a suction. Fresh air from outside enters the diffuser pipe. The kinetic energy of air and steam mixture is converted into pressure energy forcing air through the coal bed to the furnace. However, steam jet draught is used only as a booster in conjunction with chimney draught.

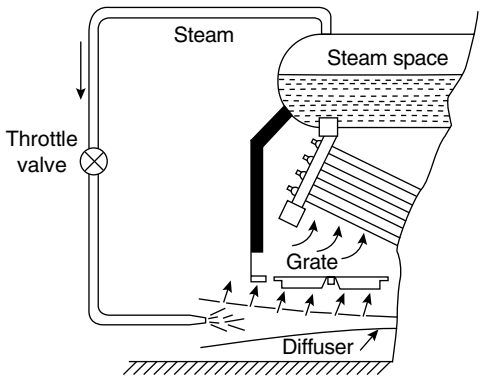


Fig. 7.5 *Forced Steam Draught or Turbine Draught*

7.3 CHIMNEYS

A *chimney* is a structure used for venting hot flue gases or smoke from a steam generator or boiler to the outside atmosphere. The working of chimney is based on natural draught or stack effect. Due to lower density, the hot air rises to the top naturally through the chimney. The taller the chimney, the more draught it will produce at a given temperature difference. The height of

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the thermal power plant chimneys is dictated by government rules and regulations that change from to time on the basis of air pollution. As the air pollution standards have become stringent, the height of the chimneys has increased – from 100 m to 150 m of earlier power stations to 220–275 m in the case of modern power plants.

7.4 CALCULATIONS INVOLVING HEIGHT OF CHIMNEY TO PRODUCE A GIVEN DRAUGHT

Consider the height of the chimney (Figure 7.6) above the grate level 'H'. The pressure acting on the grate from the *chimney* side is as follows:

$$p_1 = p_a + w_g H$$

The pressure acting on the grate from *atmospheric* side is given by

$$p_2 = p_a + w_a H$$

where p_a = atmospheric pressure, w_a = weight density of air and w_g = weight density of hot gases.

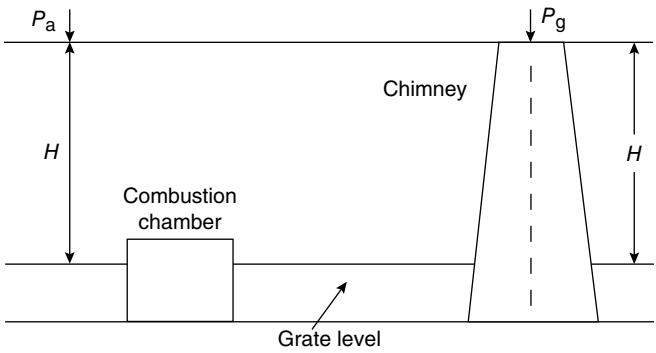


Fig. 7.6 *Chimney Height Calculation*

The net pressure acting on the grate of the combustion chamber due to the pressure exerted by gas column and air column is given by the following expression:

$$p = p_2 - p_1 \text{ (as } w_a > w_g)$$

$$= (p_a + w_a H) - (p_a + w_g H)$$

$$= H(w_a - w_g)$$

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This pressure difference is known as static draught and is responsible for causing the flow of air through the chimney. This pressure is always small (<12 mm of water) and can be increased by either increasing the density of air or reducing the density of hot gases.

7.4.1 Chimney Height and Diameter

Let us assume the volume of products of combustion to be equal to the volume of air supplied, both reduced to the same pressure and temperature conditions.

The pressure difference in terms of water head is given by

$$\frac{h_w}{1000} \times w_w = H(w_a - w_g)$$

where h_w = equivalent mm of water head and w_w = weight density of water.

$$\frac{h_w}{1000} \times \rho_w g = H(\rho_a - \rho_g) g$$

$$\frac{h_w}{1000} \times 1000 = H(\rho_a - \rho_g)$$

$$h_w = H(\rho_a - \rho_g) \text{ mm of water head} \quad (1)$$

Mass of chimney gases produced:

$$= (m_a + 1) \text{ kg of fuel burnt}$$

T_a = absolute temperature of atmosphere

T_g = average absolute temperature of chimney gas

$$\text{Then, } \frac{\text{Mass of hot gases}}{\text{Mass of air}} = \frac{m_a + 1}{m_a}$$

$$\text{Density of air at 273 K, } \rho_{273} = \frac{p}{RT} = \frac{1.03 \times 10^4}{29.27 \times 273}$$

$$= 1.293 \text{ kg/m}^3$$

$$\text{Density of air at } T_a\text{K, } \rho_a = 1.293 \times \frac{273}{T_a}$$

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Density of gases at T_gK, $\rho_g = \frac{p}{RT_g} \times \frac{m_a + 1}{m_a}$

$$= 1.293 \times \frac{273}{T_g} \times \frac{m_a + 1}{m_a}$$

Substituting in expression (1):

$$h_w = H\left[1.293 \times \frac{273}{T_a} - 1.293 \times \frac{273}{T_g} \times \frac{m_a + 1}{m_a}\right]$$

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g} \times \frac{m_a + 1}{m_a}\right] \text{ mm of water}$$

Assuming that the same draught is produced by gas column of height H_1

Then, $H_1 \times 1.293 \times \frac{m_a + 1}{m_a} \times \frac{273}{T_g} = 353H\left[\frac{1}{T_a} - \frac{1}{T_g} \times \frac{m_a + 1}{m_a}\right]$

$$H_1 = \frac{353H\left[\frac{1}{T_a} - \frac{1}{T_g} \times \frac{m_a + 1}{m_a}\right]}{1.293 \times 273 \times \left(\frac{m_a + 1}{m_a}\right) \times \frac{1}{T_g}}$$

$$= H\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]$$

Chimney diameter:

Assuming that there are no losses, the velocity of gases passing through the chimney is given by the following expression:

$$V = \sqrt{2gH_1}$$

If the pressure loss in the chimney is equivalent to a hot gas column of h^1 metres, then

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$$V = \sqrt{2g(H_1 - h^1)}$$

$$= 4.43\sqrt{H_1 - h^1}$$

$$= 4.43\sqrt{H_1}\sqrt{1 - \frac{h^1}{H_1}}$$

$$= K\sqrt{H_1}$$

where $K = 4.43\sqrt{1 - \frac{h^1}{H_1}}$

$= 0.825$ for brick chimneys

$= 1.1$ for steel chimneys

The mass of gases flowing through any cross section of the chimney is given by the following expression:

$$m_g = \rho_g AV$$

$$= \rho_g \times \frac{\pi}{4} D^2 \times V$$

$$\therefore \quad D = 1.128 \frac{m_g}{\rho_g V}$$

7.4.2 Condition for Maximum Discharge through a Chimney

The chimney draught is more effective when the maximum mass of hot gases is discharged in a given time.

We have velocity of gas through the chimney:

$$V = \sqrt{2gH_1} \quad \text{for } h^1 = \text{O}$$

Substituting for H_1, we have

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$$V = \sqrt{2gH\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]}$$

The density of hot gas is given by the following expression:

$$\rho_g = \frac{p}{RT_g}$$

Mass of gas discharged:

$$m_g = A \times V \times \rho_g$$

$$= A\sqrt{2gH\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]} \times \frac{p}{RT_g}$$

$$= \sqrt{\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]} \times \frac{K}{T_g}$$

$$\text{where } K = \frac{A \times p \times \sqrt{2gH}}{R}$$

The value of m_g is maximum if

$$\frac{dm_g}{dT_g} = 0$$

$$\therefore \quad \frac{d}{dT_g}\sqrt{\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]} \times \frac{K}{T_g} = 0$$

$$\frac{d}{dT_g}\sqrt{\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]} \times \frac{1}{T_g} = 0$$

$$\frac{d}{dT_g}\frac{\left(XT_g - 1\right)^{\frac{1}{2}}}{T_g} = 0$$

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where $$X = \left(\frac{m_a}{m_a + 1}\right) \times \frac{1}{T_a}$$

$$\therefore \quad \frac{d}{dT_g}\left(XT_g - 1\right)^{\frac{1}{2}} \times T_g^{-1} = 0$$

or $$\left(XT_g - 1\right)^{\frac{1}{2}} \times -1 \times T_g^{-2} + T_g^{-1} \times \frac{1}{2} \times \left(XT_g - 1\right)^{-\frac{1}{2}} \times X = 0$$

$$\frac{\left(XT_g - 1\right)^{\frac{1}{2}}}{T_g^2} + \frac{1}{2T_g \times \left(XT_g - 1\right)^{\frac{1}{2}}} = 0$$

or $$\frac{-2\left(XT_g - 1\right) + T_g^2}{2T_g^3 \times \left(XT_g - 1\right)^{\frac{1}{2}}} = 0$$

or $$XT_g = 2XT_g - 2$$

$$\therefore \quad XT_g = 2$$

or $$\frac{T_g}{T_a} = 2\left(\frac{m_a + 1}{m_a}\right)$$

$$\therefore \quad (H_1)_{max} = H\left[\frac{m_a + 1}{m_a} \times \frac{m_a}{m_a + 1}\right]$$

$$(H_1)_{max} = H$$

The draught in millimetre of water column for maximum discharge is given by the following expression:

$$(h_w)_{max} = 353H\left(\frac{1}{T_a} - \frac{1}{2T_a}\right)$$

$$(h_w)_{max} = \frac{176.5H}{T_a} \text{ mm of water}$$

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7.4.3 Chimney Efficiency

It is evident that draught is created at the cost of thermal efficiency of the boiler plant installation since a portion of the heat carried away by the flue gases to produce the required draught could have been used in either heating the air entering the furnace or heating the feed water entering the boiler, which would increase the thermal efficiency of the plant.

If T' = absolute temperature of flue gases leaving the chimney to create the draught of h_w millimetre of water

T'' = absolute temperature of flue gases leaving the chimney in thecase of artificial draught of h_w millimetre of water

C_p = mean specific heat of flue gases

The extra heat carried away by unit mass of flue gas due to higher temperature required to produce the natural draught is

$$= C_p (T' - T'') \; (T' > T'')$$

The draught pressure produced by the natural draught system in height of hot column is

$$H_1 = H\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right] m$$

The maximum energy this head would give 1 kg of flue gas, which is at the expense of extra heat carried away from the boiler plant is

$$= H\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]$$

Efficiency of chimney:

$$\eta_c = \frac{H\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]}{C_p \; (T' - T'')}$$

where $T'' = T_g$

Advantages of chimney draught

(i) External power is not required for producing draught.

(ii) Capital cost is less and maintenance cost is nil.

(iii) Air pollution is less as the chimney discharges the gas at a higher altitude.

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Limitations

(i) The maximum power available for producing natural draught is 10–20 mm of water head.

(ii) Draught decreases with an increase in outside air temperature.

(iii) For producing sufficient draught, the flue gases have to be discharged at high temperature since the heat from flue gases cannot be extracted in the economizer and air preheater. This results in the loss of overall plant efficiency.

(iv) Specific fuel consumption is high because of poor efficiency.

(v) There is no thorough mixing of air and fuel in the combustion chamber due to low air velocity. Therefore, combustion is poor and this results in high specific fuel consumption.

(vi) Chimney is not flexible. Available draught is constant for a particular height of chimney.

Let m_a = mass of air in kilogram required per kilogram of fuel.

Example 7.1

A chimney is 28 m high and the temperature of hot gases inside the chimney is 320°C. The temperature of outside air is 23°C and furnace is supplied with 15 kg of coal burnt. Calculate the following:

(i) Draught in millimetre of water

(ii) Draught head in metres of hot gases

Solution:

(i)

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g} \times \frac{m_a + 1}{m_a}\right]$$

$$= 353 \times 28\left[\frac{1}{296} - \frac{1}{593} \times \frac{15 + 1}{15}\right]$$

$$= 15.61 \text{ mm of water}$$

(ii)

$$H_1 = H\left[\left(\frac{T_g}{T_a} - 1\right) \times \frac{m_a}{m_a + 1}\right]$$

$$= 28 \times \left[\left(\frac{593}{296} - 1\right) \times \frac{15}{15 + 1}\right]$$

$$= 26.34 \text{ m}$$

Example 7.2

Determine the height of a chimney to produce a static draught of 22 mm of water if the mean flue gas temperature in a chimney is 290°C and ambient temperature in boiler house is 20°C.

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The gas constant for air is 29.26 kgm/kgK and for chimney flue gas is 26.2 kgfm/kgK. Assume barometer reading as 760 mm of mercury.

Solution: Density of air at 290 K, $\rho_a = \frac{p}{RT} = \frac{1.033 \times 10^4}{29.26 \times 290}$

$$= 1.217 \text{ kg/m}^3$$

Density of flue gas at 563 K, $\rho_g = \frac{1.033 \times 10^4}{26.2 \times 563}$

$$= 0.7 \text{ kg/m}^3$$

Static draught, $p = H\left(r_a - r_g\right)$

$$22 = H(1.217 - 0.7)$$

$$H = 42.55 \text{ m}$$

Example 7.3

Determine the height of a chimney to get net draught of 12 mm if the total draft losses are 4 mm. The temperatures of air and chimney gases are 25°C and 300°C, respectively. The mass of air used per kilogram of fuel used is 18 kg; 1 kg of air occupies a volume of 0.7734 m^3 at normal temperature and pressure or NTP.

Solution: Density of air at NTP $= \frac{1}{0.7734}$

$$= 1.293 \text{ kg/m}^3$$

Density of air at 298 K, $\rho_a = 1.293 \times \frac{273}{298}$

$$= 1.1845 \text{ kg/m}^3$$

Density of gases at 573 K, $\rho_g = 1.293 \times \frac{18+1}{18} \times \frac{273}{573}$

$$= 0.65 \text{ kg/m}^3$$

But $p = 12 + 4$

$$= 16 \text{ mm of water}$$

$$p = H(\rho_a - \rho_g)$$

i.e. $16 = H(1.1845 - 0.65)$

$$H = 29.93 \text{ m}$$

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Example 7.4

Fifteen kilogram of air supplied per kilogram of fuel was burnt to the combustion chamber of a boiler using fuel 600 kg/hr. The temperatures of flue gases and ambient air are 273°C and 32°C, respectively. If the minimum draught required to start the flue is 9.5 mm of water, find out the minimum height of the chimney.

Solution: Temperature of fuel gases $T_g = 237 + 273 = 510$ K

Temperature of ambient air $T_a = 32 + 273 = 305$ K

Draught in mm of water $h_w = 9.5$ mm

Max of air $m_g = 15$ kg

We have

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g} \times \frac{m_a + 1}{m_a}\right]$$

$$9.5 = 353 \times H\left[\frac{1}{305} - \frac{1}{510} \times \frac{15 + 1}{15}\right]$$

$$0.419H = 9.5$$

$$H = 22.66 \text{ m}$$

Example 7.5

Calculate the power of a motor required to drive a fan that produces a draught of 48 mm of water column using (i) ID fan and (ii) FD fan.

The temperatures of flue gases and ambient air temperature in both the cases are 175°C and 30°C, respectively. Air required per kilogram of fuel burnt is 18 kg. If the coal burnt per minute is 25.4 kg and efficiency of the fan is 78 per cent.

Solution: Temperature of gases $T_g = 175 + 273 = 448$ K

Temperature of air $T_a = 30 + 273 = 303$ K

Mass of air used $m_a = 18$ kg

Mass of coal used $M = 25.4$ kg/min $= 1524$ kg/hr

Draught produced by the fan $h_w = 48$ mm of water

Efficiency of the fan $\eta_f = 78\%$

(i) Power of motor required to drive ID fan

$$P_{ID} = \frac{0.998 \times 10^{-8}\, h_w \times V_o m_a M T_g}{h_f}$$

$$= \frac{0.998 \times 10^{-8} \times 48 \times 0.7734 \times 18 \times 25.4 \times 60 \times 448}{0.78}$$

$$P_{ID} = 5.84 \text{ kW}$$

where $V_o = 0.7734$ m^3 at 0°C and 760 mm of H_g

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(ii) Power required to drive FD fan

$$P_{FD} = \frac{0.998 \times 10^{-8}\ h_w V_o m_a M T_a}{h_f}$$

$$= \frac{0.998 \times 10^{-8} \times 48 \times 0.7734 \times 18 \times 1.524 \times 303}{0.78}$$

$$P_{FD} = 3.95 \text{ kW}$$

Example 7.6

A chimney of height 30 m is used for producing a draught of 15 mm of water. The temperatures of flue gases and ambient air are 297°C and 27°C, respectively. The coal burned in the combustion chamber contains 80 per cent carbon and 5 per cent moisture, remaining ash. Neglecting losses and assuming the value of burnt products equivalent to those of air supplied and complete combustion of fuel, find the percentage of excess air supplied.

Solution: Height of the chimney, $H = 30$ m

Draught in millimetre of water, $h_w = 15$ mm

Ambient air temperature, $T_a = 27 + 273 = 300$ K

Temperature of flue gas, $T_g = 297 + 273 = 570$ K

Percentage of carbon in the fuel = 80%

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g} \times \frac{m_a + 1}{m_a}\right] \text{ mm of water}$$

$$15 = 353 \times 30\left[\frac{1}{300} - \frac{1}{570} \times \frac{m_a + 1}{m_a}\right]$$

$$m_a = 10.795 \text{ kg/kg of fuel}$$

We have $C + O_2 = CO_2$

$$12 + 32 = 44$$

1 kg of carbon requires $\frac{32}{12} = \frac{8}{3}$ kg of oxygen and $\frac{8}{3} \times \frac{100}{23} = 11.594$ kg of air.

1 kg of fuel contains only 0.80 carbon.

∴ Air required for complete combustion

$$= 11.594 \times 0.80$$

$$= 9.275 \text{ kg/kg of fuel}$$

∴ Percentage of excess air supplied $= \frac{11.594 - 9.275}{9.275} = 0.25$

$$= 25\%.$$

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7.5 QUESTIONS

7.5.1 Objective Questions

1. For maximum discharge through the chimney, the draught h in millimetre of water is

   (a) $\frac{176.5H}{T}$ (b) $\frac{T}{176.5H}$

   (c) $\frac{176.5T}{H}$ (d) $\frac{353H}{T}$

2. The ratio of height of hot gas column generating draught to the height of the chimney should be (for maximum discharge)

   (a) 1:2 (b) 1:1

   (c) 2:1 (d) 1:4

3. For maximum discharge through a chimney

   (a) $\frac{T_f}{T_a} = \frac{2m}{m+1}$ (b) $\frac{T_f}{T_a} = \frac{2(m+1)}{m}$

   (c) $\frac{T_a}{T_f} = \frac{2(m+1)}{m}$ (d) $\frac{T_f}{T_a} = \frac{2(m-1)}{m}$

   where T_f = temperature of flue gas, T_a = temperature of atmospheric air and m = mass of air supplied/kg of coal.

4. Efficiency of chimney draught is about

   (a) 10 per cent (b) 70 per cent

   (c) 1 per cent or less (d) 90 per cent

5. Steam jet draught is used in

   (a) Locomotive boiler (b) Lancashire boiler

   (c) Benson boiler (d) Cornish boiler

Answers

1. c 2. c 3. b 4. b 5. a

7.5.2 Review Questions

1. Name the various draughts. Describe the operation of balanced draught system with a neat sketch.
2. What are the limitations of chimney draft?
3. Define the chimney efficiency and find out the expressions for the same.
4. What are the various draughts that are used in usual practice?
5. Why are different types of chimneys in practice? Explain the basis of their selection.
6. What are the required conditions for the stability of a chimney?
7. What are the limitations of chimney draught?

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8. What are the different losses that are generally taken into account in designing the draught system.
9. Calculate the quantity of air supplied per kilogram of fuel burnt in the combustion chamber of a boiler when the required draught of 1.89 cm of water is produced by a chimney of 32 m height. The temperatures of the flue gases and ambient air recorded are 380°C and 30°C, respectively.
10. Calculate the power of a motor required to drive a fan that produces a draught of 50 mm of water column using (i) induced draught fan and (ii) forced draught fan. Temperature of flue gases and ambient air temperature in both the cases are 180°C and 30°C, respectively. Air required per kilogram of fuel burnt is 17 kg. The coal burnt per minute is 26 kg and the efficiency of the fan is 80 per cent.
11. A draught of 5 cm of water column is produced by using ID fan. The temperatures of the flue gases and ambient air are 202°C and 37°C, respectively. The consumption of coal is 1600 kg/hr and A:F ratio used is 15:1. Assuming the mechanical and transmission efficiency to be 85 per cent and 88 per cent, find out the power of the prime mover required to run the fan. If the induced draught fan is replaced by forced draught fan, assuming other data to be the same, find out the percentage reduction in the power of the prime mover required to run the fan.

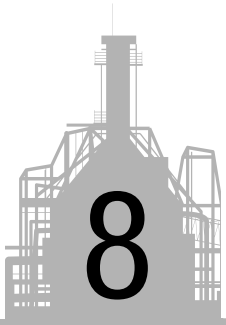
(**Ans:** 8.52 hp, 38.8 per cent)

12. The draught produced by a forced draught fan is sufficient to overcome the total frictional losses of the system at 20 mm of water level and to impact the velocity of 10 m/sec to the flue gases passing through the chimney. The amount of coal burned per hour is 1200 kg and air supplied is 12 kg/kg of coal burned. (i) Assuming the atmospheric pressure of 1.03 bar and 27°C and mechanical efficiency of 80 per cent, find out the power required for a prime mover assuming the mechanical efficiency of 80 per cent. (ii) If the same draught is produced by using ID fan and allowing the flue gases to exhaust at 120°C, find out the power of the prime mover required assuming the same mechanical efficiency.

(**Ans:** 14.75 hp, 20.8 hp)

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Feed Water Treatment



8

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8.1 Feed water system – necessity
8.2 Feed water treatment
8.3 Questions

8.1 FEED WATER SYSTEM – NECESSITY

With the invention of modern high-pressure boilers, it has become a necessity to supply feed water without any impurities. Natural water contains impurities in colloidal or suspension form in addition to dissolved impurities. Major impurities found in colloidal or suspension form include mud, silt, clay, silica, etc., and that found in the dissolved form include carbonates, bicarbonates, phosphates and sulphates of calcium, magnesium, sodium and gases such as CO_2 and O_2.

These impurities when operated at high pressure get deposited on the heat transfer surfaces, thereby reducing the rate of heat transfer. Deposits such as silica result in localized heating of the boiler drum as well as water tubes that may result in premature failure. In addition, the deposits inside the boiler tubes reduce the flow rate of feed water. Hence, it is vital to treat feed water using a feed water system.

In a modern steam-condensing plant, it is very essential to use nearly 3–5 per cent of external water as *make-up* to counterbalance the losses. For a power plant of 500 MW capacity, the make-up water requirement is anywhere between 200 and 225 t/h. Thus, it is a prerequisite to have a separate feed water treatment plant in order to treat impure water taken from the external sources such as river, lake or pond. Figure 8.1 shows a general layout of feed water supply system.

8.1.1 Feed Water Impurities

The impurities present in feed water are mainly classified as follows:

1. *Undissolved and suspended solids*

(i) *Turbidity and sediments*

Turbidity in water is due to suspended soluble matters such as mud, sand, sediments, etc., which rapidly settle down when there is no disturbance. Turbidity of feed water should not exceed 5–10 ppm. These produce hard stony deposits on heating, which cling to the heat transfer surface.

(ii) *Chlorides, sodium and potassium salts*

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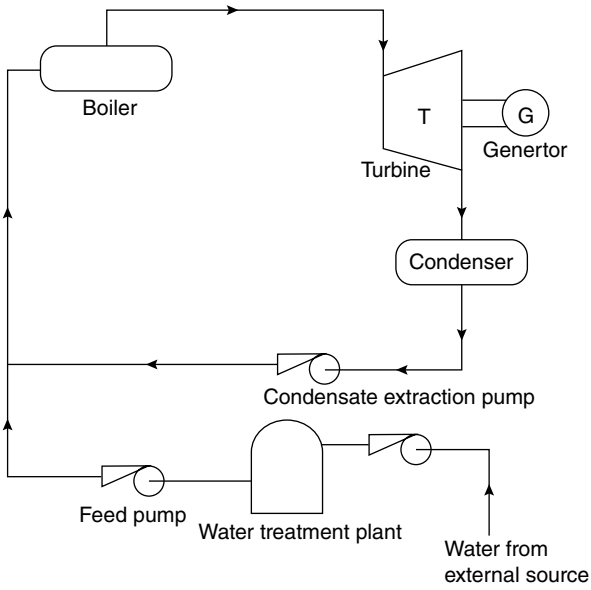


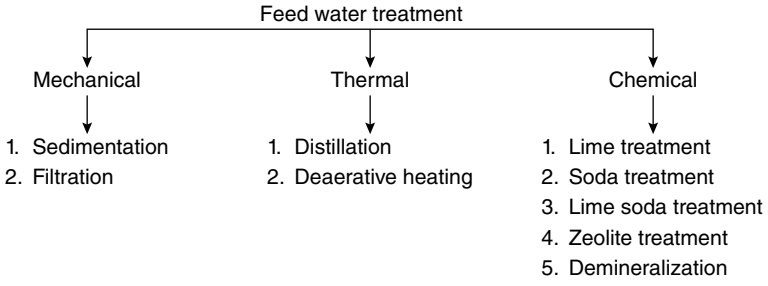
Fig. 8.1 *Feed Water Supply System*

2. *Dissolved solids*

These include carbonates and bicarbonates of calcium and magnesium, namely calcium carbonates ($CaCO_3$), calcium bicarbonate [$Ca(HCO_3)_2$], magnesium carbonate ($MgCO_3$) and magnesium bicarbonate [$Mg(HCO_3)_2$]. Other solids include calcium sulphate ($CaSO_4$); sodium bicarbonate; sodium sulphate; magnesium sulphate; nitrates of calcium and magnesium; chlorides of calcium, sodium and magnesium; silica; iron oxide; and gases such as O_2 and CO_2.

8.2 FEED WATER TREATMENT

Different methods of feed water treatment can be classified as follows:



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8.2.1 Mechanical Methods

1. *Sedimentation*

In this method, water with suspended solid matter is made to either stand still or flow with very low velocity. The solid matter settles down due to gravity and is removed regularly. Clear water is drained from the surface.

2. *Filtration*

This method is useful to remove suspended solid matter that cannot be removed by sedimentation. Here, water is allowed to flow downwards through a fine strainer or porous beds of *graded sand* or *anthrafilt*. By increasing the fineness of the filtering element, maximum suspended impurities are removed. The suspended matter clings to the filter material and clear water drains from the bottom. After many hours of operation, the voids of the filter material are filled with dirt. Hence, filter material must be cleaned periodically by *backwashing*.

To improve the filtering process, coagulants such as *aluminium sulphate* and *sodium aluminate* are used with water. These additives react with alkaline salts in water to produce gelatinous substance that settles down or gets filtered very easily.

Different types of filters are available for mechanical filtration, namely pressure filters, gravity filters, tubular filters, horizontal filters, cartridge-type filters, etc. Figure 8.2 is an example of a typical pressure filter.

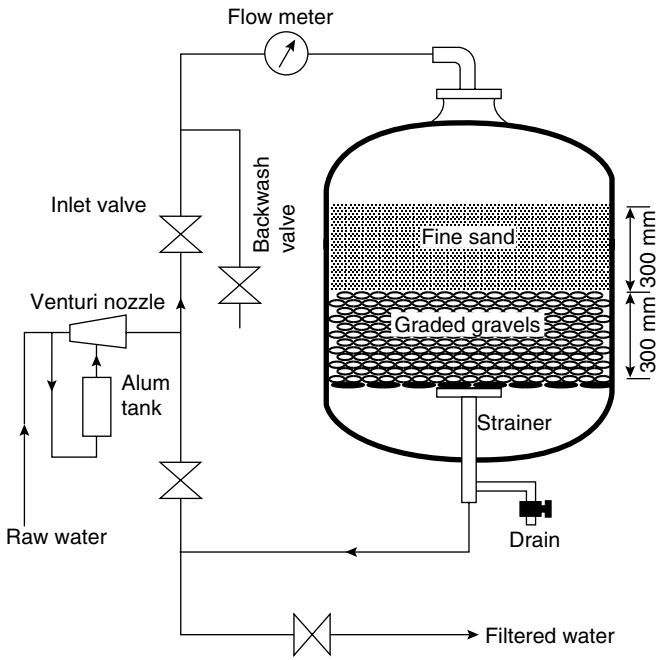


Fig. 8.2 *Pressure Filter*

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8.2.2 Thermal Methods

1. *Distillation*

It is the process of evaporating raw water using steam. Vapours are collected and condensed to supply pure feed water.

2. *Deaerative heating*

Gases such as oxygen, carbon dioxide, air, H_2S, etc., which are in dissolved form in water, can be removed by using deaerative heating method. This is done by heating water between 105 and 110°C added by agitation in a *deaerator*.

Figure 8.3 shows a tray-type deaerator used in modern power plant. The water to be deaerated is passed through a vent condenser for preheating by the gases, and air is liberated from water and steam. Water is then passed through a spray distributor from where it falls in the form of spray over heating trays and air separator trays. Steam enters the deaerator shell through the nozzles sideways and covers the entire shell and the trays. It enters the top plate and flows downwards through perforations and comes in intimate contact with water sprayed from the distributor. The air and gases released leave the shell along with steam to the vent condenser. In the vent condenser, heat transfer takes place between incoming raw water and outgoing hot gases along with steam. Steam gets condensed during the process and gets collected in the shell.

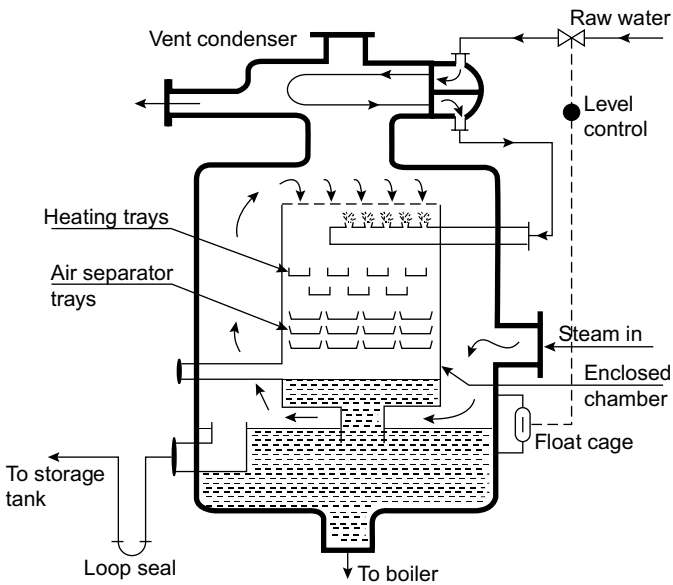


Fig. 8.3 *Tray-Type Deaerator*

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8.2.3 Chemical Methods

1. *Hot lime–soda process*

In this process, *hot lime* (calcium hydroxide) and *soda ash* (sodium carbonate) are used to treat calcium and magnesium salts. The reactions are carried out in the presence of hot water (100°C) heated by steam.

1. The *carbonate hardness* is removed by *hot lime* as follows:

$$Ca(HCO_3) + Ca(OH_2) \rightarrow 2CaCO_3(\downarrow) + 2H_2O$$

Calcium hydroxide

$$Mg(HCO_3)_2 + 2Ca(OH_2) \rightarrow 2CaCO_3(\downarrow) + Mg(OH)_2(\downarrow) + 2H_2O$$

2. The *sulphate hardness* of calcium and magnesium is removed by *soda ash* as follows:

$$MgSO_4 + Na_2CO_3 \rightarrow MgCO_3(\downarrow) + CaCl_2$$

The magnesium carbonate further reacts with lime to form magnesium hydroxide sludge.

$$MgCO_3 + Ca(OH_2) \rightarrow Mg(OH_2)(\downarrow) + CaCO_3(\downarrow)$$

The magnesium oxide formed has coagulating properties and helps in rapid precipitation. It also absorbs soluble silica from the solution. Hence, it is purposely added to water to remove silica.

3. The *chloride hardness* of calcium and magnesium are removed by *soda* and *lime*, respectively, as follows:

$$CaCl_2 + Na_2CO_3 \rightarrow CaCO_3(\downarrow) + 2NaCl$$

$$MgCl_2 + Ca(OH)_2 \rightarrow Mg(OH)_2(\downarrow) + CaCl_2$$

Trisodium phosphate and caustic soda can also be used in addition to lime and soda. Phosphate is very effective in removing calcium and magnesium salts.

$$3Ca(HCO_3)_2 + 6NaOH \rightarrow 3CaCO_3(\downarrow) + 3Na_2CO_3 + 6H_2O$$

$$Mg(HCO_3)_2 + 4NaOH \rightarrow Mg(OH_2)(\downarrow) + 2Na_2CO_3 + 2H_2O$$

$$3CaSO_4 + 2Na_3PO_4 \rightarrow Ca_3(PO_4)_2(\downarrow) + 3Na_2SO_4$$

$$3MgSO_4 + 2Na_3PO_4 \rightarrow Mg_3(PO_4)_2(\downarrow) + 3Na_2SO_4$$

2. *Ion-exchange or zeolite process*

The dissolved impurities in water dissociate to form positively and negatively charged particles known as ions. Some ion-exchange materials have the ability to exchange one ion for another, hold it temporarily in chemical combination and give it up to a strong regenerative solution. One such material is *zeolite*, which is either natural or artificial.

Natural zeolites have a sodium base that can be exchanged for calcium or magnesium in raw water. The most commonly used zeolite is *sodium zeolite*, $Na_2(Al_2Si_2O_3)$, which is denoted by the symbol Na_2Z.

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Figure 8.4 shows a sodium zeolite water softener used for water treatment. It is similar to a pressure filter in construction. The water under pressure enters the softener at the top in the form of spray. It then flows through the zeolite bed, and the hardness of water is removed by ion exchange.

$$CaSO_4 + Na_2Z \rightarrow CaZ + Na_2SO_4$$

$$Ca(HCO_3)_2 + Na_2Z \rightarrow CaZ + 2Na(HCO_3)$$

$$CaCl_2 + Na_2Z \rightarrow CaZ + 2NaCl$$

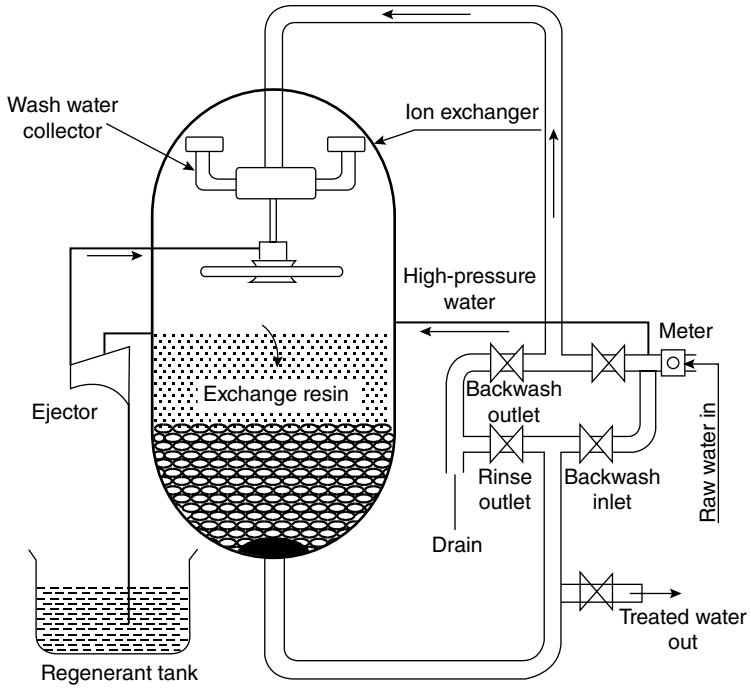


Fig. 8.4 *Sodium Zeolite Softener*

Zeolite reacts with magnesium salts in a similar manner as above. The hard water flowing through the exchanger emerges out as soft water by ion exchange. During this process, zeolite loses sodium and gains calcium. Hence, zeolite should be reactivated by backwashing the bed regularly using brine solution or caustic soda (NaOH). During regeneration, the following reactions take place:

$$CaZ + 2NaCl \rightarrow Na_2Z + CaCl_2$$

$$MgZ + 2NaCl \rightarrow Na_2Z + MgCl_2$$

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After regeneration, the bed is backwashed again by the strong current of water. This process loses and brings the zeolite into a semi-suspended condition in addition to removing dirt collected at the top portion of the bed. After this, a predetermined quantity of NaCl solution is circulated into the bed using ejector to reactivate the bed again. Finally, rinse water is passed through the bed to flush out the remains of calcium, magnesium and brine. Thus, the softener becomes ready for another service.

3. *Demineralizing water treatment*

This type of treatment is the most economical method of supplying mineral-free water to high-pressure boilers. The system consists of a *catexer* (cation exchanger) and an *anexer* (ion exchanger) along with degasifying tower and silica absorber as shown in Figure 8.5.

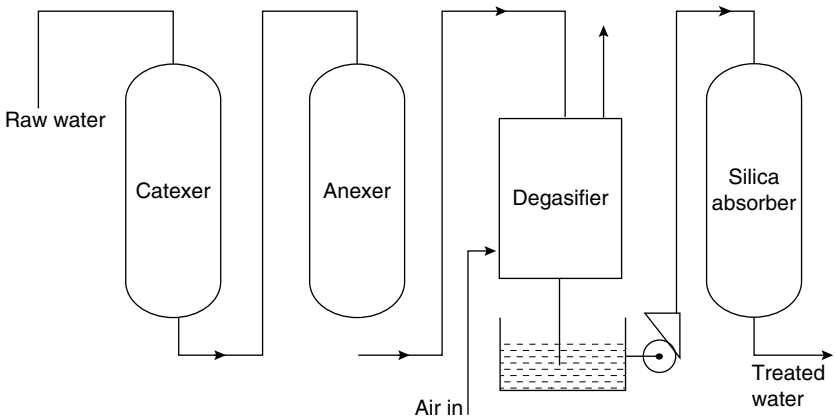


Fig. 8.5 *Demineralization Plant*

Raw water is first passed through the catexer that contains a weak acid to remove bicarbonates. The acidic water that contains weak acids such as carbonic acid, hydrochloric acid and sulphuric acid is then passed through the anexer. The anion exchange resins in the anexer remove anions such as chlorides, sulphates and nitrates from the water in addition to removing acid radicals.

$$2HCl + RCO_3 \text{ (resin)} \rightarrow RCl_2 + H_2CO_3$$

Water free from dissolved solids now enters the degasifier containing porcelain packing. As water flows downwards, it comes in contact with the air blown from a blower. Carbon dioxide is liberated and escapes from the vent. The degassed water is collected in the tank below the degasifier. Water is then pumped to the silica absorber that contains a strong base anion resin. The resin absorbs silica and reduces it to 0.02 ppm.

$$R_4NOH \text{ (resin)} + H_2SiO_3 \rightarrow R_4NHSiO_3 + H_2O$$

After many hours of operation, both catexer and anexer are reactivated. The catexer is reactivated by using an acidic solution, whereas the anexer is reactivated by using either NaOH or Na_2CO_3 as follows:

$$RCl_4 + Na_2CO_3 \rightarrow RCO_3 + 2NaCl$$

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8.3 QUESTIONS

8.3.1 Objective Questions

1. Turbidity in boiler feed water results in
   (a) sludge formation (b) caustic embrittlement
   (c) scale formation (d) foaming
2. Dissolved oxygen in feed water is removed by
   (a) sodium sulphite (b) calcium sulphite
   (c) alum (d) chloride
3. Temporary hardness of boiling water is due to
   (a) carbonates (b) bicarbonates
   (c) sulphates (d) chlorides
4. Dissolved oxygen in feed water results in
   (a) corrosion (b) foaming
   (c) priming (d) none of these
5. Which of the following has least density?
   (a) Silica scale (b) Sodium scale
   (c) Potassium scale (d) Platinum scale
6. Mill saline water can be economically desalted using
   (a) distillation (b) filtration
   (c) electrolysis (d) treatment with alum
7. Which of the following acts as a bleaching agent?
   (a) Chlorine (b) Alum
   (c) Potassium (d) Nitrogen
8. Chlorine acts as a bleaching agent in the pressure of
   (a) air (b) moisture
   (c) sunlight (d) oxygen
9. For removal of permanent hardness of water, which one of the following processes is suitable?
   (a) Distillation (b) Filtration
   (c) Zeolite (d) Sedimentation
10. Blowdown is necessary on boilers because.
    (1) the boiler water level is lowered rapidly in case it accidentally rises too high
    (2) the precipitated sediment or sludge is removed when the boiler is in service
    (3) the concentration of suspended solids in the boiler is controlled
    Which of these statements are correct?
    (a) 1, 2 and 3 (b) 1 and 2
    (c) 3 alone (d) 1 and 3
11. Treatment of boiler feed water for the control of hardness is necessary to prevent
    (a) carryover (b) excessive feed water alkalinity
    (c) waterside scale deposits (d) foaming

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12. Boiler water hardness is increased by
    (a) improper operation of the DC heater
    (b) scale forming salts in the feed water
    (c) zero alkalinity in water
    (d) dissolved gases in water
13. Excess free oxygen in the boiler feed water can result from
    (a) improper operation of the DC heater
    (b) steam leaks through the turbine glands
    (c) improper operation of the gland exhauster
    (d) vapour lock in the boiler feed pump
14. Failure to remove calcium and magnesium from feed water before it reaches the boiler can result in tube
    (a) scaling
    (b) pitting
    (c) sludging
    (d) erosion
15. 0.22 kg of oxygen, dissolved in 226.58 t of water, is a concentration of
    (a) 4.0 ppm
    (b) 8.0 ppm
    (c) 16.0 ppm
    (d) 1 ppm
16. An increase in the concentration of total dissolved solids in boiler water can result from
    (a) routine treatment with phosphates
    (b) dissolved oxygen deaeration
    (c) zero water hardness
    (d) frequent prolonged surface blows

Answers

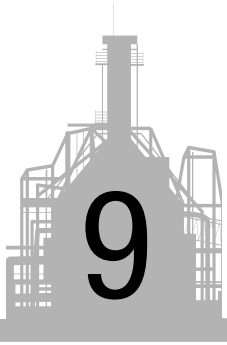
1. a 2. c 3. b 4. a 5. b 6. a 7. a 8. c 9. c 10. a 11. c 12. b
13. a 14. a 15. d 16. a

8.3.2 Review Questions

1. Why is feed water treatment a necessity? Explain.
2. Discuss briefly about the types of impurities in boiler feed water.
3. What are the different methods available for feed water treatment?
4. Sketch and explain the method of filtration used to remove sedimentation.
5. Explain the zeolite process of feed water treatment.
6. How is demineralization process done to treat feed water?

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Flow Through Nozzles



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9.1 INTRODUCTION

Steam nozzle is an essential device used to generate high kinetic energy. This energy is used for different purposes, namely in injectors to pump feed water into boilers, to maintain vacuum in condensers and also in turbines to run rotors. In steam turbines, the nozzle plays an important role by converting enthalpy of steam into kinetic energy. This chapter deals with steam nozzles and steam injectors. The basic governing equation for nozzles is discussed along with the condition for maximum discharge. The concept of supersaturated flow is also discussed in greater detail.

9.2 TYPES OF STEAM NOZZLES

High-pressure, high-temperature steam from the boiler entering the nozzle expands to low-pressure, low-temperature steam gaining kinetic energy. This high-velocity jet of steam is used to run the turbine. As steam expands in the nozzle its velocity and specific volume both will increase. The dryness fraction will also change due to condensation of steam. As the mass flow rate through the nozzle is constant at any given cross section, the cross section of the nozzle varies according to velocity, specific volume and dryness fraction.

Depending on the cross section of the nozzle, steam nozzles are classified as follows:

1. *Convergent nozzle*

If the cross section of the nozzle decreases continuously from the entrance to exit, the nozzle is known as *convergent nozzle*. A convergent nozzle is shown in Figure 9.1(a).

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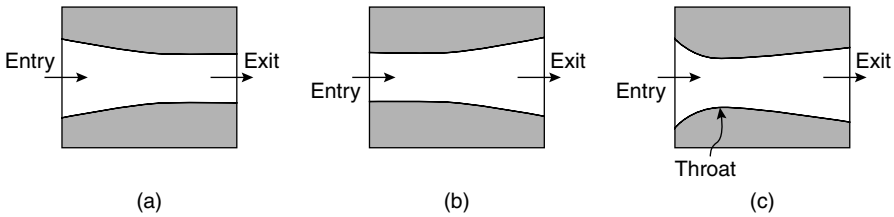


Fig. 9.1 *Types of Nozzles*

2. *Divergent nozzle*

If the cross section of the nozzle increases from entrance to exit, such a nozzle is called as *divergent nozzle*. Figure 9.1(b) shows a divergent nozzle.

3. *Convergent divergent nozzle*

If the cross section of the nozzle first decreases and then increases, such a nozzle is called as *convergent divergent nozzle*. The divergent portion of the nozzle increases the pressure ratio beyond the *critical value*. The area of minimum cross section at which the convergent section ends and divergent section begins is known as *throat* of the nozzle.

Area of application: It may be noted that in a typical turbine application, converging or subsonic nozzles are generally used. In particular, two available types of nozzles are reamed or foiled type.

Reamed or round nozzles: These nozzles are used in high-pressure impulse stages of steam turbines. These nozzles come with low cost and are characterized by better adaptability to standardization with sharp convergent part and a rounded entrance. However, they possess lower efficiency and underutilize the area of flow annulus. The lengths of the nozzles are often large, the divergence angle being 12°–15°.

Foil nozzles: These nozzles are characterized by aerofoil section with sharp rounded edges and short rounded exit, enabling a better issuing jet. They are costly to manufacture but have higher efficiency. They are used in large steam turbines.

9.3 FLOW OF STEAM THROUGH NOZZLE

Consider the flow of steam through a convergent nozzle between sections 1 and 2 as shown in Figure 9.2.

Let

h_1, h_2 = Enthalpy/kg of steam at sections 1 and 2, respectively, kJ/kg

V_1, V_2 = Velocity of steam at sections 1 and 2, respectively, m/s

w = Work done/kg of steam

q = Heat transferred/kg of steam

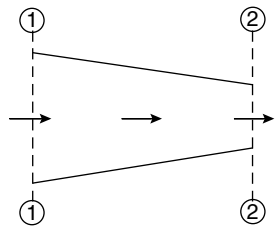


Fig. 9.2 *Flow Through a Nozzle*

Applying steady flow energy equation at sections 1 and 2, we have

$$h_1 + \frac{V_1^2}{2} + w = h_2 + \frac{V_2^2}{2} + q \qquad (1)$$

For nozzles, expansion is assumed as isentropic.

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Hence, $w = 0$ and $q = 0$. This means nozzle is neither a work-absorbing nor a work-generating device.

Equation (1) is now modified to

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$\therefore \quad \frac{V_2^2}{2} = (h_1 - h_2) + \frac{V_1^2}{2}$$

$$\therefore \quad V_2 = \sqrt{2(h_1 - h_2) + \frac{V_1^2}{2}} \tag{2}$$

The above equation gives *exit velocity* of steam through the nozzle.

But usually *the velocity of steam entering the nozzle is negligible compared with exit velocity*.

Hence, ignoring V_1 in the above Equation (2), we have

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2 \times \text{Isentropic enthalpy drop}} \tag{3}$$

$$= 44.72\sqrt{(h_1 - h_2)}$$

where h_1 and h_2 are in kJ/kg.

Equation (3) gives the general expression for the exit velocity of steam from the nozzle irrespective of its shape.

Note: Expansion through the nozzle is neither free expansion nor throttling. This is evident from Equation (2). When steam expands through the nozzle, the loss in enthalpy is gained in the form of kinetic energy.

Exit velocity through the nozzle if the expansion is polytropic can be determined as follows:

Let us assume that expansion follows the law pv^n = constant, where n = polytropic index

Work done during the flow process, if the expansion is polytropic, is given by

$$W = \frac{n}{n-1}(p_1v_1 - p_2v_2) \tag{4}$$

For a nozzle, this work is equal to work done during expansion, which in turn is equal to change in kinetic energy.

$$\therefore \quad \frac{V_2^2}{2} - \frac{V_1^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2) \tag{5}$$

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Ignoring the velocity of steam at entry, Equation (5) modifies to

$$\frac{V_2^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2) \tag{6}$$

According to polytropic law

$$p_1v_1^n = p_2v_2^n$$

or

$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} \tag{7}$$

Substituting (7) in (6)

$$\frac{V_2^2}{2} = \frac{n}{n-1}p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right)$$

$$= \frac{n}{n-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$V_2 = \sqrt{2\frac{n}{n-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]} \tag{8}$$

9.4 DISCHARGE THROUGH NOZZLE AND CRITICAL PRESSURE RATIO

Nozzles should always be designed for maximum discharge. Discharge through the nozzle is determined by applying *continuity equation*.

Let
m = Mass flow rate of steam through the nozzle, kg/s
A_2 = Cross-section area at the nozzle exit, m^2
V_2 = Velocity at the nozzle exit, m/s
v_2 = Specific volume of steam at nozzle exit, m^3/kg

Mass flow rate or *discharge* through the nozzle is given by

$$m = \frac{A_2V_2}{v_2} \tag{1}$$

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If expansion is polytropic, then exit velocity

$$V_2 = \sqrt{2\frac{n}{n-1}p_1v_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

Substituting r_p = pressure ratio = $\frac{p_2}{p_1}$, we have

$$V_2 = \sqrt{2\frac{n}{n-1}p_1v_1\left[1-\left(r_p\right)^{\frac{n-1}{n}}\right]} \tag{2}$$

Substituting (2) in (1), we have

$$m = \frac{A_2}{v_2}\times\sqrt{2\frac{n}{n-1}p_1v_1\left[1-\left(r_p\right)^{\frac{n-1}{n}}\right]} \tag{3}$$

But $p_1v_1^n = p_2v_2^n$

$$\therefore \quad \frac{1}{v_2} = \frac{1}{v_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = \frac{1}{v_1}\left(r_p\right)^{\frac{1}{n}} \tag{4}$$

Substituting (4) in (3)

$$m = \frac{A_2}{v_1}\times r_p^{\frac{1}{n}}\times\sqrt{2\frac{n}{n-1}p_1v_1\left[1-\left(r_p\right)^{\frac{n-1}{n}}\right]}$$

$$m = \frac{A_2}{v_1}\times\sqrt{2\frac{n}{n-1}p_1v_1\left[r_p^{\frac{2}{n}}-\left(r_p\right)^{\frac{n+1}{n}}\right]} \tag{5}$$

(i) *Optimum pressure ratio or critical pressure ratio*

The above equation gives the discharge of steam through a convergent nozzle. It can be observed that discharge through the nozzle mainly depends on the pressure ratio for a given value of n. Thus, for maximum discharge through the nozzle

$$\frac{dm}{dr_p} = 0$$

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$$\frac{d}{dr_p}\left(\frac{A_2}{v_1}\times\sqrt{2\frac{n}{n-1}p_1v_1\left[r_p^{\frac{2}{n}}-\left(r_p\right)^{\frac{n+1}{n}}\right]}\right)=0$$

or $$\frac{d}{dr_p}\left(\left[r_p^{\frac{2}{n}}-\left(r_p\right)^{\frac{n+1}{n}}\right]\right)=0$$

$\therefore$ $$\frac{2}{n}\times r_p^{\frac{2}{n}-1}-\frac{n+1}{n}\left(r_p\right)^{\frac{n+1}{n}-1}=0$$

or $$\frac{2}{n}\times r_p^{\frac{2-n}{n}-}=\frac{n+1}{n}\left(r_p\right)^{\frac{n+1-n}{n}}=0$$

$$\frac{2}{n+1}=r_p^{\frac{1}{n}-\frac{2+n}{n}}$$

Or $$r_p^{\frac{n-1}{n}}=\frac{2}{n+1}$$

$$r_p=\frac{p_2}{p_1}=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}=r_{pc} \tag{6}$$

This equation gives the optimum pressure ratio for maximum discharge. It is also known as *critical pressure ratio*.

Note:

For saturated steam

$$n=1.135$$

$\therefore$ $$r_p=\frac{p_2}{p_1}=0.58$$

For superheated steam

$$n=1.3$$

$\therefore$ $$r_p=\frac{p_2}{p_1}=0.546$$

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(ii) *Maximum discharge*

The maximum mass flow rate from the nozzle occurs at critical pressure ratio.

Substituting Equation (6) in Equation (5), we have

$$(m)_{\max} = \frac{A_2}{v_1} \times \sqrt{2\frac{n}{n-1} p_1 v_1 \left[r_p^{\frac{2}{n}} - \left(r_p\right)^{\frac{n+1}{n}} \right]}$$

$$= \frac{A_2}{v_1} \times \sqrt{2\frac{n}{n-1} \times \frac{p_1}{v_1} \left[\frac{2}{n+1}^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \right]}$$

$$= A_2 \times \sqrt{2\frac{n}{n-1} \times \frac{p_1}{v_1} \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1}\right)^{-1} - 1 \right]}$$

$$= A_2 \times \sqrt{2\frac{n}{n-1} \times \frac{p_1}{v_1} \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left[\left(\frac{n-1}{2}\right) \right]}$$

$$(m)_{\max} = A_2 \times \sqrt{n \times \frac{p_1}{v_1} \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

$$(m)_{\max} = A_2 \times \sqrt{n \times \frac{p_1}{v_1} \times \left(r_{pc}\right)^{\frac{n+1}{n}}} \tag{7}$$

(iii) *Maximum velocity*

Maximum velocity through the nozzle for maximum discharge can be determined as follows:

Maximum velocity

$$V_2 = \sqrt{2\frac{n}{n-1} p_1 v_1 \left[1 - \left(r_p\right)^{\frac{n-1}{n}} \right]}$$

$$(V_2)_{\max} = \sqrt{2\frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \right]^{\frac{n-1}{n}}}$$

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$$= \sqrt{2\frac{n}{n-1}p_1v_1\left[1-\left(\frac{2}{n+1}\right)\right]}$$

$$(V_2)_{max} = \sqrt{2\frac{n}{n+1}p_1v_1} \tag{8}$$

Figure 9.3 shows the variation of mass flow, rate, steam velocity and specific volume with back pressure for a convergent–divergent nozzle.

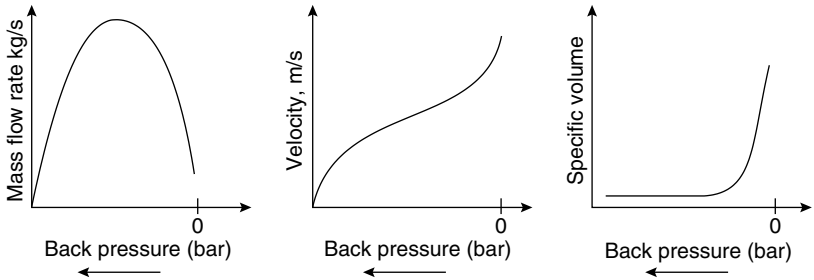


Fig. 9.3 *Variation of m, V and v with Respect to Back Pressure*

9.5 EFFECT OF FRICTION AND NOZZLE EFFICIENCY

When steam expands in the nozzle, steam pressure drops on account of the following reasons:

1. *Friction* existing between nozzle surface and steam and also internal fluid friction in steam.
2. Losses due to shock.

Due to friction the exit velocity of steam reduces, and both dryness fraction and specific volume increase as evident from Figure 9.4.

In Figure 9.4, line 1–2–3 shows isentropic expansion process through a nozzle without friction. But due to friction expansion follows as shown in line 1–2–3′. In case of a convergent–divergent nozzle, maximum friction loss occurs between the throat and exit. Thus

Isentropic enthalpy drop = $h_1 - h_3$

Actual enthalpy drop due to friction = $h_1 - h_{3'}$

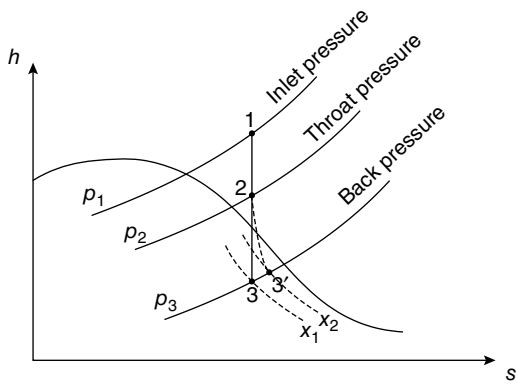


Fig. 9.4 *Effect of Friction During Flow Through a Nozzle*

It is evident that $(h_1 - h_{3'}) < h_1 - h_3$, *which means that there is reduction in enthalpy drop under the influence of friction*. Hence, *there will be subsequent reduction in exit velocity of steam*.

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Further, due to friction, the final state of steam is corresponding to point 3′ instead of point 3. But, dryness fraction of steam at 3′ is higher than dryness fraction of steam at 3. Thus, under the *influence of friction steam becomes more wet*.

Similarly, the specific volume of steam at point 3′ is higher compared with specific volume of steam at 3. Hence, *specific volume of steam also increases due to friction*.

We can summarize the effect of friction as follows:

(i) The expansion is not isentropic

(ii) Enthalpy drop is reduced, which results in reduced exit velocity

(iii) Final dryness fraction of steam increases, i.e. steam becomes more wet

(iv) Specific volume of steam increases

Due to friction efficiency of the nozzle is reduced. Nozzle efficiency is given by

$$\eta_n = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

$$\eta_n = \frac{h_1 - h_3}{h_1 - h_3}$$

Exit velocity of steam from the nozzle can be modified to

$$V_2 = \sqrt{2 \times h_n \times \Delta h_i}$$

where $(\Delta h)_i$ = Isentropic enthalpy drop.

9.6 SUPERSATURATED FLOW

Consider the isentropic expansion of superheated steam in a nozzle from pressure p_1 to pressure p_b as shown in Figure 9.5 by the line 1–2–3. Under normal condition, condensation of steam must begin at point 2 (vapour phase changes to liquid phase below the saturation curve). But due to the high velocity of steam passing through the nozzle at sonic or supersonic speed, time available for condensation at point 2 is only a fraction of a second (about 0.001 s). Because of this, steam expands further in its vapour phase beyond the saturation curve corresponding to pressure p_3 and point 2′ by an amount 2–2′. Thus, steam remains dry even along the line passing through point 2′. This line, which is a *limit to the supersaturated state, is known as Wilson line*.

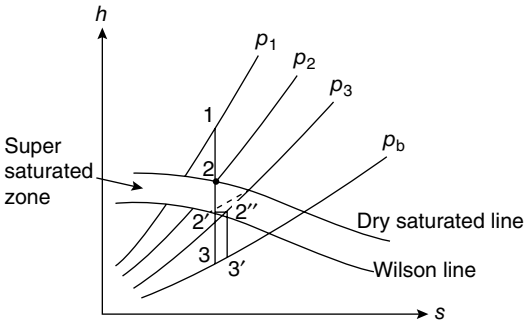


Fig. 9.5 *Metastable Flow*

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The vapour between pressures p_2 and p_3 is said to be supersaturated or super cooled. This is because between points 2 and 2′ corresponding to pressures p_2 and p_3, respectively, the temperature of vapour is lesser than the corresponding saturation temperature. The difference between these temperatures is known as *degree of undercooling* and the flow is known as *supersaturated* flow or *metastable* flow.

After reaching the Wilson line corresponding to point 2′, partial condensation begins at constant enthalpy along the line 2′–2″. Due to partial condensation of steam, heat is released which further raises the steam temperature to saturation temperature. Further, steam expands isentropically to back pressure along 2″–3′.

For supersaturated flow, the following relations are used for calculation purpose:

1. *Exit velocity of steam at the end of expansion*

$$V_2 = \sqrt{2\frac{n}{n-1}p_1v_1\left[1-\left(r_p\right)^{\frac{n-1}{n}}\right]}$$

2. *Specific volume*

$$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$$

3. *Apparent temperature*

$$T_2 = T_1\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}$$

9.7 RELATION BETWEEN AREA, VELOCITY AND PRESSURE IN NOZZLE FLOW

Consider flow through a nozzle as shown in Figure 9.6. Let us further consider two sections 1–1 and 2–2 over a small distance δx. Assuming steady, isentropic and uniform flow across any cross section, we have

At section 1–1, $m = \dfrac{A \times V}{v}$

At section 2–2, $m = \dfrac{(A+\delta A)\times(V+\delta V)}{(v+\delta v)}$

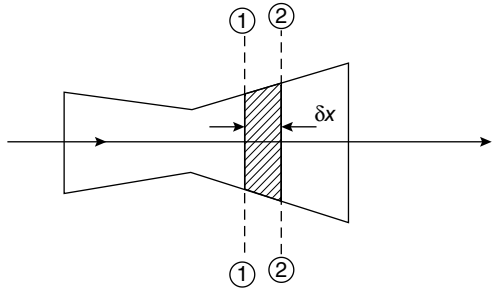


Fig. 9.6 *Variation of Area, Velocity and Volume in a Nozzle*

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where δA, δV and δv are small increments in area, velocity and specific volume between the two sections, respectively.

Using continuity equation,

$$m = \frac{A \times V}{v} = \frac{(A + \delta A) \times (V + \delta V)}{(v + \delta v)}$$

On simplifying

$$\frac{\delta A}{A} + \frac{\delta V}{V} - \frac{\delta v}{v} = 0$$

or between the limits

$$\frac{dA}{A} + \frac{dV}{V} - \frac{dv}{v} = 0 \tag{1}$$

For an isentropic flow

$$pv\gamma = \text{constant} = C$$

$$\therefore \quad \ln p + \gamma \ln v = \ln C \tag{2}$$

Differentiating Equation (2) and dividing throughout by pv

$$\frac{dp}{p} + \gamma \frac{dv}{v} = 0$$

or

$$\frac{dv}{v} = \frac{1}{\gamma} \times \frac{dp}{p}$$

Applying steady flow energy equation between the two sections, we have

$$\delta q = dh + d\frac{V^2}{2} + \delta w$$

For a nozzle $\delta q = 0$ and $\delta w = 0$

$$\therefore \quad dh + d\frac{V^2}{2} = 0$$

But $dh = Tds + vdp = vdp$ (for isentropic flow)

Substituting in the above equation

$$d\frac{V^2}{2} = -vdp$$

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or $$VdV = -vdp$$

or $$\frac{dV}{V} = -\frac{v}{V^2}dp = -\frac{dp}{\rho V^2}$$

From Equation (1)

$$\frac{dA}{A} = \frac{dv}{v} - \frac{dV}{V} = -\frac{1}{\gamma}\frac{dp}{p} + \frac{vdp}{V^2}$$

$$\frac{dA}{A} = \frac{1}{\gamma} \times \frac{dp}{p}\left(\frac{\gamma pv}{V^2} - 1\right)$$

$$= \frac{1}{\gamma} \times \frac{dp}{p}\left(\frac{C^2}{V^2} - 1\right)$$

$$\frac{dA}{A} = \frac{1}{\gamma} \times \frac{dp}{p}\left(\frac{1}{M^2} - 1\right) \quad (3)$$

where $C = \sqrt{\gamma pv} = \sqrt{\gamma RT}$ = Sonic velocity

$$M = \frac{V}{C} = \text{Mach number}$$

1. *Accelerated flow*

For accelerated flow dp is negative in Equation (3), which means pressure decreases along the flow direction (Figure 9.7).

(i) When $V < C$ or $M < 1$, dA is negative; nozzle is convergent and flow is subsonic.

(ii) When $V > C$ or $M > 1$, dA is positive; nozzle is divergent and flow is supersonic.

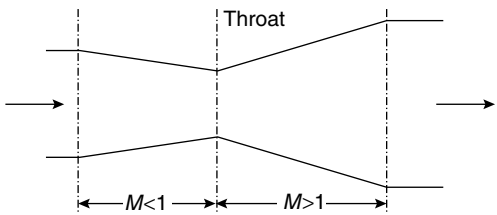


Fig. 9.7 *Accelerated Flow (Nozzle)*

2. *Decelerated flow (diffuser)*

For decelerated flow dp is positive in Equation (3), which means pressure increases along the flow direction (Figure 9.8).

(i) When $V < C$ or $M < 1$, dA is positive; diffuser is converging and velocity is supersonic.

(ii) When $V > C$ or $M > 1$, dA is negative; diffuser is diverging and velocity is subsonic.

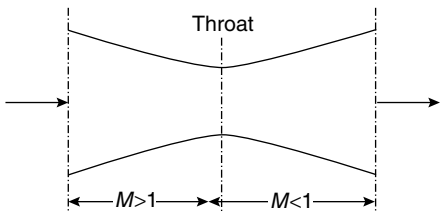


Fig. 9.8 *Decelerated Flow (Diffuser)*

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It may be noted that Equation (3) may also be reduced to the following form:

$$\frac{dA}{A} = -\frac{dV}{V}\left(1 - M^2\right) \tag{4}$$

Note:

1. At subsonic speeds ($M < 1$), a decrease in area increases the speed of flow. Hence, a subsonic nozzle assumes a convergent profile and a subsonic diffuser assumes a divergent profile. The flow behaviour therefore qualitatively remains the same as in incompressible flows in the region of $M < 1$.

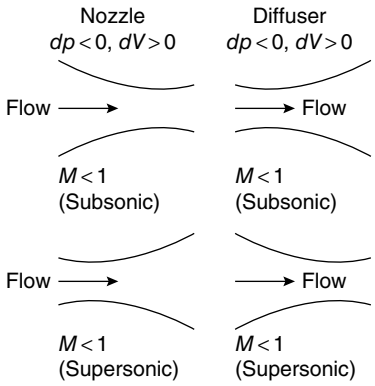


Fig. 9.9 *Arrangement of Nozzles and Diffusers*

2. At supersonic speeds ($M > 1$), the effect of area changes are different. According to Equation (4), a supersonic nozzle must assume a divergent profile with an increasing area in the flow direction, whereas a supersonic diffuser must be a converging channel. Divergent nozzles are used in missiles and launch vehicles to generate supersonic flow.
3. Figure 9.9 shows the arrangement of nozzles and diffusers to generate the required flow based on Mach Number, M.

9.8 CHARACTERISTICS OF CONVERGING–DIVERGING NOZZLE

Consider a converging–diverging nozzle (De Laval nozzle) as shown in Figure 9.10.

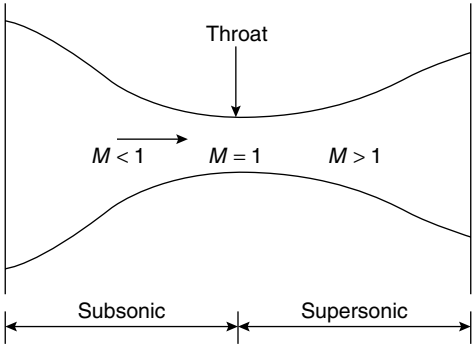


Fig. 9.10 *Converging–Diverging Nozzle*

Let subscript 0 represent stagnation condition and let us assume that the stagnation enthalpy and pressure at the inlet are known. The pressure at the nozzle exit is p_E, back pressure is p_B and critical pressure is p^*.

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Before making the analysis, let us see how pressure distribution occurs in a converging nozzle as shown in Figure 9.11.

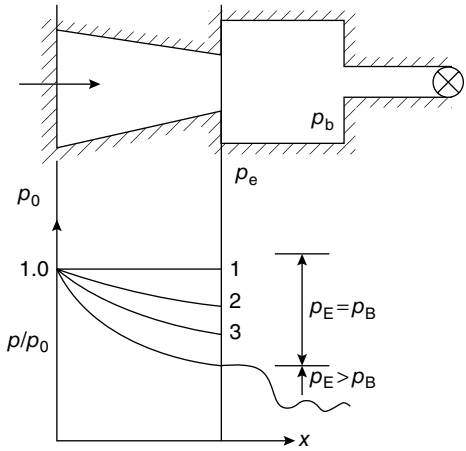


Fig. 9.11 *Pressure Distribution in a Convergent Nozzle*

When the back pressure is varied by operating the valve, the mass flow rate and the exit plane pressure p_B/p_0 also varies as shown in Figure 9.12.

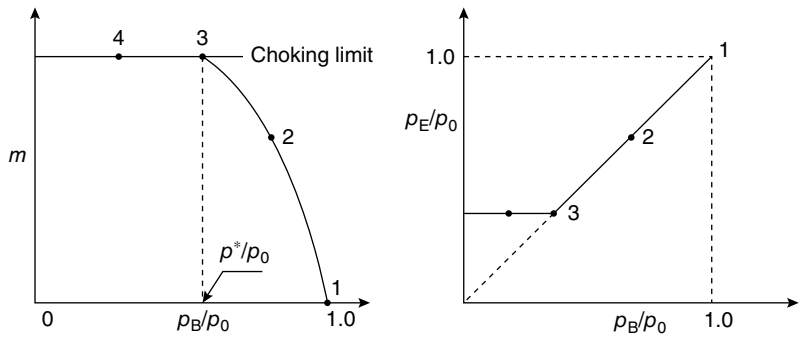


Fig. 9.12 *Variation of Mass Flow Rate and Exit Pressure versus Back Pressure in a Convergent Nozzle*

(i) When $p_B/p_0 = 1$, there is no flow and $p_E/p_0 = 1$, as shown by point 1.

(ii) By decreasing the pressure to point 2 such that $p_B/p_0 \rangle p^*/p_0$, then $p_E = p_B$ and $M_E \langle 1$, flow is subsonic.

(iii) By decreasing the pressure to point 3(critical pressure) such that $p_E = p_B$, the Mach number $M_E = 1$, flow is sonic.

(iv) By decreasing the pressure to point 4 (below the critical pressure) such that $p_E = p^*$ and the Mach number $M_E = 1$, there will be no further increase in mass flow rate. The drop in pressure from p_E to p_B occurs outside the nozzle exit and this limit is known as choking limit.

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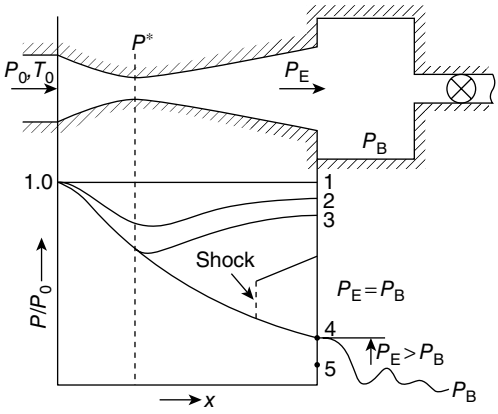


Fig. 9.13 *Pressure Distribution in a Convergent–Divergent Nozzle*

Choking limit refers to the maximum possible mass flow through the nozzle irrespective of further decrease in exit pressure.

Let us now consider a convergent–divergent nozzle to understand the pressure distribution and the phenomenon of shock as shown in Figure 9.13.

In case of a convergent–divergent nozzle

(i) When $p_B/p_0 = 1$, there is no flow as shown by curve 1.

(ii) By decreasing the pressure to point 2 such that $p_B/p_0 < 1$, but $p_B/p_0 > p^*/p_0$, the velocity in the converging section while $M < 1$ at the throat. In the diverging section, which acts as a subsonic diffuser, pressure increases and velocity decreases.

(iii) By decreasing the pressure to point 3, flow becomes sonic at the throat (Mach number $M = 1$), and diverging portion still functions as a diffuser, reducing the flow speed from sonic to subsonic.

(iv) By decreasing the exit pressure below the point as specified by curve 4, such that $p_E = p_B$, the diverging section acts as a supersonic nozzle as pressure reduces and velocity increases continuously. The flow is isentropic and the pressure ratio corresponds to designed conditions.

(v) By decreasing the pressure ratio further to point 5, there will be no further decrease in exit pressure. The drop in pressure from p_E to p_B occurs outside the nozzle exit.

The flow is not isentropic between pressures corresponding to curves 3 and 4 in the diverging section. This is due to the phenomenon known as *shock where the flow that starts accelerating downstream of the throat suddenly decelerates at a certain section through a nearly normal surface of discontinuity*. At supersonic speeds, shocks occur and subsequently flow becomes subsonic; after that the rest of the duct acts like a diffuser. By gradually lowering the exit pressure p_E, the position of shock moves down the nozzle and finally disappears at the exit end, when p_E reaches a value corresponding to point 4.

Note: When $M = 1$ at the throat, the discharge is maximum and the nozzle is said to be *choked*. At this point, the nozzle is incapable of allowing more discharge even if the exhaust pressure is reduced further. Hence, the discharge corresponding to this condition is also known as *critical discharge*.

9.9 STEAM INJECTOR

A steam injector is a device used to supply feed water at high pressure to the boiler. It works similar to a steam nozzle. Figure 9.14 shows the working of a steam injector. High-pressure steam from the boiler passes through a convergent nozzle. The high-velocity steam coming from the convergent nozzle mixes with the feed water coming from the feed tank in the combined nozzle.

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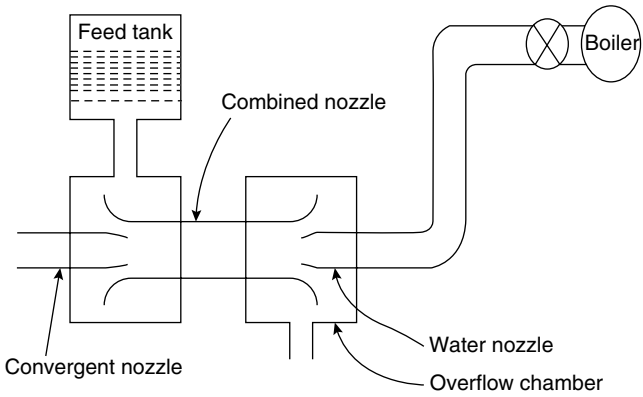


Fig. 9.14 *Steam Injector*

The kinetic energy of the mixture is converted into pressure energy in the divergent portion of the combined nozzle. High-pressure water then enters the water nozzle from where it is delivered to the boiler at boiler pressure. Water, which overflows during starting, is collected in an overflow chamber and then discharged to the drain.

Example 9.1

Find the velocity of steam issuing from a nozzle in which the steam has been expanded in an initial condition of 10.3 bar absolute dry to a pressure of 0.686 bar (i) neglecting friction and (ii) considering frictional loss as 10 per cent of the heat drop.

Solution: From Mollier chart

Heat drop = 456 kJ/kg = $(h_1 - h_2)$

Velocity $V_2 = 44.72\sqrt{(h_1 - h_2)}$

$$V_2 = 44.72\sqrt{456}$$

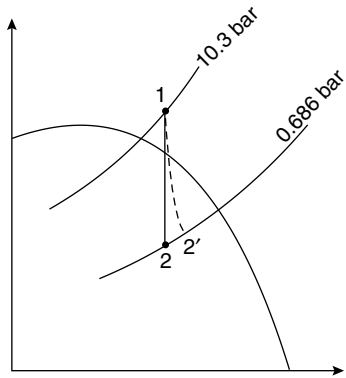
$$= 954.96 \text{ m/s}$$

$\eta_{\text{nozzle}} = 90\%$

$\therefore$ $$V_2 = 44.72\sqrt{\eta_n (h_1 - h_2)}$$

$$= 44.72\sqrt{0.9 \times 456}$$

$$= 905.95 \text{ m/s}.$$



$\therefore$ Diameter of nozzle mouth

$$d = \sqrt{\frac{4 \times 8.757}{\pi}}$$

$$d = 3.34 \text{ cm}.$$

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Example 9.2

Steam at a pressure of 6.85 bar and 0.9 dry expands through a nozzle having a throat area of 4.65 cm^2. The back pressure is 1.03 bar. Determine (i) the mass of steam flowing per minute; (ii) the diameter of the mouth of nozzle for maximum discharge; and (iii) the final velocity of the steam.

Solution: Since the steam is saturated, for the maximum discharge, theoretical pressure at the throat of nozzle will be

$$p_2 = 0.58 p_1$$
$$= 0.58 \times 6.85$$
$$= 3.97 \text{ bar}$$

From Mollier chart, the heat drop from the inlet of nozzle to the nozzle throat is

$$h_1 = h_2 = 92.95 \text{ kJ}$$

Dryness fraction

$$x_2 = 0.872$$

∴ Velocity of steam at throat

$$V_2 = 44.72\sqrt{92.95}$$
$$V_2 = 431 \text{ m/s}$$

From steam tables, the specific volume of steam at nozzle throat is

$$V_2 = 0.463 \text{ m}^3/\text{kg}$$

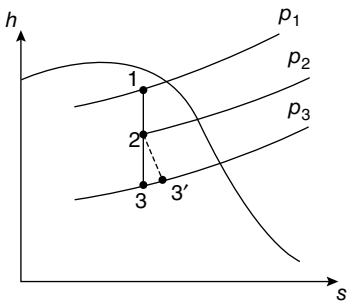
(i) Mass flow rate, $m = \dfrac{A_2 V_2}{x_2 v_2} = \dfrac{4.65 \times 431}{0.872 \times 0.463 \times 10^4}$

$$= 0.496 \text{ kg/s}.$$

(ii) Again from Molliar chart, heat drop from the inlet to the mouth of nozzle is

$$h_1 = h_3 = 297 \text{ kJ}$$

$$x_3 = 0.83$$



From steam tables, the specific volume of steam is

$$v_3 = 1.64 \text{ m}^3/\text{kg}$$

Velocity of steam at the nozzle mouth is

$$V_3 = 44.72\sqrt{297} = 771 \text{ m/s}$$

$$m = \frac{A_3 V_3}{x_3 v_3}$$

$$A_3 = \frac{m x_3 v_3}{V_3} = \frac{0.496 \times 0.83 \times 1.64}{771}$$

$$A_3 = 8.757 \text{ cm}^2.$$

Example 9.3

Dry saturated steam enters a convergent nozzle at a pressure of 10 bar with a velocity of 90 m/s. The exit pressure is 5 bar and steam leaves the nozzle at a velocity of 435 m/s. The friction is 8 kJ/kg of steam flow. Determine

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(i) Velocity coefficient

(ii) Final dryness fraction

(iii) Area at the exit if the inlet area is 1260 mm^2

Solution: Neglecting losses

$$h_2 + \frac{V_1^2}{2\times 100} = h_2 + \frac{V_2^2}{2\times 100} \qquad (1)$$

at 10 bar $h_1 = 2776.2$ kJ/kg $v_1 = 0.1943$ m^3/kg

at 5 bar $h_{w2} = 640.1$ kJ/kg $h_{fg2_{2'}} = 2107.4$ kJ/kg

From (1)

$$2776.2 + \frac{90^2}{200} = h_2 + \frac{455^2}{200}$$

$$h_2 = 2685.64 \text{ kJ/kg}$$

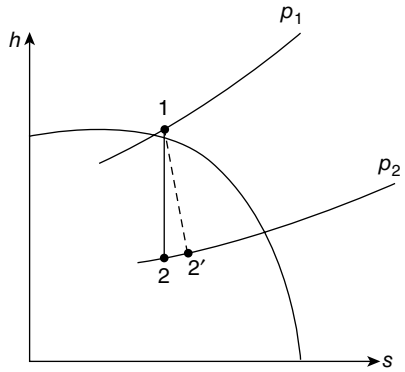
isentropic enthalpy drop $= h_1 - h_2$

$= 2776.2 - 2685.64$

$= 90.56$ kJ/kg

Actual enthalpy drop, $h_1 - h_2 = (90.56 - 8)$

$= 82.56$ kJ/kg



(i) Velocity coefficient

$$\text{Velocity coefficient} = \sqrt{\frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}}$$

$$= \sqrt{\text{nozzle efficiency}}$$

$$= \sqrt{\frac{82.56}{90.56}}$$

$$= 0.95.$$

(ii) Final dryness fraction

$$h_1 - h_{2'} = 82.56$$

$$h_{2'} = (2776.2 - 82.56)$$

$$= 2693.64$$

But, $h_{2'} = h_{2'} + x_{2'} h_{fg2'}$

$$= 640.1 + 2107.4$$

$$640.1 + x_2\, 2107.4 = 2693.64$$

$$x_{2'} = 0.97.$$

(iii) Area at exit A_2

$$\frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{x_2 v_{g2}}$$

$$\frac{0.00126 \times 90}{0.1943} = A_2 \times 435$$

$$A_2 = 487.6 \text{ mm}^2$$

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Example 9.4

Steam at 15 bar and 300°C expands in a nozzle to a pressure of 1 bar. If the efficiency of the nozzle is 80 per cent, calculate the mass of steam discharged when the exit area is 0.18×10^{-3} m^2.

Solution:

Initial pressure $p_1 = 15$ bar

Final pressure $p_2 = 1$ bar

Exit area $A_2 = 0.18 \times 10^{-3}$ m^2

Nozzle efficiency = 0.80%

Friction factor = 0.80.

From Mollier chart

$h_1 = 3037$ kJ/kg

$h_2 = 2515$ kJ/kg

$x_{2'} = 0.93$.

From steam tables $v_{s2} = 1.6937$ m^3/kg

Exit velocity $v_2 = 44.72\sqrt{\eta_n (h_1 - h_2)}$

$= 44.72\sqrt{0.8(3037 - 2515)}$

$= 914$ m/s.

Mass of steam discharged,

$$m = \frac{A_2 V_2}{x'_2 \, v_{g2}}$$

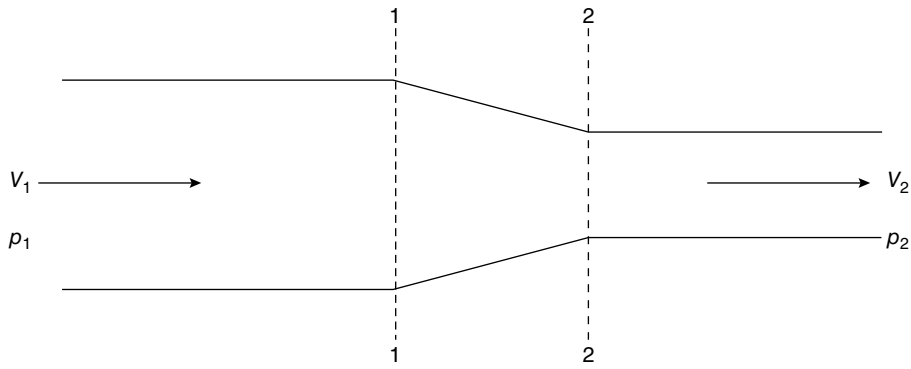
$$= \frac{0.18 \times 10^{-3} \times 914}{0.93 \times 1.6937}$$

$= 0.105$ kg/s.

9.10 Questions

9.10.1 Objective Questions

1. A fluid is flowing through a nozzle from 1 to 2 as shown in the figure.



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It can be concluded that

(a) $p_1 > p_2$ and $V_1 < V_2$ (b) $p_1 < p_2$ and $V_1 > V_2$

(c) Cannot be concluded (d) $p_1 > p_2$ and $V_1 > V_2$

2. In a steam nozzle if subscripts 1 and 2 represent inlet and outlet conditions and p_1, v_1, T and T_s are pressure, specific volume, temperature and saturation temperature of steam, which of the following ratios is less than 1?

(a) $\frac{p_1}{p_2}$ (b) $\frac{v_1}{v_2}$

(c) $\frac{T_1}{T_2}$ (d) $\frac{T_{s1}}{T_{s2}}$

3. Function of the steam nozzle is to convert

(a) enthalpy of steam into kinetic energy

(b) enthalpy of steam into pressure energy

(c) enthalpy of steam into pressure energy

(d) none of these

4. In a steam nozzle velocity at inlet V_1 and velocity at outlet V_2 are related as

(a) $V_1 = V_2$ (b) $V_1 > V_2$

(c) $V_2 > V_1$ (d) None of these

5. In a steam nozzle enthalpy at inlet h_1 and enthalpy at outlet are related as

(a) $h_1 = h_2$ (b) $h_1 > h_2$

(c) $h_2 > h_1$ (d) None of these

6. For a steam nozzle having varying area passage A, which of the following relations are true?

(a) $\frac{dA}{A} = \frac{dV}{V}\left(M^2 - 1\right)$ (b) $\frac{dA}{A} = \frac{dV}{V}\left(M - 1\right)$

(c) $\frac{dV}{V} = \frac{dA}{A}\left(M^2 - 1\right)$ (d) $\frac{dV}{V} = \frac{dA}{A}\left(M - 1\right)$

7. Expansion of steam through the nozzle is assumed as

(a) adiabatic (b) polytropic

(c) isothermal (d) isochoric

8. In the convergent section of a nozzle, Mach number M is

(a) $M > 1$ (b) $M = 1$

(c) $M < 1$ (d) $M = 0$

9. At the throat of the nozzle Mach number is

(a) $M > 1$ (b) $M = 1$

(c) $M < 1$ (d) $M = 0$

10. The critical pressure ratio for maximum discharge through the nozzles is

(a) $r_c = \left(\frac{2}{n+1}\right)^{\frac{n-1}{n}}$ (b) $r_c = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

(c) $r_c = \left(\frac{n-1}{n}\right)^{\frac{2}{n+1}}$ (d) $r_c = \left(\frac{2}{n+1}\right)^{\frac{1}{n}}$

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11. For superheated flow through nozzle
    (a) $pv^{1.3} = C$ (b) $pv^{1.4} = C$
    (c) $pv^{1.135} = C$ (d) $pv = C$
12. For saturated flow through nozzle
    (a) $pv^{1.135} = C$ (b) $pv^{1.4} = C$
    (c) $pv^{1.3} = C$ (d) $pv = C$
13. Critical pressure ratio for superheated steam is
    (a) 0.545 (b) 0.578
    (c) 0.530 (d) 0.587
14. Critical pressure ratio for dry saturated steam is
    (a) 0.545 (b) 0.578
    (c) 0.530 (d) 0.587
15. Friction in the nozzle
    (a) reduces exit velocity (b) increases specific volume
    (c) increases dryness fraction (d) all of the above
16. For a well-designed nozzle friction loss is limited to
    (a) 0–5% (b) 5–10%
    (c) 10–15% (d) 15–20%
17. Zeuner's relation, to determine index of expansion η for flow through a nozzle is
    (a) $\eta = 1.035 + 1x$ (b) $\eta = 1.135 + 0.1x$
    (c) $\eta = 1.035 + 0.1x$ (d) $\eta = 1.135 - 0.1x$
18. For supersaturated flow through a nozzle
    (a) $pv^{1.2} = C$ (b) $pv^{1.3} = C$
    (c) $pv^{1.4} = C$ (d) $pv^{1.135} = C$
19. The exit velocity (V_2) through a nozzle is given by
    (a) $V_2 = 91.5\sqrt{h_1 - h_2}$ (b) $V_2 = 44.7\sqrt{h_1 - h_2}$
    (c) $V_2 = 91.5\sqrt{h_2 - h_1}$ (d) $V_2 = 44.7\sqrt{h_2 - h_1}$
20. The cross section of the nozzle is
    (a) circular (b) square
    (c) elliptical (d) all of these types
21. If the exit pressure is equal to or more than critical pressure
    (a) a convergent nozzle is used
    (b) a divergent nozzle is used
    (c) a convergent–divergent nozzle is used
    (d) none of these
22. If the exit pressure is less than the critical pressure
    (a) a convergent nozzle is used
    (b) a divergent nozzle is u sed
    (c) a convergent–divergent nozzle is used
    (d) none of these

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23. At the throat of the nozzle
    (a) ratio V/v is maximum and A is minimum
    (b) ratio V/v is minimum and A is minimum
    (c) ratio V/v is maximum and A is maximum
    (d) ratio V/v is minimum and A is maximum
24. For supersonic and subsonic flow through the nozzle
    (a) pressure decreases and velocity decreases
    (b) pressure increases and velocity decreases
    (c) pressure decreases and velocity increases
    (d) pressure remains constant but velocity increases
25. Friction in nozzle
    (a) increases mass flow rate
    (b) does not affect mass flow rate
    (c) decreases mass flow rate
    (d) none of these
26. When a nozzle is operated below its designed value, it is known as
    (a) under expanding
    (b) over expanding
    (c) choked
    (d) stalled
27. A steam nozzle is said to be choked when
    (a) flow rate through it is completely stopped
    (b) flow rate through it is maximum
    (c) flow rate through it is minimum
    (d) none of these
28. Under maximum discharge conditions, the flow velocity in the convergent section of the nozzle is given by Mach number
    (a) $M = 1$
    (b) $M < 1$
    (c) $M > 1$
    (d) None of these
29. Wilson's line is a characteristic of
    (a) saturated flow through nozzle
    (b) supersaturated flow through nozzle
    (c) friction horse power
    (d) none of these
30. Supersaturated flow through the nozzle means
    (a) expansion of steam beyond the saturation line
    (b) expansion of steam up to the saturation line
    (c) expansion of steam above the saturation line
    (d) none of these
31. Supersaturated flow through the nozzle occurs due to
    (a) short time interval available for expansion
    (b) long time interval available for expansion
    (c) insufficient time available for expansion
    (d) none of these

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32. Shock waves in the nozzle are generally observed
    (a) in the convergent section of the nozzle
    (b) in the divergent section of the nozzle
    (c) at the entry of the nozzle
    (d) none of these
33. The effect of supersaturated flow through the nozzle is
    (a) reduction in heat drop
    (b) increase in entropy
    (c) decrease in exit velocity
    (d) all of the above

Answers

1. a	2. b	3. a	4. c	5. b	6. a	7. a	8. c	9. b	10. b
11. a	12. a	13. a	14. b	15. d	16. b	17. c	18. b	19. b	20. d
21. b	22. a	23. b	24. c	25. b	26. d	27. b	28. b	29. b	30. a
31. a	32. b	33. d							

9.10.2 Review Questions

1. What do you understand by the term critical pressure as applied to steam nozzles?
2. Why are turbine nozzles made divergent after the throat?
3. Explain the term nozzle efficiency.
4. A convergent–divergent nozzle is required to discharge 360 kg of steam per hour. The nozzle is supplied with steam at 10 bar and 0.97 per cent dry and discharges against a back pressure of 0.5 bar. Neglecting the effect of friction and using H–s chart, the throat and exit diameters.
5. Steam at a pressure of 10 bar and 0.98 per cent dry is passed through a convergent–divergent nozzle to a back pressure of 0.1 bar. The mass flow rate is 0.55 kg/s. Find
   (i) pressure at the throat and
   (ii) number of nozzles used if each nozzle has a throat area of 0.5 cm^2.

   The enthalpy drop used for reheating the steam by friction in the divergent part is 10 per cent of the overall isentropic drop.

   Take index of expansion = 1.13.
6. A convergent–divergent nozzle is required to discharge 350 kg of steam/hour. The nozzle is supplied with steam at 8.5 bar and 90 per cent dry and discharge against a back pressure of 0.4 bar. Neglecting the effect of friction, find the throat and exit areas. Show the process on H–s diagram.

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Steam Turbines 10

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10.1 Introduction to steam turbines
10.2 Turbine types
10.3 Simple impulse turbines
10.4 Reaction turbines
10.5 Losses in steam turbines
10.6 Reheat factor and condition line
10.7 Governing of turbines
10.8 Questions

10.1 INTRODUCTION TO STEAM TURBINES

Steam turbine is one of the principal prime movers used for electric power generation. The steam turbine obtains its motive power from change of momentum of a jet of steam flowing over a curved vane. The steam jet exerts a pressure on the blade when it is moved over the curved surface of the blade, owing to its centrifugal force. The centrifugal pressure is exerted normal to the blade surface and acts along the whole length of the blade. The resultant of centrifugal pressure, plus the effect of change of velocity, is the motive force on the blade.

In a steam turbine, a number of small blades are fixed to the rim of a revolving wheel or rotor. High-velocity jets of steam are obtained by expansion through nozzles and are directed on to the blades. The effective force of these jets, acting on the blades, rotates the wheel. This chapter deals with steam turbines briefly.

10.2 TURBINE TYPES

There are two types of steam turbines:*impulse* and *reaction*. These two turbines work in a distinct manner. The main distinction is the manner in which the steam is expanded as it passes through the turbine. In the *impulse turbine*, the steam expands in the nozzles and remains at constant pressure while passing over the blades. In the *reaction turbine*, the steam continuously expands as it flows over the blades.

10.2.1 Impulse Turbine Working Principle

In impulse turbine, high-velocity steam coming out from the nozzle strikes the blades fixed on the periphery of the rotor. The passage of steam through the moving part of the turbine

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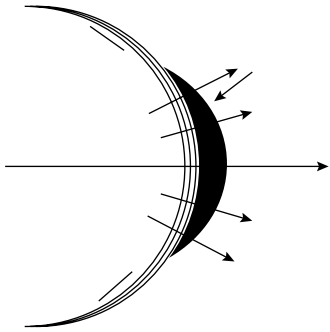


Fig. 10.1 *Impulse Turbine Working Principle*

(blades) takes place in such a manner that the pressure at the outlet end of the blade equals to that at the inlet end. The net force acting on the blades results in change in momentum due to which the turbine shaft rotates.

The steam turbine obtains its motive power due to *change in momentum* of a jet of steam flowing over a curved vane or blade as shown in Figure 10.1. When the jet containing steam is moved over the curved surface of the blades, pressure is exerted on the blades due to *centrifugal action*. This centrifugal pressure is exerted normal to the blade surface and acts along the whole blade length. The blade is designed in such a way that the steam jet glides on and off the blade without striking it.

Thus, the *resultant of all these centrifugal pressures and the effect of change of velocity is the motive force that causes the turbine to move*.

In steam turbines, a number of such small blades are fixed to the rim of the revolving shaft or rotor. As the jet of steam strikes these blades, the rotor gains the momentum and rotates. Examples of an impulse turbine include Curtis turbine, De Laval turbine, Rateau turbine, etc.

10.2.2 Reaction Turbine

In this type of turbine, steam does not get expanded in the nozzles, but expanded as it flows over the blades. Unlike impulse turbines, the blades are not symmetrical. The blades also act as nozzles. The drop in pressure of the steam during its flow through nozzles and moving blades results in an increase in the kinetic energy. The resulting propelling force gives a rotary motion to the turbine rotor. Due to change of velocity, the rotating element exerts a thrust or reaction on the blades tangentially as shown in Figure 10.2(a). The reaction constitutes the driving force, and hence is called reaction turbine.

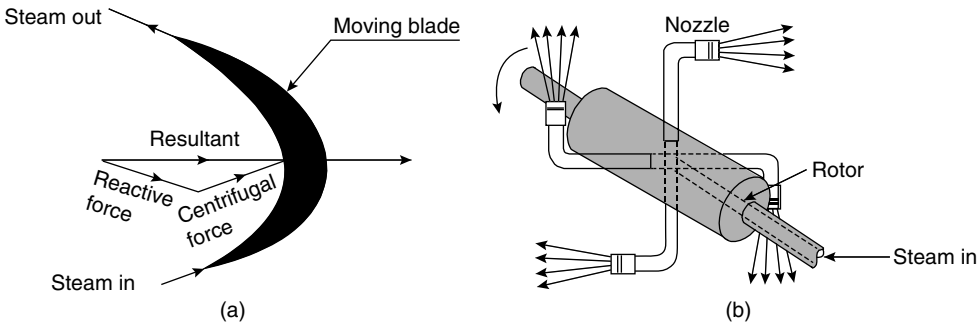


Fig. 10.2 *Reaction Turbine Working Principle*

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10.3 SIMPLE IMPULSE TURBINES

10.3.1 Flow Through Turbine Blades and Velocity Diagram

Figure 10.3 shows a single-stage impulse turbine. It consists of a set of nozzles and moving blades. High-pressure steam at boiler pressure p_b enters the nozzles and expands in it. Due to the expansion, steam pressure drops to condenser pressure p_c entirely in the nozzle itself as represented by the curve PQ. This results in an increase in the kinetic energy of the steam, and the steam leaves the nozzle at a very high velocity V_e. It can be observed that the final velocity of the steam increases at the nozzle exit and drops again in the moving blades. This is because the moving blades absorb the kinetic energy of the blades. However, the final velocity V_e is much higher than the initial velocity V_i. The steam pressure remains constant throughout after expansion in the moving blades as indicated by the straight line QR.

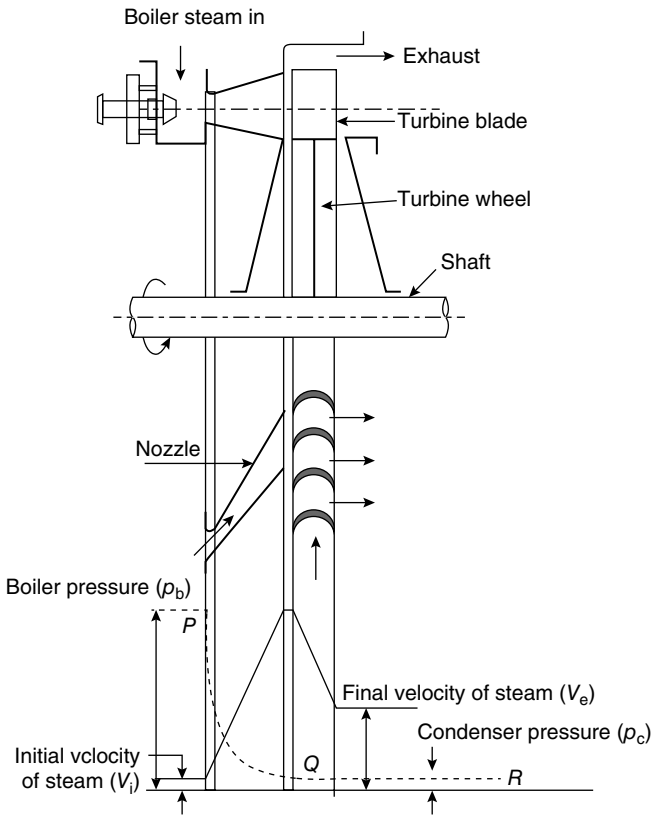


Fig. 10.3 *A Single-Stage Impulse Turbine*

10.3.2 Compounding of Steam Turbines

If the steam pressure drops from boiler pressure to condenser pressure in a single stage, exit velocity of steam from the nozzle will become very high, and the turbine speed will be of the

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order of 30,000 rpm or more. As turbine speed is proportional to steam velocity, the *carryover loss* or *leaving loss* will be more (10–12 per cent). Due to this very high speed, centrifugal stresses are developed on the turbine blades resulting in *blade failure*. To overcome all these difficulties, it is necessary to reduce the turbine speed by the method of compounding.

Compounding is the method of reducing blade speed for a given overall pressure drop. Compounding of steam turbines is done by the following methods.

10.3.2.1 Pressure Compounded Impulse Turbines

If a number of simple impulse stages are clubbed together, then the arrangement is known as pressure compounding. The arrangement contains one row of fixed blades or nozzles at the entry of each row of moving blades. The total pressure drop does not take place in the first row of nozzles, but is divided equally between all the nozzles as shown in Figure 10.4.

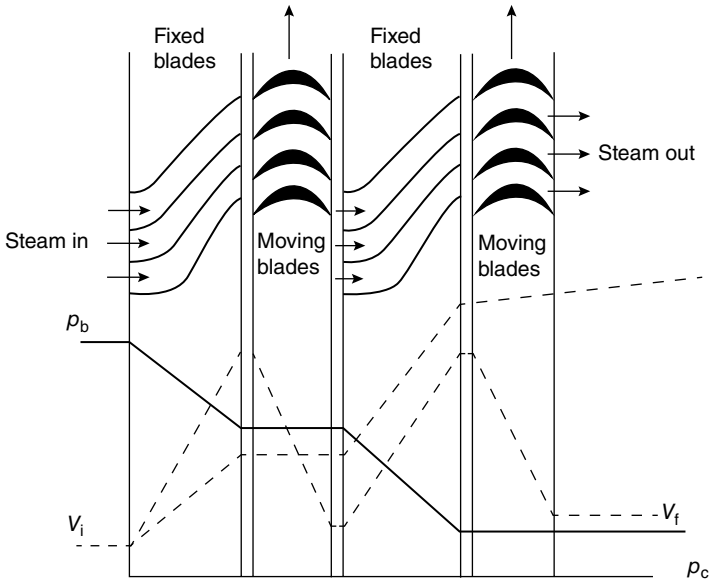


Fig. 10.4 *A Simple Pressure Compounded Impulse Turbine*

The steam from the boiler is passed through the first set of fixed nozzles (blades) in which it is partially expanded. The steam then passes over the first row of moving blades where almost all its velocity is absorbed. This complete expansion of steam is *one stage*. In the *next stage*, the steam again enters the second set of fixed nozzles and partially expands and enters the moving blades. Again the steam velocity is absorbed. This process continues till the steam reaches the condenser pressure, p_c.

Due to pressure compounding, a smaller transformation of heat energy into kinetic energy takes place. Hence, steam velocities become much lower and reduce the blade velocities and rotational speed.

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10.3.2.2 Velocity Compounded Impulse Turbines

A simple velocity compounded impulse turbine is shown in Figure 10.5. It consists of a set of nozzles and a rotating wheel fitted with two or more rows of moving blades. One row of fixed blades between the moving blades directs steam from the nozzle over the moving blades.

The steam from the boiler expands in the nozzle and enters the first row of the moving blades. A portion of kinetic energy is absorbed by the blades before passing over the fixed blades or guide blades. Velocity slightly drops in guide blades before steam is sent to the second row of the moving blades. Steam does work on the second row of the moving blades losing its kinetic energy and leaves axially with residual velocity V_f. Due to this, the rotor speed decreases considerably.

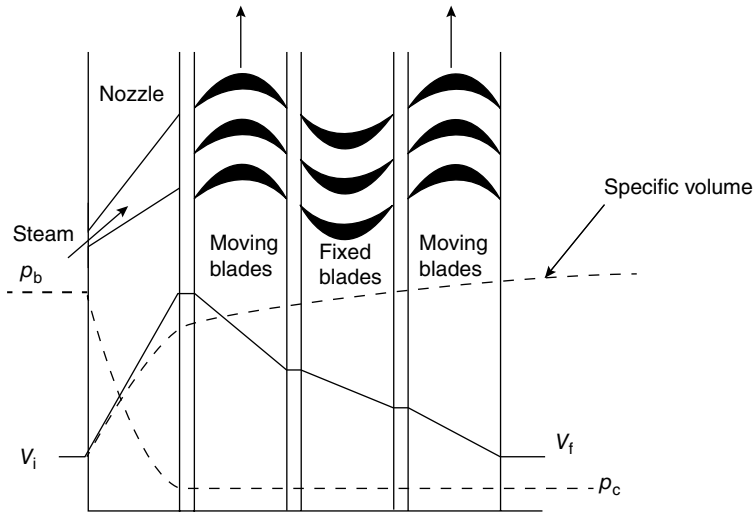


Fig. 10.5 *A Simple Velocity Compounded Impulse Turbine*

Advantages and disadvantages of velocity compounding
Advantages

(i) Due to large enthalpy drop, the pressure drop in the nozzle's first stage is also large. This reduces stress in the turbine housing.

(ii) Velocity compounding principle leads to a relatively large number of enthalpy drop per stage. Hence, fewer number of stages are required, which leads to low initial cost.

(iii) With a single row, steam temperature used is about 300°C, which is high enough for ordinary cast iron to withstand. By using two rows, the steam temperature is also lowered and ordinary CI can be used, provided steam chest is constructed using stainless steel.

(iv) Due to fewer number of stages, space required is less.

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Disadvantages

(i) Specific volume of steam continuously increases with more number of stages, imposing limitations to blade design.

(ii) Friction losses become higher due to high-velocity steam at the blade outlet.

(iii) As enthalpy drop per stage is large, efficiency ratio decreases with increased number of rows as shown in Figure 10.6.

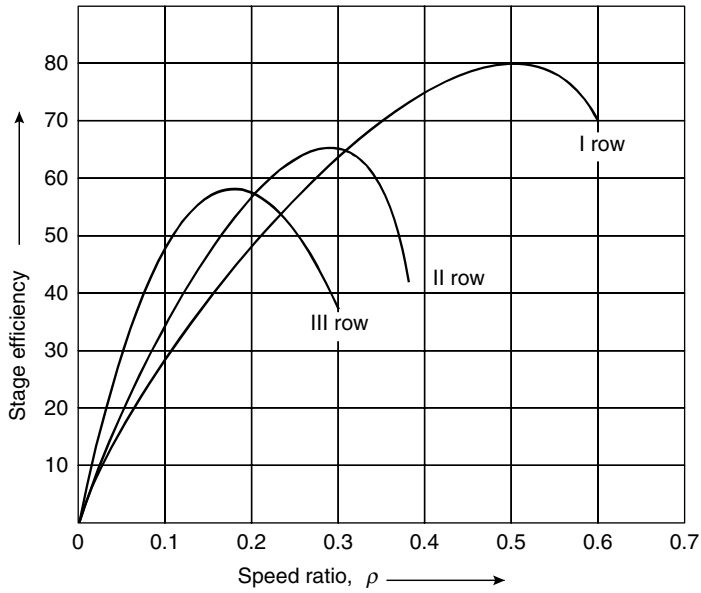


Fig. 10.6 *Variation of Stage Efficiency with Speed Ratio*

(iv) As the work done per kilogram of steam decreases progressively with the number of stages, space, material cost and initial cost add up.

10.3.2.3 Pressure–Velocity Compounding

If pressure and velocity are both compounded using two or more number of stages by having a series arrangement of simple *velocity compounded* turbines on the same shaft, it is known as pressure–velocity compounding. In this type of turbine, both pressure compounding and velocity compounding methods are used. The total pressure drop of the steam is divided into two stages, and the velocity obtained in each stage is also compounded. Pressure drop occurs only in nozzles and remains constant throughout. As pressure drop is large in each stage, only a few stages are necessary. This makes the turbine more compact than the other two types. Pressure–velocity compounding (Figure 10.7) is used in *Curtis turbine*.

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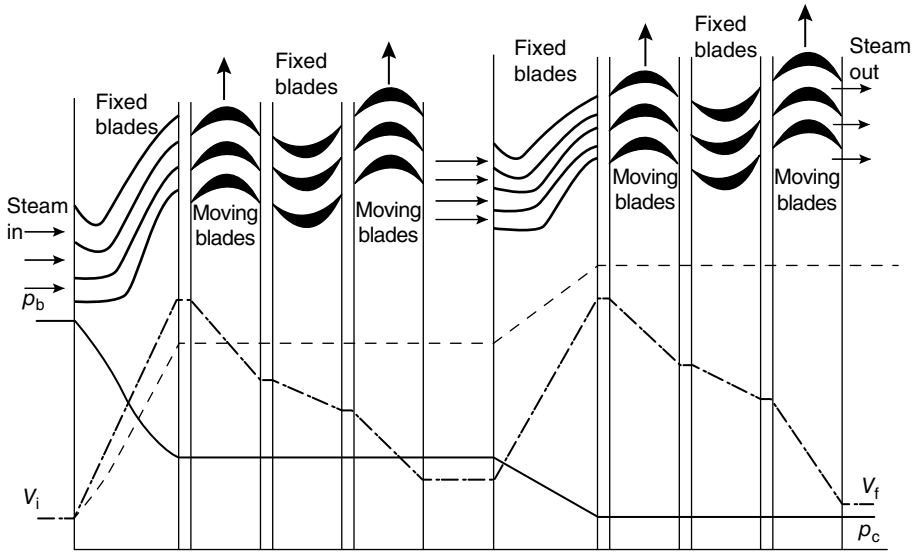


Fig. 10.7 *Pressure–Velocity Compounding*

10.3.3 Impulse Turbine Power and Related Calculations

10.3.3.1 Single-Stage Impulse Turbine

The essential parts of an impulse steam turbine are nozzles and blades. Steam expands in the nozzles producing high-velocity jet at the nozzle exit. Blades change the direction and hence momentum of the jet of steam. The force produced due to change in momentum propels the blades. Hence, it is of primary interest to estimate the propelling force that is applied at the turbine rotor. As force is due to *change in momentum* caused by change in direction of steam, the variation of steam velocity can be studied by drawing the velocity triangles. Impulse force acting on the blades can be calculated in the plane of rotation of blades tangential to the turbine runner.

Figure 10.8 shows the velocity diagram for an *impulse turbine*. Due to the expansion of steam in nozzles, steam comes out with an absolute velocity V_1 at an angle α_1 with the plane of moving blades. If the blade velocity is represented by u, then the relative velocity of steam with respect to the casing is V_{r1}. *As steam flows over the blades, the relative velocity remains constant for an impulse turbine.*

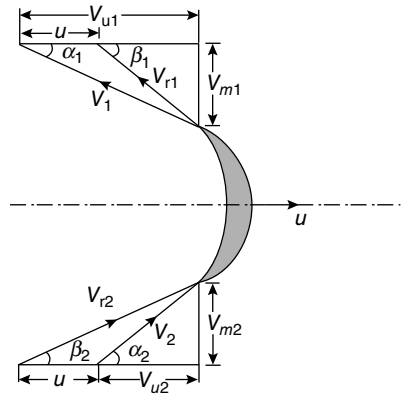


Fig. 10.8 *Velocity Diagram for an Impulse Turbine*

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The combined velocity diagrams (Figure 10.9) at the inlet and outlet for an impulse turbine is drawn using the following notations:

V_1 = steam velocity at the outlet from nozzles
= initial absolute velocity at the inlet of moving blades
V_2 = final absolute velocity at the outlet of moving blades
V_{r1} = steam velocity relative to moving blades at the inlet
V_{r2} = steam velocity relative to moving blades at the outlet
V_{u1}, V_{u2} = tangential component of absolute velocity at inlet and outlet respectively
V_{m1}, V_{m2} = radial component of absolute velocity at inlet and outlet respectively
α_1 = angle made by absolute velocity V_1 with the plane of moving blades
= nozzle or jet angle or fixed blade angle at the inlet
α_2 = angle made by absolute velocity V_2 with the plane of moving blades
= fixed blade angle at the outlet
β_1 = inlet blade angle
β_2 = outlet blade angle
m = steam flow through blades
$\Delta V_u = V_{u1} + V_{u2}$

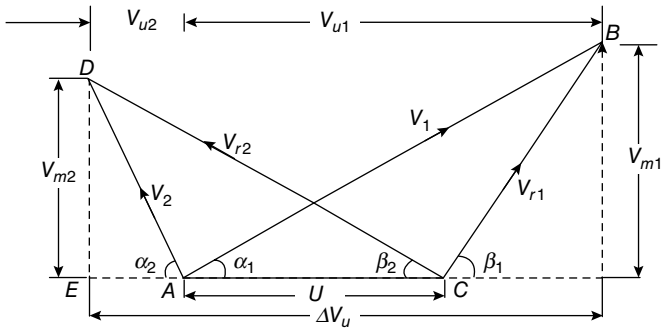


Fig. 10.9 *Velocity Triangle on Common Base*

1. *Power developed*

Resultant force necessary to produce change of velocity is equal to the rate of change of momentum.

$\therefore$ EF = initial velocity – final velocity

$= V_{u1} - (-V_{u2})$

$F = V_{u1} + V_{u2}$ (1)

$F = V_u$

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Work done per kilogram of steam,

$$W_d = F \times u$$

$$W_d = V_u \times u \tag{2}$$

Power developed by the runner

$$P = mW$$

$$P = \frac{m \times V_u \times U}{1000} \tag{3}$$

2. *Blade efficiency or diagram efficiency*

For a single-stage impulse turbine blade efficiency is given by the following equation:

$$\eta_b = \frac{\text{work done on blades per unit mass of steam}}{\text{Energy supplied per unit mass of steam}}$$

$$\eta_b = \frac{2uV_u}{V_1^2} \tag{4}$$

3. *Effect of friction on blade efficiency*

Due to the presence of friction in blade channels, the relative velocity of steam at the outlet is lesser than the relative velocity of steam at the inlet.

$$\therefore \qquad \frac{V_{r2}}{V_{r1}} = k$$

where $k = A$ constant lesser than unity

$$\eta_b = 2\,(1 + kC)\,(\rho \cos\alpha_1 - \rho^2)$$

The above equation refers to the equation of a parabola as shown in Figure 10.10. The curves show that blade efficiency increases as ρ is increased and reaches a maximum value approximately at 0.5 and then decreases.

As α_1, k and C are all constant, η_0 depends only on the value ρ. Blade efficiency will be maximum when $\frac{d\eta_b}{d\rho} = 0$

Or $$\rho_{opt} = \frac{\cos\alpha_1}{2} = \left(\frac{U}{V_1}\right)_{opt}$$

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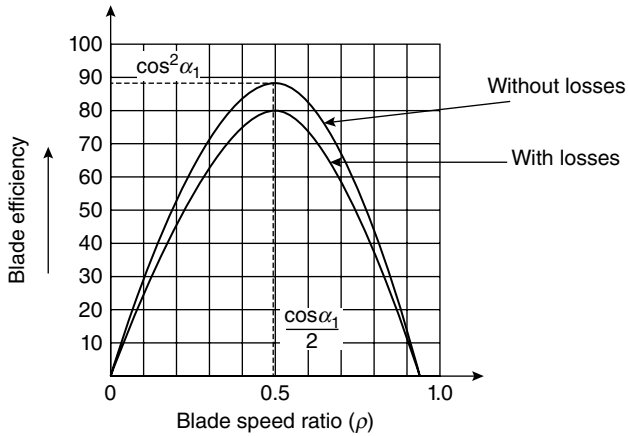


Fig. 10.10 *Blade Efficiency versus Blade Speed Ratio*

Maximum blade efficiency

$$\eta_{b\,max} = 1 + kC\frac{\cos^2\alpha_1}{2}$$

If blades are symmetrical and friction is absent, then $k = 1$ and $C = 1$

$$\eta_{b\,max} = \cos^2\alpha_1$$

Maximum work done per kilogram of steam is given by the following equation:

$$W_{max} = \left(\frac{U(1+kC)}{1000}\right)$$

If blades are symmetrical and friction is absent, $k = 1$ and $C = 1$

$$W_{max} = \frac{2U^2}{1000}$$

At the optimum value of ρ, the absolute velocity at the outlet is axial. In the absence of friction, for equiangular blades and axial discharge,

$$\alpha_2 = 90°;\ V_2 = V_{m2};\ V_{u2} = 0$$

$$\therefore\ V_{u1} = V_1\cos\alpha_1$$

$$= 2U$$

4. *Gross stage efficiency*

It is the ratio of work done on the blades per kilogram of steam to the adiabatic enthalpy drop per stage.

Therefore,
$$\eta_{gs} = \left(\frac{V_u U}{\Delta h_a}\right)$$

$$\eta_{gs} = \eta_b \times \eta_n$$

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where η_n = nozzle efficiency

$(\Delta h)_a$ = adiabatic enthalpy drop

5. *Axial thrust*

The axial thrust on the rotor is due to change in axial component of absolute velocity.

Therefore, $$F_a = \left(\frac{m(V_{m1} - V_{m2})}{1000} \right) \text{kN}$$

10.3.4 Multistage Impulse Turbine

As the absolute velocity becomes very high in a single turbine, blade velocity also becomes very high. As high velocity causes *centrifugal stresses* on the blades resulting in structural failure, multistage is preferred to overcome these problems.

The velocity diagrams for a two-row impulse turbine are shown in Figure 10.11. For maximum efficiency and minimum *carry-over loss* (due to kinetic energy), absolute velocity of steam leaving the second row must be axial.

If *suffix* 1 and 2 are used to indicate the *first* and *second rows* or stages, respectively, then total work done per kilogram of steam passing through both the stages is given by

$$W = W_1 + W_2$$
$$= {}_1V_u U + {}_2V_u U$$

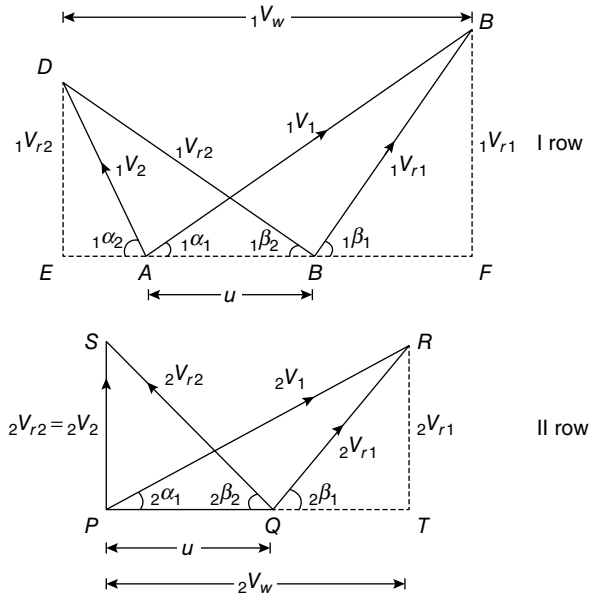


Fig. 10.11 *Velocity Triangles for a Multistage Impulse Turbine*

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From the velocity triangles,

$$W_d = \{(AF + AE) + (QT + QP)\}U$$

$$W_d = \{({}_1V_{r1}\cos_1\beta_1 + {}_1V_{r2}\cos_1\beta_2) + ({}_2V_{r1}\cos_2\beta_1 + {}_2V_{r2}\cos_2\beta_2)\}U$$

If there is no friction and symmetrical blade profile is used, then $\beta_1 = \beta_2$ and $V_{r1} = V_{r2}$ for both stages:

$$W_d = 2({}_1V_{r1}\cos_1\beta_1 + {}_2V_{r1}\cos_2\beta_1)U$$

$$= 2U\{{}_1V_1\cos_1\alpha_1 - U + {}_2V_1\cos_2\alpha_1 - U\}$$

As the magnitude of absolute velocity of steam leaving the first row and entering the second row remains the same,

$${}_2V_1 = {}_1V_2$$

$${}_2\alpha_1 = {}_1\alpha_2$$

$$\therefore \quad {}_2V_1\cos_2\alpha_1 = {}_1V_2\cos_1\alpha_2$$

$$= {}_1V_{r2}\cos_1\beta_2 - U$$

$$= {}_1V_{r1}\cos_1\beta_1 - U$$

$$= {}_1V_1\cos_1\alpha_1 - U - U$$

$$= {}_1V_1\cos_1\alpha_1 - 2U$$

$$W_d = 2U\{{}_1V_1\cos_1\alpha_1 + {}_1V_1\cos_1\alpha_1 - 4U\}$$

$$W_d = 4U\{{}_1V_1\cos_1\alpha_1 - 2U\}$$

Blade efficiency

$$\eta_b = 8\rho(\cos_1\alpha_1 - 2\rho)$$

Maximum blade efficiency

$$\eta_{max} = \cos^2{}_1\alpha_1$$

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Maximum work done per kilogram of steam

$$W_{d\,max} = \frac{8U^2}{1000} \text{ kNM}$$

In general, if there are n number of stages,

$$\rho_{opt} = \frac{\cos_1 \alpha_1}{2n}$$

As the utility of the last row decreases with increased number of stages, in practice not more than two rows are preferred.

10.4 REACTION TURBINES

10.4.1 Flow Through Blades and Velocity Diagram

Figure 10.12 shows a simple axial flow reaction turbine consisting of many rows of moving blades and fixed blades. The moving blades are attached to the turbine rotor, whereas the fixed blades are attached to the turbine casing. The fixed blades serve the purpose of nozzles in the impulse turbine. Steam after passing through a number of fixed blades experiences small drop in pressure gaining velocity. It then enters the first row of moving blades suffering a *change in direction and momentum*. This results in an impulse acting on the blades. The pressure of steam again drops with further gain in velocity giving rise to a *reaction* opposite to that of the gained velocity. The moving blades are also known as *rotor blades* (*R*) and the fixed blades are also known as *stator blades* (*S*).

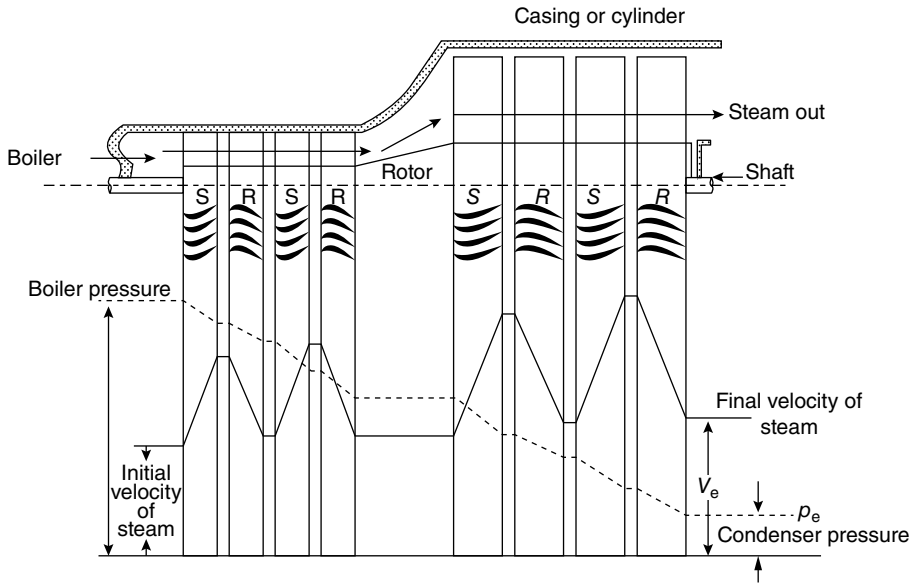


Fig. 10.12 *A Simple Axial Flow Reaction Turbine*

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10.4.2 Degree of Reaction

Degree of reaction is defined as the ratio of enthalpy drop in moving blades to the sum of enthalpy drop in moving and fixed blades.

$$\therefore R = \frac{\text{Enthalpy drop in moving blades}}{\text{Enthalpy drop in moving blades} + \text{Enthalpy drop in fixed blades}}$$

$$R = \frac{h_b}{h_n + h_b} \tag{1}$$

Enthalpy drop in moving blades

$$h_b = \frac{V_{r2}^2 - \phi V_{r2}^2}{2\eta_n} \tag{2}$$

But $$V_{r1}^2 = V_1^2 + U^2 - 2UV_1 \cos\alpha_1$$

$$= V_1^2 \ (1 + \rho_1^2 - 2\rho_1 \cos\alpha_1)$$

Substituting in Equation (2),

$$h_b = \left(\frac{V_{r2}^2 - \varphi V_1^2 \left[1 + \rho_1^2 - 2\rho_1 \cos\alpha_1 \right]}{2\eta_n} \right) \tag{3}$$

where $\phi = k^2$

= carryover coefficient, which takes into account the inlet losses

η_n = nozzle efficiency

$$\rho_1 = \frac{u}{V_1}$$

Enthalpy drop in fixed blades

$$h_n = \frac{V_{12}^2 - \phi V_{12}^2}{2\eta_n} \tag{4}$$

But $V_1^2 = V_{r2}^2 + u^2 - 2u\, V_{r2} \cos\beta_2$ (from velocity diagram).

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Substituting in Equation (4)

$$h_n = \frac{V_1^2 - \varphi V_{r2}^2 \left[1 + \rho_2^2 - 2\rho_2 \cos\beta_2\right]}{2\eta_n} \tag{5}$$

where $\rho_2 = \frac{u}{v_{r_2}}$; substituting Equations (3) and (5) in Equation (1), value of R is determined.

Note: As carryover to the stage is equal to carryover from the stage, total enthalpy drop in moving and fixed blades can also be given by

$$(h_b + h_n) = u\, V_w$$

Assuming $\phi = 1$,

$$R = \frac{V_{r2}^2 - V_{r1}^2}{2uV_w} \tag{6}$$

From velocity triangles,

$$V_{r2} = V_f \operatorname{cosec}\beta_2$$

$$V_{r1} = V_f \operatorname{cosec}\beta_1 \; (V_{f1} = V_{f2} = V_f)$$

$$V_w = V_f \cot\beta_1 + V_f \cot\beta_2$$

Substituting in Equation (6)

$$R = \frac{V_f^2 \left(\operatorname{cosec}^2\beta_2 - \operatorname{cosec}^2\beta_1\right)}{2U\left(V_f \cot\beta_1 + V_f \cot\beta_2\right)}$$

$$= \frac{V_f \left(\cot^2\beta_2 - \cot^2\beta_1\right)}{2U\left(\cot\beta_1 + \cot\beta_2\right)}$$

$$R = \frac{V_f}{2U}\left(\cot\beta_2 - \cot\beta_1\right) \tag{7}$$

For Parson's turbine, degree of reaction is 0.5 such that $\beta_1 = \alpha_2$ and $\beta_2 = \alpha_1$.

$$\therefore \qquad 1 = \frac{V_f}{2U}\left(\cot\beta_2 - \cot\beta_1\right)$$

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or $$U = V_f \left(\cot \beta_2 - \cot \beta_1\right)$$

$$U = V_f \left(\cot \alpha_2 - \cot \alpha_1\right) \tag{8}$$

10.4.3 Parson's Reaction Turbine

In Parson's turbine, section of blades is the same for moving as well as fixed rows of blades. It is assumed that the mean diameter of the blades is kept constant. The height of the blades is very much increased that the steam velocity at the exit from each row of blades is uniform and velocity triangles are congruent. The combined velocity diagram for the Parson's Reaction Turbine is shown in Figure 10.13.

As the area at the outlet from the blade channel is restricted by the section of moving blades, relative velocity increases such that $V_{r2} = V_1$.

∴ Enthalpy drop in moving blades,

$$h_b = \frac{V_1^2 - \varphi V_2^2}{2\eta_n}$$

$$= \frac{V_1^2 - \varphi Vr^2_1}{2\eta_n} \tag{1}$$

But $$V_{r1}^2 = V_1^2 + u^2 - 2u\, V_1 \cos\alpha_1$$

$$V_{r1}^2 = V_1^2 + (1 + \rho^2 - 2\rho \cos\alpha_1)$$

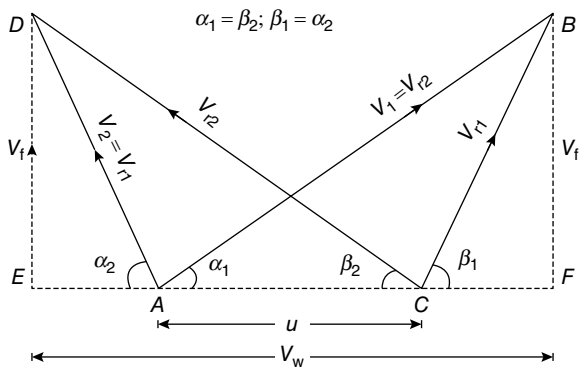


Fig. 10.13 *Velocity Triangles for a Parson's Reaction Turbine*

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∴ Enthalpy drop in moving rows

$$h_b = \frac{V_1^2}{2\eta_n}\left[1-\alpha\left(1+\rho_2^2-2\rho_2\cos\alpha_1\right)\right] \tag{2}$$

where $\rho_1 = \frac{u}{V_1}$ and $\rho_2 = \frac{u}{V_2}$

In each row of fixed blades, steam enters with a velocity V_2 (= V_{r1}) and leaves with a velocity V_1 except in the first row. Hence, enthalpy drop in fixed blades,

$$h_n = \frac{V_1^2 - \varphi V r_1^2}{2\eta_n} \tag{3}$$

∴ $$h_b = h_n \text{ and } R = \frac{2\cos^2\alpha_1}{1+\cos^2\alpha_1}$$

∴ Total adiabatic enthalpy drop,

$$h_n + h_b = \frac{V_1^2}{2\eta_n}\left[1-\varphi\left(1+\rho^2-2\rho\cos\alpha_1\right)\right] \tag{4}$$

10.4.4 Power and Related Calculations

When the pressure drop occurs in both moving and fixed blades, the turbine is known as an *impulse–reaction turbine*. Figure 10.13 shows a single-stage impulse reaction turbine. Steam comes out from the fixed blades or nozzles with an absolute velocity V_1 at an angle α_1 with the plane of moving blades. The relative velocity V_{r2} at the outlet is k times V_{r1}. Due to change in absolute velocity, an impulse force will act on the blades. Further expansion of steam in the turbine results in increased relative velocity at the outlet giving rise to a reactive force.

Work done per kilogram of steam

$$W_d = uV_w$$

$$= u\,(V_1\cos\alpha_1 + V_1\cos\alpha_1 - u)$$

$$W_d = V_1^2\left(2\rho\cos\alpha_1 - \rho^2\right) \tag{1}$$

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Gross stage efficiency

$$\eta_{gs} = \frac{V_1^2\left(2\rho\cos\alpha_1 - \rho^2\right)}{h_n + h_b}$$

$$\eta_{gs} = \frac{\eta_n\left(2\rho\cos\alpha_1 - \rho^2\right)}{1-\varphi\left(1+\rho^2 - 2\rho\cos\alpha_1\right)}$$

Blade efficiency

As $\eta_{gs} = \eta_b \times \eta_n$

$$\eta_b = \frac{\left(2\rho\cos\alpha_1 - \rho^2\right)}{1-\phi\left(1+\rho^2 - 2\rho\cos\alpha_1\right)} \tag{2}$$

Blade efficiency is maximum when $2\rho\cos\alpha_1 - \rho^2$ is maximum

i.e. $\frac{d}{d\rho}\left(2\rho\cos\alpha_1 - \rho^2\right) = 0$

or $2\cos\alpha_1 - 2\rho = 0$

$$\rho = \cos\alpha_1$$

Substituting in Equation (2)

$$\eta_{b\,max} = \frac{\left(2\cos^2\alpha_1 - \cos^2\alpha_1\right)}{1-\phi\left(1+\cos^2\alpha_1 - 2\cos^2\alpha_1\right)} \tag{3}$$

$$= \frac{\cos^2\alpha_1}{1-\phi\left(1-\cos^2\alpha_1\right)}$$

Example 10.1

The isentropic enthalpy drop from the inlet to exit of a 12-stage Parson's turbine is 450 kJ/kg. The gross stage efficiency and reheat factor are 0.76 and 1.02, respectively. If turbine develops 10 MW power, determine the steam flow rate through the turbine if all stages develop equal power. If blade outlet angle is 20° and blade speed ratio is 0.61, find the mean diameter of the

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rotor and rotor speed, assuming that steam is dry saturated at 2 bar for this stage. Assume further that the blade height is 1/10th of mean diameter, blade thickness is 1/10th of blade height and there are 20 blades.

Solution:

Given data:

$\Delta h = 450$ kJ/kg; $n = 12$; $\eta_s = 0.76$; $R_f = 1.02$

$P = 10 \times 10^6$ W; $\alpha_1 = 20° = \beta_2$; $B = \dfrac{D}{10}$; $t = B/10$

Blade efficiency is given by

$$\eta_b = \eta_s \times R_f = 0.76 \times 1.02 = 0.775$$

But blade efficiency is given by

$$\eta_b = \frac{\text{Workdone}}{\text{Isentropic enthalpy drop}}$$

i.e. $$\eta_b = \frac{W_d}{\Delta h}$$

$$W_d = \eta_b \times \Delta h$$

$$= 0.775 \times 450 = 348.75 \text{ kN/kg}$$

Since there are 15 stages, workdone/stage/kg is given by

$$= \frac{348.75}{12} = 29.06 \text{ kN}$$

Power developed per stage is given by

$$\frac{P}{n} = m \times 29.06$$

$\therefore$ m = mass flow rate

$$= \frac{P}{n \times 29.06}$$

$$= \frac{10 \times 1000}{12 \times 29.06} = 28.68 \text{ kg/s.}$$

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Workdone/stage/kg is given by

$$29.06 = \frac{uV_w}{1000} = \frac{V_1^2\left(2\rho\cos\alpha_1 - \rho^2\right)}{1000} = \frac{2uV_1\cos\alpha_1 - u^2}{1000}$$

$$= \frac{2u \times \frac{u}{0.6} \times \cos 20° - u^2}{1000}$$

$$= \frac{3.13\,u^2 - u^2}{1000}$$

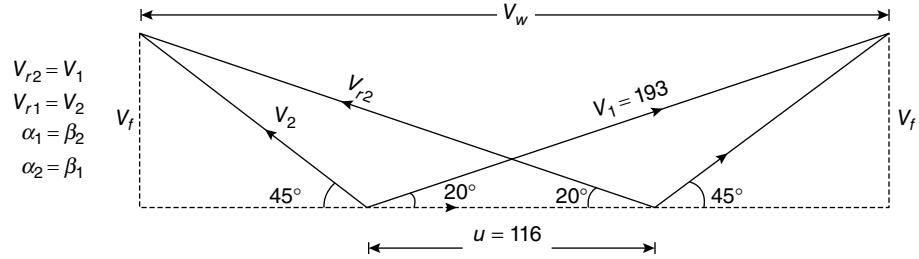
$\therefore$ Mean blade speed $u = 116.80$ m/s

Mathematically, $V_f = V_1 \sin\alpha_1$

$$= \frac{116.80}{0.61} \times \sin 20° = 66.6 \text{ m/s}$$

From velocity triangles by measurement,

$$V_f = 67 \text{ m/s}$$



Mean diameter or rotor:

From steam tables at 2 bar and dry saturated condition specific volume, $v = 0.8857$ m^3/kg

Mass flow rate:

$$m = (\pi D - nt)\, BV_f$$

$$28.68 = \left(\pi D - 20 \times \frac{B}{10}\right) \times B \times 67 = \left(\pi D - 20 \times \frac{D}{120}\right) \times \frac{D}{10} \times 67 = 16.61\, D^2$$

$$D = 1.314 \text{ m}$$

Rotor speed:

$$\text{Rotor speed, } N = \frac{60u}{\pi D} = \frac{60 \times 116.80}{\pi \times 1.314}$$

$$= 1697.65 \text{ rpm}$$

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10.4.4.1 Comparison of Impulse and Impulse Reaction Turbines

S. no.	Impulse turbine	Impulse-reaction turbine
1.	Steam expands completely in the nozzle.	Steam expands partially in the moving blades and partially in the fixed blades.
2.	Pressure of steam remains constant during its flow through the blade passages.	Pressure of steam is not constant during its flow through moving and fixed blade passages.
3.	Steam flow over the blades is uniform due to symmetrical blade profile.	Steam flow over the blades varies due to aerofoil section (unsymmetrical) blade profile.
4.	Due to complete pressure drop in the nozzle, steam velocity and rotor speed are very high, resulting in centrifugal stresses acting on the blades. This may result in blade failure.	Since pressure drop occurs partially in the fixed blades. Steam velocity and rotor speed are not high. Hence, stresses developed are less.
5.	The initial and final velocity of steam over the blades is constant, or the final relative velocity can never be greater than the initial value, that is, $V_{r1} \geq V_{r2}$.	The initial and final relative velocities over the blades are not constant, and the final relative velocity is always greater than the initial relative velocity.
6.	As pressure drop in each stage is large, number of stages required is less. Hence, turbine is compact.	Due to small pressure drop in each stage, it requires large number of stages. Hence, size of the turbine is bigger.
7.	Part-load efficiency of the turbine is poor.	Part-load efficiency of the turbine is good due to more efficient blades.
8.	Speed of the turbine is very high, and hence requires compounding.	Speed of the turbine is low, and hence does not require compounding.
9.	Turbine is compact, and hence occupies less space.	Turbine is bigger, and hence requires more space.
10.	It is installed in small capacity power.	It is generally used in medium- and large-capacity power plants.

Advantages of steam turbines over other prime movers

1. The thermal efficiency of the turbine is higher and the expansion of steam continues up to the lower pressure in the steam turbines. This results in more work done by the turbine.
2. Turbines are highly simplified in construction as no reciprocating parts are present. This reduces the wear and tear of the parts due to friction, and hence results in less maintenance.
3. Turbines have longer life due to the absence of reciprocating parts.
4. Heat energy contained in unit mass of steam is very high, and hence there is more mechanical efficiency.
5. Balancing of masses is easier as turbine contains rotating masses.

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6. Noise and vibration are very less due to rotating parts, and hence reduces cost of foundation.
7. The steam turbines can take considerable overload with only a slight reduction in efficiency.
8. The steam consumption per hour per kilowatt of power developed is less and does not increase over years, that is, the specific steam consumption is less.
9. The steam turbines can be designed in units ranging from a few kilowatt to over 1000 MW in a single unit. Hence, they are well suited for use as prime movers in large power plants.
10. The steam turbines can work at very high pressures, and hence there are higher thermal efficiencies.
11. The steam turbines are extensively used to propel ships of high tonnage and also to drive high-speed generators, compressors, etc.

10.4.5 Blade Height Calculations

Let D = mean diameter of turbine wheel
B = breadth or height of blades
t = thickness of blades
n = number of blades
$V_s = x\, V_g$ = specific volume of steam, then

Mass flow rate through blades

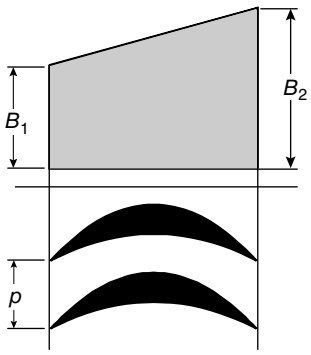


Fig. 10.14 *Blade Geometry Dimensions*

$$= \text{Area} \times \text{velocity}$$

$$mV_s = (\pi D - nt_1)\, B_1 \times V_{m1}$$

$$= (\pi D - nt_2)\, B_2 \times V_{m2} \qquad (1)$$

For most turbines, $V_{m1} = V_{m2} = V_m$ and blade height and thickness are constant.

If p is the pitch of the blades, then

$$mV_s = [n\,(p - t)\,B]\, V_m \qquad (2)$$

For reaction turbines, $t << p$

$$mV_s = npB\, V_m = \pi DB\, V_m \qquad (3)$$

where $np = \pi D$

Example 10.2

The simple impulse turbine has a mean blade speed of 200 m/s. The nozzles are inclined at 20° to the plane of rotation of the blades. The steam velocity from nozzles is 600 m/s. The turbine uses 3500 kg/h of steam. The absolute velocity at the exit is along the axis of the turbine. Determine the following:

(i) The inlet and exit angles of the blades
(ii) The power output of the turbine

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(iii) The diagram efficiency

(iv) The axial thrust (per kg steam per second)

Solution: $U = 200$ m/s; $\alpha_1 = 20°$; $V_1 = 600$ m/s; $m = 3500$ kg/h; $\alpha_2 = 90°$; $\beta_1 = \beta_2$

(i) The inlet and exit angles of the blades

$$V_{m1} = V_1 \sin\alpha_1 = 600 \sin 20° = 205.2 \text{ m/s}$$

$$\tan\beta_1 = \frac{CE}{BE} = \frac{V_{m1}}{V_1 \cos\alpha_1 - U} = \frac{205.2}{600 \cos 20° - 200}$$

$$\tan\beta_1 = 0.564$$

$$\beta_1 = \tan^{-1}(0.564) = 29.42°$$

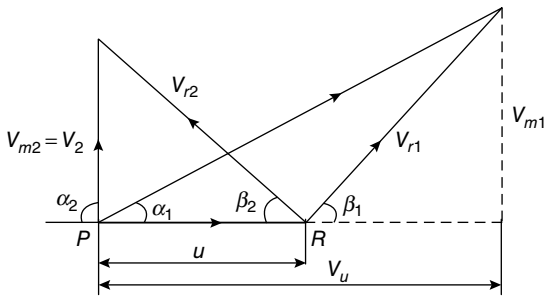
since $\beta_1 = \beta_2 = 29.42°$

(ii) Power output of the turbine

$$P = \frac{m \times U \times V_u}{1000}$$

$$= \frac{3500 \times 200 \times \left(600 \cos 20^{\circ} + 0\right)}{3600 \times 1000}$$

$$= 109.6 \text{ kW}$$



(iii) Diagram or blade efficiency $\eta_b = \dfrac{2 \times U \times \left(\Delta V_u\right)}{V_1^2}$

$$= \frac{2 \times 200 \times \left(600 \cos 20^{\circ}\right)}{600^2}$$

$$= 62.6\%$$

(iv) Axial thrust per kilogram steam per second = 1 $(V_{m1} - V_{m2})$

$$V_{m1} = V_1 \sin\alpha_1$$

$$= 600 \sin 20° = 205.2 \text{ m/s}$$

But, $\tan\beta_1 = \dfrac{V_{m2}}{U}$

$$V_{m2} = \tan 29.42 \times 200 = 112.78 \text{ m/s}$$

Axial thrust per kilogram team per second = $1 \times (205.2 - 112.78)$

$$= 92.42 \text{ N}$$

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Example 10.3

The following particulars relate to a two-row velocity compounded impulse wheel

Steam velocity at nozzle outlet = 650 m/s
Mean blade speed = 125 m/s
Nozzle outlet angle = 16°
Outlet angle first row of moving blades = 18°
Outlet angle fixed guide blades = 22°
Outlet angle second row of moving blades = 36°
Steam flow = 2.5 kg/s

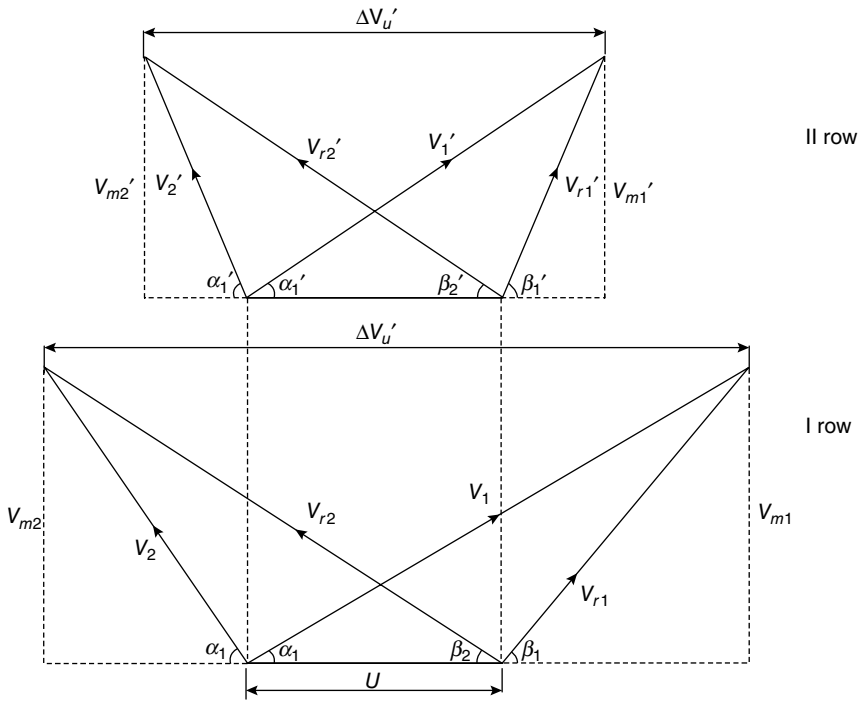
The ratio of the relative velocity at the outlet to that at the inlet is 0.84 for all blades. Determine the following:

(i) The axial thrust on the blades
(ii) The power developed
(iii) The efficiency of the wheel

Solution:

Given data:
First row exit diagram is drawn with $V_{r2} = 0.84\ V_{r1}$; $\beta_2 = 18°$; $\alpha_1 = 16°$
Second row inlet velocity diagram is drawn with

$$V_{r2'} = 0.84\ V_{r1'}$$



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$$\beta_{1'} = 36^\circ \; \alpha_{1'} = 22^\circ$$

From the diagram,

$$V_{m1} = 180 \text{ m/s} \; V_{m1'} = 122 \text{ m/s}$$

$$V_{m2} = 138 \text{ m/s} \; V_{m2'} = 107 \text{ m/s}$$

$$V_{u1} + V_{u2} = 924 \text{ m/s}$$

$$V_{u1'} + V_{u2'} = 324 \text{ m/s}$$

(i) Axial thrust on the blades

$$F_a = m\,[(V_{m1} - V_{m2}) + (V_{m1'} - V_{m2'})]$$

$$= 2.5\,[(180 - 138) + (122 - 107)]$$

$$= 142.5 \text{ N}$$

(ii) Power developed, $P = \dfrac{m\Delta V_u + \Delta V_u' U}{1000}$

$$= \frac{2.5\,924 + 324 \times 125}{1000}$$

$$= 390 \text{ kW}$$

(iii) Efficiency $\eta = \dfrac{(V_{u1} + V_{u2}) + (V_{u1'} + V_{u2'})}{\dfrac{V_1^2}{2}}$

$$= \frac{2\Delta V_u + \Delta V_u' U}{V_1^2}$$

$$= \frac{924 + 324\,125}{\dfrac{650^2}{2}}$$

$$= 73.8\%$$

Example 10.4

The velocity of steam that exits the nozzle of the impulse stage of a turbine is 400 m/s. The blades operate close to the maximum blading efficiency. The nozzle angle is 20°. Considering equiangular blades and neglecting blade friction, calculate for a steam flow of 0.6 kg/s, the diagram power and the diagram efficiency.

Solution:

Given data:

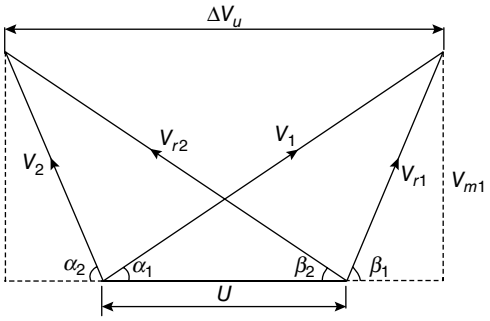
$V_1 = 400$ m/s; $m = 0.6$ kg/s; $\alpha_1 = 20^\circ$

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We have, maximum blade efficiency $= \dfrac{\cos \alpha_1}{2} = \dfrac{U}{V_1}$

i.e., $\dfrac{U}{400} = \dfrac{\cos 20°}{2} = 0.469$

$U = 187.94$ m/s



From inlet velocity triangle,

$V_{u1} = V_1 \cos\alpha_1 = 400 \cos 20° = 375.9$ m/s

$V_{m1} = V_1 \sin\alpha_1 = 400 \sin 20° = 136.8$ m/s

$$\tan\beta_1 = \frac{V_{m1}}{V_{u1} - U} = \frac{136.8}{375.9 - 187.94} = 0.7278$$

$\beta_1 = \tan - 1\ (0.7278) = 36.04°$

$V_{r2} \sin\beta_1 = V_{m1}$

$$V_{r2} = \frac{V_{m1}}{\sin \beta_1} = \frac{136.8}{\sin 36.04°} = 232.5 \text{ m/s}$$

Neglecting friction

$$V_{r1} = V_{r2} = 232.5 \text{ m/s}$$

As blades are equiangular,

$$\beta_1 = \beta_2 = 36.04°$$

$$V_{u2} = V_{r2} \cos\beta_1 - U$$

$$= 232.5 \cos 36.04° - 187.94$$

$$= 0.06 \text{ m/s}$$

$$V_u = V_{u4} + V_{u2} = 375.9 + 0.06 = 375.84 \text{ m/s}$$

Diagram power $P = \dfrac{m \times U \times \Delta V_u}{1000}$

$$P = \frac{0.6 \times 375.84 \times 187.94}{1000}$$

$$P = 42.38 \text{ kW}$$

Diagram efficiency or blade efficiency

$$\eta_b = \frac{2UV_u}{V_1^2}$$

$$= \frac{2 \times 187.94 \times 375.84}{400^2}$$

$$= 88.3\%$$

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Example 10.5

A single-stage steam turbine is supplied with steam at 5 bar, 200°C at the rate of 50 kg/min. It expands into a condenser at a pressure of 0.2 bar. The blade speed is 400 m/s. The nozzles are inclined at an angle of 20° to the plane of the wheel, and the outlet blade angle is 30°. Neglecting friction losses, determine the power developed, blade efficiency and stage efficiency.

Solution:

Given data:

$p_1 = 5$ bar; $p_2 = 0.2$ bar; $m = 50$ kg/min; $U = 400$ m/s; $\alpha_1 = 20°$; $\beta_2 = 30°$; $V_{r2} = V_{r1}$

From steam tables at 5 bar 200°C $h_1 = 2855.4$ kJ/kg

$$s_1 = 7.0592 \text{ kJ/kg K}$$

at 0.2 bar $h_{f2} = 251.5$ kJ/kg

$h_{fg2} = 2358.4$ kJ/kg

$s_{f2} = 0.8321$ kJ/kg

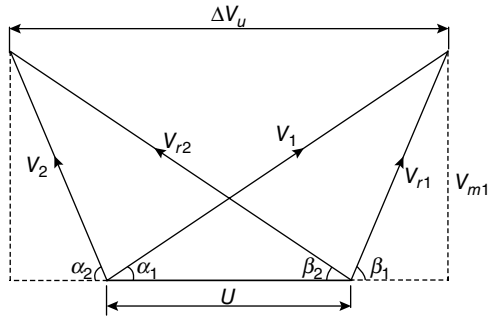
$s_{fg2} = 7.0773$ kJ/kg K

Steam expansion takes place isentropically $s_1 = s_2$

$$7.0592 = s_{f2} + x_2 s_{fg2}$$

$$7.0592 = 0.8321 + x_2 \times 7.0773$$

$$x_2 = \frac{7.0592 - 0.8321}{7.0773} = 0.88$$



Enthalpy of steam at 0.2 bar

$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$= 251.5 + 0.88 \times 2358.4$$

$$h_2 = 2326.9 \text{ kJ/kg}$$

Enthalpy drop $\Delta h = h_1 - h_2 = 2855.4 - 2326.9$

$$= 528.5 \text{ kJ/kg}$$

Velocity of steam entering the blades

$$V_1 = 44.7\sqrt{h_1 - h_2}$$

$$= 1028 \text{ m/s}$$

$$V_{u1} = V_1 \cos\alpha_1 = 1028 \cos 20° = 966 \text{ m/s}$$

$$V_{m1} = V_1 \sin\alpha_1 = 1028 \sin 20° = 351.6 \text{ m/s}$$

$$\tan\beta_1 = \frac{V_{m1}}{V_1 \cos\alpha_1 - U} = \frac{3516}{966 - 400} = 0.621$$

$$\beta_1 = \tan - 1\ (0.621) = 31.84°$$

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$$V_{r1} \sin\beta_1 = V_{m1}$$

$$V_{r1} = \frac{V_{m1}}{\sin\beta_1} = \frac{351.6}{\sin 31.84} = 666.5 \text{ m/s}$$

$$V_{u2} = V_{r2} \cos\beta_2 - U$$

$$= 666.5 \cos 30 - 400$$

$$V_{u2} = 177.2 \text{ m/s}$$

Power developed

$$P = \frac{m \times U \times \Delta V_u}{1000}$$

$$= \frac{50 \times (966 + 177.2) \times 400}{60 \times 1000}$$

$$= 381.06 \text{ kW}$$

Blade efficiency

$$\eta_b = \frac{2 \times U \times (\Delta V_u)}{V_1^2}$$

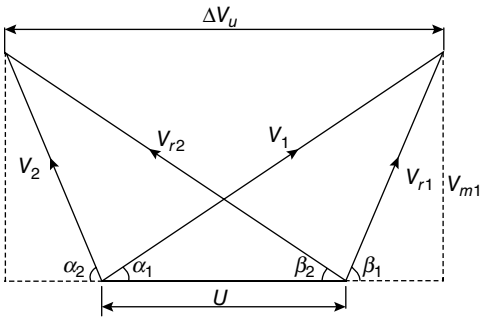
$$= \frac{2 \times 400 \times (966 + 177.2)}{1028^2}$$

$$= 86.5\%$$

Example 10.6

One stage of an impulse turbine consists of a converging nozzle ring and one ring of moving blades. The nozzles are inclined at 22° to the blades whose tip angles are both 35°. If the velocity of steam at the exit from nozzle is 660 m/s, find the blade speed so that the steam passes on without shock. Find the diagram efficiency neglecting losses if the blades are run at this speed.

Solution:
For an impulse turbine, the maximum blade efficiency is



$$\eta_{b\,max} = \frac{\cos^2\alpha_1}{2}(1 + KC)$$

K = blade velocity coefficient = 1

$$C = \frac{\cos^2\beta_2}{\cos^2\beta_1}$$

$$\eta_b = \frac{\cos^2_{\alpha 1}(1+1)}{2}$$

$$= \cos^2\alpha_1 = (\cos 22°)^2 = 0.86$$

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$$\rho_{opt} = \frac{\cos\alpha_1}{2}$$

$$\frac{U}{V_1} = \frac{\cos 22^\circ}{2}$$

$$U = V_1 \times 0.4636 = 660 \times 0.4636$$

$$= 306 \text{ m/s}$$

Example 10.7

Determine the isentropic enthalpy drop in the stage of Parson's reaction turbine, which has the following particulars:

Speed = 1500 rpm
Mean diameter of rotor = 1 m
Stage efficiency = 80%
Speed ratio = 0.7
Blade outlet angle = 20°

Solution:

Given data:

$N = 1500$ rpm; $D = 1$ m; $\eta_{stage} = 80\%$

$$\text{Speed ratio} = \frac{U}{V_1} = 0.7$$

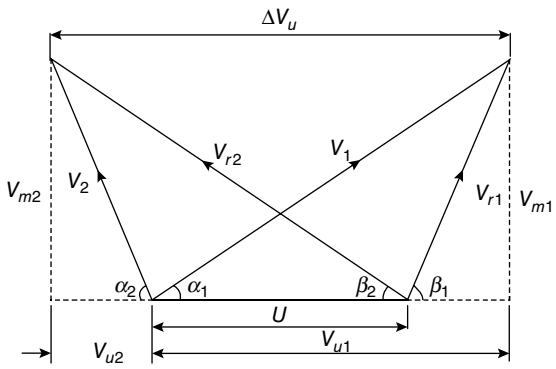
$$\beta_2 = \alpha_1 = 20^\circ$$

$$V_1 = V_{r2}$$

$$V_{r1} = V_2$$

We have, $U = \dfrac{\pi DN}{60} = \dfrac{\pi \times 1 \times 1500}{60}$

$$= 78.5 \text{ m/s}$$



$$0.7 = \frac{78.5}{V_1}$$

$$V_1 = \frac{78.5}{0.7} = 112.14 \text{ m/s}$$

$$V_{r1}^2 = V_1^2 + U^2 - 2UV_1 \cos\alpha_1$$

$$= (112.14)^2 + (78.5)^2 - 2 \times 112.14 \times 78.5 \cos 20^\circ$$

$$= 2193.42$$

$$\therefore \quad V_{r1} = 46.83 \text{ m/s}$$

Δh = Actual enthalpy drop for the stage

$$= \frac{1}{2}\left[\left(V_1^2 - V_2^2\right) + \left(V_{r2}^2 - V_{r1}^2\right)\right]$$

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Since $V_2 = V_{r1}$; $V_{r2} = V_1$

$$= \frac{1}{2}V_1^2 - V_{r1}^2$$

$$\Delta h = (112.14)^2 - (46.83)^2$$

$$\Delta h = 10.38 \text{ kJ/kg}$$

$$\text{Isentropic enthalpy drop} = \Delta h' = \frac{\Delta h}{\eta_{\text{stage}}} = \frac{10.38}{0.8} = 12.975 \text{ kJ/kg}$$

Example 10.8

A 50 per cent reaction turbine (with symmetrical velocity triangle) running at 400 rpm has the exit angle of the blades as 20°, and the velocity of steam relative to the blades at the exit is 1.35 times the mean blade speed. The steam flow rate is 8.33 kg/s, and at a particular stage the specific value is 1.381 m^3/kg. For this stage, calculate (i) a suitable blade height assuming the rotor mean diameter 12 times the blade height and (ii) the diagram work.

Solution:

Given data:

$N = 400$ rpm; $\alpha_1 = 20°$; $V_{r1} = V_1 = 1.35$ U; $m = 8.33$ kg/s; $V = 1.381$ m^3/kg; $D = 12$ h.

Axial flow velocity $V_{m1} = V_{m2} = V_m = V_1 \sin\alpha_1$

$$= 1.35 \text{ U } \sin 20$$

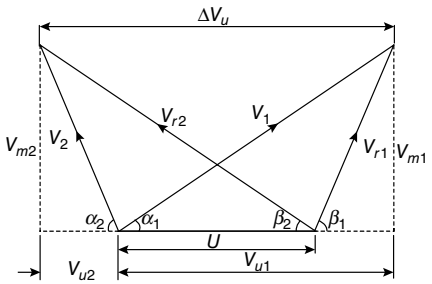
$$= 0.4617 \text{ U}$$

$$\text{Area of flow } A = \pi Dh = \pi D \frac{D}{12} = \frac{\pi D^2}{12}$$

$$\text{Mass flow rate } m = \frac{AU}{v} = 8.33 = \frac{A \times 0.4617\text{U}}{1.381}$$

$$UA = \frac{8.33 \times 1.381}{0.4617}$$

$$\frac{\pi DN}{60} \times \frac{\pi D^2}{12} = 24.916$$



$$\pi^2 D^3 = \frac{24.916 \times 720}{400} = 44.848$$

$$D^3 = 4.548$$

$$D = 1.656 \text{ m}$$

(i) Blade height $h = \frac{D}{12} = \frac{1.656}{12} = 0.138$ m

$$h = 138 \text{ mm}$$

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(ii) Diagram work

Diagram work $= mU\,(2\,V_{u1} + V_{u2})$

$= mU\,(2\,V_1 \cos\alpha_1 - U)$

$= 8.33 \times U\,(2 \times 1.35\,U \cos 20° - U)$

$= 8.33\,U^2\,(2 \times 1.35 \cos 20° - 1)$

$$= 8.33 \times \left(\frac{\pi \times D \times N}{60}\right)^2 \times 1.537$$

$$= 8.33 \times \left(\frac{\pi \times 1.656 \times 400}{60}\right)^2 \times 1.537$$

$= 15401.2$ W

$= 15.4$ kW

Example 10.9

300 kg/min of steam (2 bar 0.98 dry) flows through a given stage of a reaction turbine. The exit angle of fixed blades as well as moving blades is 20° and 3.68 kW of power is developed. If the rotor speed is 360 rpm and tip leakage is 5 per cent, calculate the mean drum diameter and the blade height. The axial flow velocity is 0.8 times the blade velocity. (Roorkee University)

Solution:

Mass flow rate of steam $m = \dfrac{300}{60} = 5$ kg/s

Pressure and dryness fraction $p = 2$ bar; $x = 0.98$

Power developed $P = 3.68$ kW

Speed $N = 360$ rpm

Tip leakage = 5%

Exit angles of fixed and moving blades $= \alpha_1 = \beta_2 = 20°$

Axial velocity $V_m = 0.8\,U$

Mean drum diameter D

$$\text{Mean blade velocity } U = \frac{\pi DN}{60} = \frac{\pi D \times 360}{60}$$

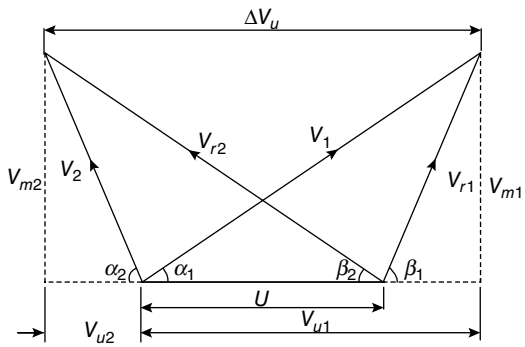
$$U = 18.85\,D \text{ m/s}$$

$$\text{Power developed } P = \frac{mUV_u}{1000}$$

$$3.67 = \frac{5 \times 0.95 \times 18.85 D \times V_u}{1000}$$

$$V_u = \frac{3.67 \times 1000}{5 \times 0.95 \times 18.85 D} = \frac{40.988}{D}$$

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Assuming Parson's reaction turbine

We have $V_{m1} = V_1 \sin\alpha_1$

$$V_1 = \frac{V_{m1}}{\sin\alpha_1} = \frac{0.8U}{\sin 20°} = \frac{0.8\times 18.85D}{\sin 20°}$$

$$V_1 = 44.091\ D$$

But, $V_u = 2\ V_1 \cos\alpha_1 - U$

$$\frac{40.988}{D} = 2 \times 44.091\ D \cos 20° - 18.65\ D$$

$$40.\ 988 = 64.01\ D^2$$

$D = 0.8$ m

$D = 800$ mm

Blade height, h

mean steam flow rate $m = \dfrac{\pi D h V_m}{x v_g}$

$$5 \times 0.95 = \frac{\pi \times 0.8 \times h \times V_1 \sin\alpha_1}{0.98 \times 0.885}$$

$$5 \times 0.95 = \frac{\pi \times 0.8 \times h \times 44.091 \times 0.8 \sin 20°}{0.98 \times 0.885}$$

$$h = \frac{4.1196}{30.3047}$$

$h = 0.1359$ m

$h = 135.9$ mm

Example 10.10

In a 50 per cent reaction turbine, the speed of rotation of blade group is 3000 rpm with mean blade velocity of 120 m/s. The velocity ratio is 0.8 and the exit angle of the blade is 20°. If the mean blade height is 30 mm, calculate the total steam flow rate through the turbine. Neglect the effect of blade-edge thickness of the annular area, but consider 10 per cent of the total steam flow rate as the tip leakage loss. The mean condition of steam in that blade group is found to be 2.7 bar and 0.95 dry.

Solution:

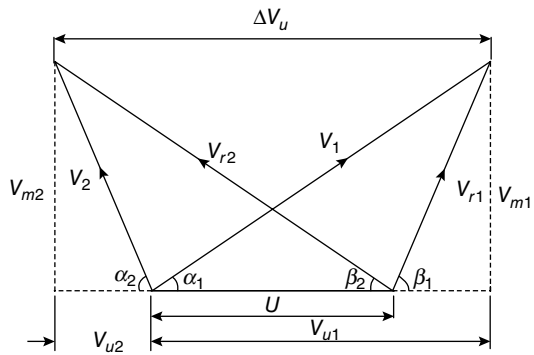
Given data:

$N = 3000$ rpm; $\alpha_1 = \beta_2 = 20°$; $U = 120$ m/s

$$\frac{U}{V_1} = 0.8$$

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$$\therefore V_1 = \frac{120}{0.8} = 150 \text{ m/s}$$



Blade speed $U = \frac{\pi DN}{60} = 120$

$$D = \frac{120 \times 60}{\pi \times 3000} = 0.764 \text{ m}$$

From steam tables at 2.7 bar

$$v_g = 0.668 \text{ m}^3/\text{kg}$$

$$\therefore v = 0.95 \times 0.668 = 0.6346 \text{ m}^3/\text{kg}$$

Flow area $A = \pi Dh = \pi \times 0.764 \times \frac{30}{1000} = 0.072 \text{ m}^2$

Flow velocity, $V_m = V_1 \sin\alpha_1 = 150 \sin 20° = 51.3$ m/s

Mass flow rate $m = \frac{AV_m}{v} = \frac{0.072 \times 5.13}{0.6346} = 5.82$ kg/s (since $V_m = V_{m2} = V_m$)

For 10% leakage of (total steam flow),

Total steam flow rate is = 6.467 kg/s

Example 10.11

A two-stage impulse turbine has two rows of moving blades and one row of intermediate fixed blades. Steam exit velocity from the nozzle is 600 m/s at an angle of 16°. If discharge angles of rotor and stator blades are 32° and 22°, respectively, find the blade velocity, rotor efficiency and specific steam consumption. Assume that 10 per cent loss takes place both in stator and rotor blades and steam exits axially.

Solution:

Given data:

Absolute steam velocity at inlet, $V_1 = 450$ m/s

Nozzle angle, $\alpha_1 = 16°$

Rotor discharge blade angles, ${}_1\beta_2 = {}_2\beta_2 = 32°$

Stator discharge blade angle, ${}_2\alpha_1 = 22°$

Blade velocity coefficient, $= 0.90$.

Construction of velocity triangles for the second row:

Since blade speed is not given, we have to first draw the velocity triangles for the second row and then proceed towards the first row.

(i) Draw *PR* of any convenient length. Draw a line *PS* perpendicular to *PR* as exit velocity is given as axial.

(ii) From point *R* measure angle ${}_2\beta_2 = 32°$ and draw a line passing through it so that it cuts the vertical line from *P* at *S*. Join *RS* and *PS*. This completes exit velocity triangle.

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(iii) Now with $_2V_{r1} = \frac{_2V_r 2}{0.90}$, find out the length $_2V_{r1}$. Draw a line from point P at angle of 22° to the horizontal. From point R, draw an arc equal to length $_2V_{r1}$ so that it cuts the line from P at Q. Join PQ and RQ. This completes the inlet velocity triangle.

From the triangles by measurement we have,

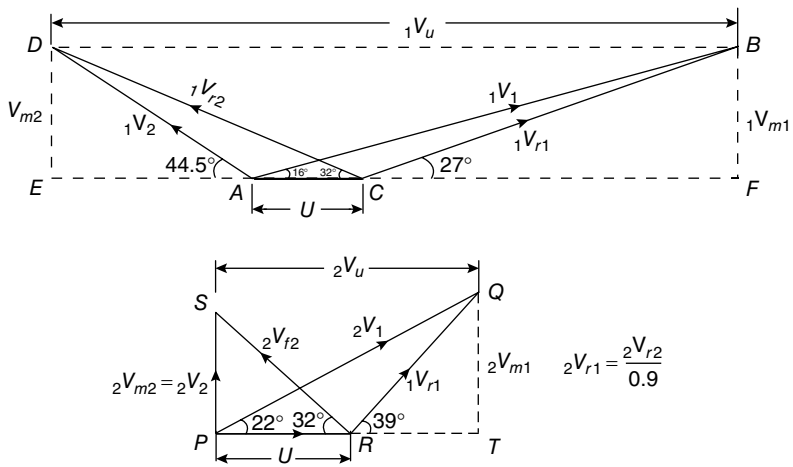
$$_2V_u = 6.1 \text{ cm}; \; U = 3 \text{ cm}$$
$$_2V_1 = 6.65 \text{ cm}; \; _2V_{r2} = 3.55 \text{ cm}$$

Construction of velocity triangles for the first row:

(i) Draw $PR = AB = U$. Taking $_1V_2 = \frac{_2V_1}{0.9}$, draw an arc with A as centre so that it cuts the line drawn from C at an angle of 32° to the horizontal line at D. Complete the exit velocity triangle. By measurement,

$$_2V_{r2} = 10.1 \text{ cm}$$

(ii) From A, draw a line at an angle of 22° to the horizontal. With C as centre cut an arc equal to length $_1V_{r1} = \frac{_1V_{r2}}{0.9}$ so that it cuts at point B. Join AB and CB. This completes the inlet velocity triangle.



Now from the triangle,

$$AB = 14.9 \text{ cm} = 600 \text{ m/s}$$
$$\therefore \; 1 \text{ cm} = 40.27 \text{ m/s}$$
$$\therefore \; _1V_u = 18.4 \times 40.27 = 740.97 \text{ m/s}$$
$$_2V_u = 6.15 \times 40.27 = 247.66 \text{ m/s}$$
$$U = 3 \times 40.27 = 120.81 \text{ m/s}$$

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Blade efficiency or rotor efficiency

$$\eta_b = \frac{2 \times U \times \left({}_1\Delta V_u + {}_2\Delta V_u\right)}{V_1^2} \times 100$$

$$= \frac{2 \times U \times \left(740.97 + 247.66\right)}{600^2} \times 100$$

$$= 66.35\%$$

Specific steam consumption

Power developed, $P = mU\,({}_1V_u + {}_2V_u) \times 10^{-3}$

$$1 \text{ kW} = m \times 120.81\,(740.97 + 247.66) \times 10^{-3}$$

$$= m \times 119.44$$

$\therefore$ Specific steam consumption

$$m = \frac{1}{119.44} \text{ kg/s/kW}$$

10.4.6 Bleeding in Steam Turbines

In steam power plants, it has become a standard practice to modify the usual cycle of operation to increase the thermal efficiency. One such practice is to extract steam at one or more points from a steam turbine when steam expands. This heat carried by steam is utilized to heat the feed water as shown in Figure 10.15.

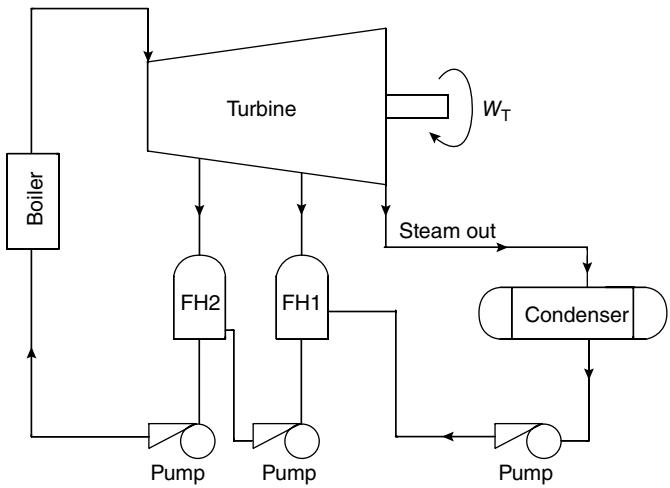


Fig. 10.15 *Turbine Bleeding*

The extracted steam produces heat by condensing either in a direct contact feed heater (FH) or in surface condenser. This boiler is supplied with high-temperature feed water. This process of tapping steam at intermediate points in the steam turbine is known as *bleeding*.

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10.5 LOSSES IN STEAM TURBINES

As steam expands through the turbine, both internal and external losses take place in the turbine. While internal losses are connected with the internal flow of steam within the turbine, external losses are mainly attributed to the losses outside the turbine casing.

1. *Internal losses*

Internal losses may occur at the following locations:

(i) *Blade friction*

Blade friction losses may be due to various reasons such as impingement losses, frictional losses, turning losses and wake losses. Impinging losses occur when jet of steam leaving the nozzle encounters the turbine blade edge, forming eddies. This happens if the edges are not smooth enough. When the steam passes through blade passages, frictional losses occur. The magnitude of loss depends on the roughness of the blades.

(ii) *Nozzle friction*

Major losses in nozzles happen due to the growth of the boundary layer and formation of eddies. The magnitude of loss depends on the frictional resistance of walls, length and height of the passage and the type of boundary layer (laminar or turbulent). A reaction stage has lower nozzle friction losses compared with an impulse stage, due to lower steam velocity enabling laminar flow in major length of the passage. It may be noted that losses increase with increased number of stages, as steam encounters higher surface area in the stages. Apart from this, nozzle design parameters such as size, shape, surface roughness, nozzle's spacing, divergence angle, etc., also contribute to frictional losses.

(iii) *Valve friction*

As steam leaves the generator, it has to pass through main and regulating valves, resulting in pressure and enthalpy drop. As this process is irreversible adiabatic, net turbine output decreases due to this. The amount of pressure drop varies between 3 and 5 per cent of the turbine inlet line pressure.

(iv) *Gland leakage*

Gland leakage is bound to happen between stages, along the inlet and exit portion of the casings. Losses also occur through the radial clearance between the stationary nozzle diaphragm and the rotating drum/shaft. Leakage is also observed at the high-pressure (HP) and low-pressure (LP) ends of the radial clearance between the shaft and the casing. At the HP end, steam leaks to atmosphere, while at the LP end, air leaks into the system. In order to reduce the leakage, labyrinth seals are used along with carbon rings and gland leak-off.

(v) *Carry-over loss*

As the steam flows from one stage to another stage, part of its kinetic energy is lost due to carry-over. Carry-over loss is accounted by introducing a factor known as 'carry-over coefficient'. Even if the steam is wet, part of kinetic energy is lost to drag the water molecules present in steam.

(vi) *Windage losses disc friction and part load admission*

Losses due to disc friction and partial admission of steam through nozzles over the blade periphery are collectively referred as windage losses. Disc friction forces are set up due

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to the surface friction arising out of relative motion between the steam and disc. It is a centrifugal action where surface friction is due to drag action between the disc surface and the steam. When steam is partially admitted, part of kinetic energy is utilized to drive away the already existing steam in the blade passage. Loss due to this is sometimes referred to as scavenging loss.

2. *External losses*

If the lubrication system is not efficient, losses in bearing and governing systems are not uncommon. This increases the power consumption by pumps. External heat losses also take place if turbine piping and chest are not properly insulated. Effectively insulated turbo-generator systems used in modern power plants have external losses limited to 2–3 per cent.

10.6 REHEAT FACTOR AND CONDITION LINE

Consider expansion of steam through an impulse turbine. If steam expands in a number of stages say $A_1\ B_1$, $A_2\ B_2$, etc., isentropically, the final state of steam say at pressure p_c is represented by B_5 as shown in Figure 10.16. Due to blade friction, steam is reheated to the stage A_2, A_3, etc. in each stage. If steam expands from chest pressure p_1 to condenser pressure p_c adiabatically and isentropically, then $A_1\ D$ represents Rankine or isentropic enthalpy drop.

The sum of adiabatic heat drops ($A_1\ B_1 + A_2\ B_2 + \ldots$) from the initial pressure down to any given pressure is known as cumulative enthalpy drop represented by h_c. A line drawn through the state points A_1 through A_6, which shows the actual state of steam exiting each stage is called condition line (dotted line passing through the state points A_1 to A_6).

10.6.1 Stage Efficiency per Stage

$$\eta_s = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

$$= \frac{A_1C_1}{A_1B_1}, \text{ etc.} \qquad (1)$$

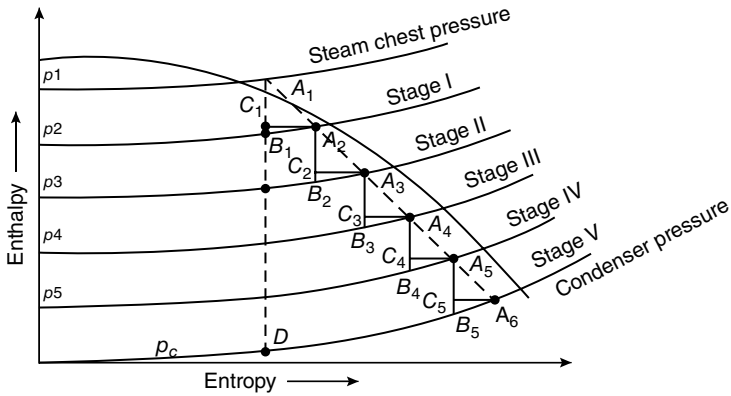


Fig. 10.16 *Turbine Reheat Stages*

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10.6.2 Reheat Factor

Reheat factor is defined as the ratio of cumulative enthalpy drop to adiabatic enthalpy drop:

$$R_f = \frac{\text{Cumulative enthalpy drop}}{\text{Adiabatic enthalpy drop}}$$

$$= \frac{A_1B_1 + A_2B_2 + \ldots}{A_1D}$$

$$R_f = \frac{h_c}{h_a} \tag{2}$$

The value of reheat factor depends on the turbine stage efficiency, initial and final pressure and superheat. The value of R_f ranges from 1.02 to 1.06.

10.6.3 Internal Turbine Efficiency

Internal turbine efficiency is defined as the ratio of sum of actual enthalpy drop in each stage to adiabatic enthalpy drop:

$$\eta_t = \frac{\text{Sum of actual enthalpy drop in each stage}}{\text{Adiabatic enthalpy drop}}$$

$$= \frac{A_1C_1 + A_2C_2 + \ldots}{A_1D}$$

$$= \frac{A_1C_1 + A_2C_2 + \ldots}{h_a} \tag{3}$$

$$= \frac{A_1C_1 + A_2C_2 + \ldots}{A_1B_1 + A_2B_2 + \ldots} \times \frac{A_1B_1 + A_2B_2 + \ldots}{h_a}$$

i.e., $$\eta_t = \eta_s \times R_f \tag{4}$$

10.7 GOVERNING OF TURBINES

A governor is a device used to maintain constant speed of the turbine irrespective of the load applied to it. Figure 10.17 shows a centrifugal flyball-type governor. When the load on the turbine is deduced, speed of the turbine increases. This results in an increased centrifugal force, throwing the flyballs apart, away from the shaft centreline. Due to this, the sleeve raises, causing the main valve to close, via the fulcrum and the lever attached to the sleeve. This reduces steam flow rate to the turbine.

In an oil-operated servo system (see Figure 10.17), the force exerted by the governor is amplified to move a frictionless pilot valve, controlling the flow of oil into a piston. This piston

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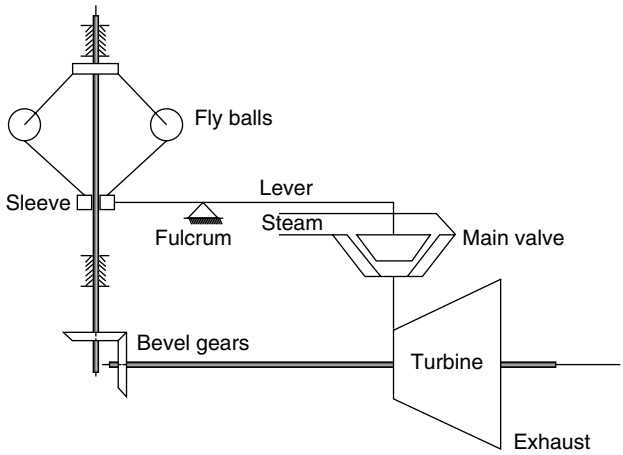


Fig. 10.17 *A Centrifugal Flyball-Type Governor*

operates with the high-pressure oil supplied to the governor valve in the desired position, thereby regulating the flow. The speed change of the governor from full-load condition to no-load condition varies between 3 and 4 per cent. The percentage change in regulated speed is given by the following equation:

$$\% \text{ change in regulated speed } = \frac{\left(N_F - N_0\right)}{N}$$

where N = speed at rated speed

N_0 = speed at no-load condition

N_F = Speed at full-load condition

Four different methods are available for governing a steam turbine. They are throttle governing, nozzle control governing, bypass governing and emergency governing. These are discussed below in detail.

1. *Throttle governing*

Power available at the shaft, P is given by $P = 2\Pi NT$, where T is the torque and N is the shaft speed in revolutions per second. From the above equation, it is evident that as load decreases, the shaft speed increases. As the main valve is closed partially to maintain the speed, power generated by the turbine decreases. Due to this restricted entry of the steam, steam is throttled from the initial pressure of p_b to p_{b1}. Due to this, the enthalpy drop reduces from $(h_1 - h_{2'})$ to $(h_3 - h_{4'})$. The steam consumption reduces further when the throttle valve is closed further, establishing a linear relationship. Using Willan's line method, steam consumption rate in the turbine is given by the following equation:

$$m = a + bL,$$

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where a = steam consumption under no-load conditions, kg/s
b = specific steam consumption, kg/s/kW
L = load on the turbine

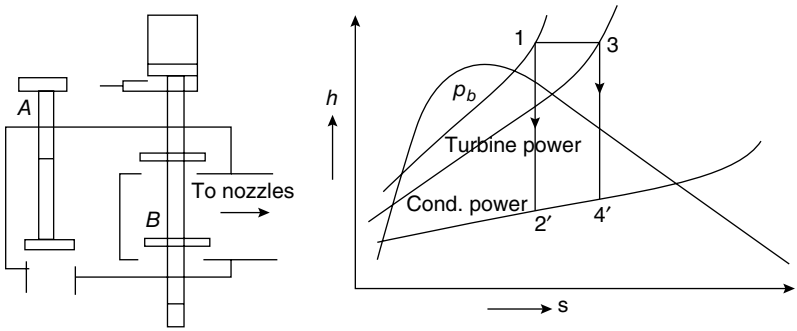


Fig. 10.18 *Throttle Governor*

The throttle valve is used to regulate steam flow during the start and stop of the turbine. In bigger turbines, the stop valve is hydraulically operated, whereas in smaller turbines, it could be manually operated. Figure 10.18 shows a throttle governing system.

2. *Nozzle control governing*

The major limitation of throttle governing is that the turbine efficiency reduces at low loads. Nozzle control governing is a better option in this case. As shown in Figure 10.19, the nozzles are arranged in sets, each set being controlled by a separate valve. When load on the turbine is decreased, the corresponding set of nozzles may be shut off.

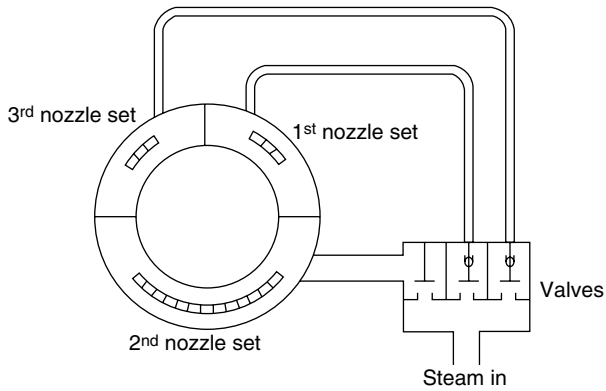


Fig. 10.19 *Nozzle Control Governing*

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3. *Bypass governing*

Where the requirement is more for power, additional steam is supplied to the turbine by providing bypass valves. This is usually done at the later stages of the turbine, and in a throttle-governed turbine as shown in Figure 10.20.

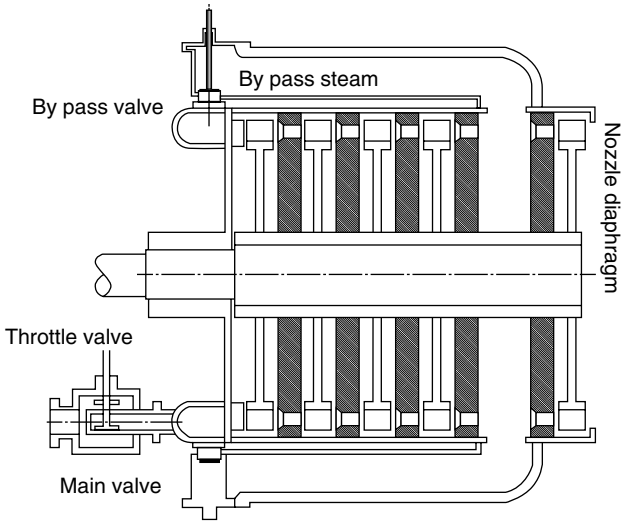


Fig. 10.20 *Bypass Governing*

4. *Emergency governor*

An emergency governor helps in tripping (stopping the turbine) when the turbine overspeeds (110 per cent of its rated speed) or the lubrication system fails or when balancing of the system is improper or condenser vacuum is disturbed (less). A typical emergency trip system contains a pin or weight placed on the turbine shaft, acting against a spring pressure. During normal operation, the centrifugal force acting on the pin is opposed by the spring pressure. Beyond the overspeed point, the centrifugal force becomes dominant, forcing the pin out and thereby striking a trigger. Due to this action, the spring closes the stop valve, shutting down the turbine.

10.8 QUESTIONS

10.8.1 Objective Questions

1. A steam turbine compared to a steam engine has
   (a) higher power output/unit weight
   (b) higher thermodynamic efficiency
   (c) higher part load efficiency and overload efficiency
   (d) All of the above

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2. Blade efficiency of a turbine is also known as
   (a) stage efficiency
   (b) diagram efficiency
   (c) isothermal efficiency
   (d) nozzle efficiency
3. Blade efficiency of an impulse turbine is maximum when blade speed ratio is
   (a) $\rho = \frac{\cos\alpha_1}{2}$
   (b) $\rho = \frac{\cos\alpha_1}{3}$
   (c) $\rho = 2\cos\alpha_1$
   (d) $\rho = \cos^2\alpha_1$
4. Maximum work done per kilogram of steam for an impulse turbine is given by
   (a) $W_{\max} = \frac{u^2(1+kC)}{1000}$
   (b) $W_{\max} = \frac{2u^2}{1000}$
   (c) $W_{\max} = \frac{u^2(1-kC)}{1000}$
   (d) none of these
5. In the absence of friction and for equiangular blades axial discharge, velocity of whirl for an impulse turbine is
   (a) $V_{w1} = 2u$
   (b) $V_{w1} = u$
   (c) $V_{w1} = \frac{u}{2}$
   (d) $V_{w1} = \frac{u}{3}$
6. If the blades are symmetrical and friction is absent, then maximum blade efficiency is given by (impulse turbine)
   (a) $\eta_{\text{b max}} = \cos^2\alpha_1$
   (b) $\eta_{\text{b max}} = \cos^2 2\alpha_1$
   (c) $\eta_{\text{b max}} = \cos^2\frac{\alpha_1}{2}$
   (d) $\eta_{\text{b max}} = \frac{1}{2}\cos^2\alpha_1$
7. Blade efficiency of an impulse turbine is given by
   (a) $\eta_{\text{b}} = \frac{2uV_w}{V_1^2}$
   (b) $\eta_b = \frac{2uV_w}{V_1}$
   (c) $\eta_{\text{b}} = \frac{uV_w}{V_1^2}$
   (d) None of these
8. Gross stage efficiency of an impulse turbine is given by
   (a) $\eta_{\text{b}} \times \eta_{\text{n}}$
   (b) $\eta_{\text{b}} / \eta_{\text{n}}$
   (c) $\eta_{\text{n}} / \eta_{\text{b}}$
   (d) None of these where η_n = nozzle efficiency, η_b = blade efficiency
9. The following method of governing a steam turbine is least efficient:
   (a) Nozzle control governing
   (b) Bypass governing
   (c) Throttle governing
   (d) The above combination of (a) and (b).

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10. Reheating in a steam turbine is done to
    (a) increase plant efficiency
    (b) reduce steam consumption rate
    (c) control moisture in the LP stages
    (d) None of the above

11. The axial thrust on the rotor of the turbine is due to
    (a) Radial component of absolute velocity
    (b) Tangential component of absolute velocity
    (c) The resultant of the above two components
    (d) either (a) or (b)

12. For an impulse turbine axial thrust on the rotor per kg of steam is given by
    (a) $F_a = \left(V_{f1} - V_{f2}\right)$
    (b) $F_a = m\left(V_{f1} - V_{f2}\right)$
    (c) $F_a = m\left(V_{f1} + V_{f2}\right)$
    (d) none of these

13. Carry-over loss in a steam turbine is due to
    (a) kinetic energy
    (b) potential energy
    (c) heat energy
    (d) datum energy

14. If a turbine (impulse) has η number of stages, then optimum blade speed ratio is
    (a) $\frac{\cos\alpha_1}{2\eta}$
    (b) $\frac{\cos_1\alpha_1}{4\eta}$
    (c) $\frac{\cos_1\alpha_1}{\eta}$
    (d) $\frac{\cos_1 2\alpha_1}{2\eta}$

15. Velocity compounding is advantages, due to
    (a) reduction in number of stages
    (b) reduction in stresses acting on turbine housing
    (c) reduction in working temperature
    (d) all the above

16. Degree of reaction is defined as
    (a) $R = \frac{h_b}{h_n + h_b}$
    (b) $R = \frac{h_n}{h_n + h_b}$
    (c) $R = \frac{h_n}{h_n - h_b}$
    (d) $R = \frac{h_b}{h_b - h_n}$

    where h_b = enthalpy drop in moving blades; h_n = enthalpy drop in fixed blades

17. Enthalpy drop in moving blades is given by
    (a) $\frac{V_{r2}^2 - \phi V_{r1}^2}{2\eta_n}$
    (b) $\frac{V_1^2 - \phi V_2^2}{2\eta_n}$
    (c) $\frac{V_{r2}^2 + \phi V_{r1}^2}{2\eta_n}$
    (d) $\frac{V_{r2}^2 - \phi V_{r1}^2}{\eta_n}$

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18. Enthalpy drop in fixed blades is given by

(a) $\dfrac{V_{r2}^2 - \phi V_{r1}^2}{2\eta_n}$ (b) $\dfrac{V_1^2 - \phi V_2^2}{2\eta_n}$

(c) $\dfrac{V_1^2 + \phi V_2^2}{2\eta_n}$ (d) $\dfrac{V_{r2}^2 - \phi V_{r1}^2}{\eta_n}$

19. In terms of inlet and outlet blade angles, degree of reaction is given by

(a) $R = \dfrac{V_f}{2u}(\cot\beta_2 - \cot\beta_1)$ (b) $R = \dfrac{V_f}{u}(\cot\beta_2 - \cot\beta_1)$

(c) $R = \dfrac{V_f}{2u}(\cot\beta_2 + \cot\beta_1)$ (d) $R = \dfrac{V_f}{u}(\cot\beta_2 + \cot\beta_1)$

20. For Parson's reaction turbine

(a) $\beta_1 = \alpha_1$ and $\beta_2 = \alpha_2$ (b) $\beta_1 = \alpha_2$ and $\beta_2 = \alpha_1$

(c) $\beta_1 \neq \alpha_1$ and $\beta_2 \neq \alpha_2$ (d) $\beta_1 \neq \alpha_2$ and $\beta_2 = \alpha_1$

21. Degree of reaction of a Parson's turbine is

(a) 1 (b) -1

(c) 0.5 (d) 0.25

22. For Parson's reaction turbine blade efficiency is maximum when blade speed ratio is

(a) $\rho = \cos\alpha_1$ (b) $\rho = 2\cos\alpha_1$

(c) $\rho = \dfrac{\cos\alpha_1}{2}$ (d) $\rho = 4\cos\alpha_1$

23. A Curtis turbine is an example of

(a) a reaction turbine (b) velocity compounded impulse turbine

(c) pressure compounded impulse turbine (d) none of these

24. A De-Laval turbine is

(a) a simple impulse turbine (b) a reaction turbine

(c) a velocity compounded reaction turbine (d) a pressure compounded impulse turbine

25. A single stage impulse turbine has

(a) high rpm of the shaft (b) low rpm of the shaft

(c) advantage over multistaging (d) low efficiency

26. For a reaction turbine maximum blade efficiency is given by

(a) $\dfrac{2\cos^2\alpha}{1+\cos^2\alpha}$ (b) $\dfrac{2\cos^2\alpha}{1+\cos\alpha}$

(c) $\dfrac{2\cos^2\alpha}{1+2\cos^2\alpha}$ (d) $\dfrac{\cos^2\alpha}{1+\cos\alpha}$

where α = Nozzle angle

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27. Ljungström turbine is an example of
    (a) a tangential flow reaction turbine
    (b) a radial flow reaction turbine
    (c) a single rotor reaction turbine
    (d) none of the above
28. A Rateau turbine is an example of
    (a) a reaction turbine
    (b) an impulse turbine
    (c) a pressure compounded impulse turbine
    (d) a velocity compounded impulse turbine
29. The value of reheat factor is
    (a) less than 1
    (b) greater than 1
    (c) equal to 1
    (d) may be more or less than 1
30. Reheat factor of a turbine is defined as
    (a) $R_f = h_c \times h_a$
    (b) $R_f = h_c / h_a$
    (c) $R_f = h_a / h_c$
    (d) $R_f = h_c - h_a$

    where h_c = cumulative enthalpy drop; h_a = adiabatic enthalpy drop
31. Internal turbine efficiency is defined as
    (a) $\eta_t = \eta_s \times R_f$
    (b) $\eta_t = \dfrac{\eta_s}{R_f}$
    (c) $\eta_t = \dfrac{R_f}{\eta_s}$
    (d) None of the above

    where η_s = stage efficiency/stage
32. Due to blade friction, the relative velocity of steam reduces (in impulse turbine) to about
    (a) 10–15 per cent
    (b) 80–85 per cent
    (c) 20–30 per cent
    (d) 5–10 per cent
33. Blade speed ratio is defined as
    (a) $\rho = \dfrac{u}{v_1}$
    (b) $\rho = \dfrac{u}{v_2}$
    (c) $\rho = \dfrac{v_1}{u}$
    (d) $\rho = u \times v_1$
34. A reaction turbine has
    (a) symmetrical blade profile
    (b) asymmetrical blade profile
    (c) aerofoil blade profile
    (d) None of the above
35. An impulse turbine has
    (a) symmetrical blade profile
    (b) asymmetrical blade profile
    (c) aerofoil blade profile
    (d) None of the above
36. The optimum value of nozzle angle is
    (a) 5–10°
    (b) 10–15°
    (c) 15–20°
    (d) 0–5°
37. Shaft speed of a De-Laval turbine is
    (a) 5000 rpm
    (b) 15,000 rpm
    (c) 20,000 rpm
    (d) 30,000 rpm

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Answers

1. d	2. b	3. a	4. a	5. c	6. a	7. b	8. a	9. c	10. c
11. b	12. a	13. a	14. a	15. d	16. a	17.	18. b	19. a	20. b
21. c	22. a	23. b	24. a	25. a	26. a	27. b	28. c	29. d	30. b
31. a	32. a	33. a	34. b	35. a	36. c	37. d			

10.8.2 Review Questions

1. Define steam turbine. Explain different types of steam turbines.
2. Derive the condition for maximum efficiency for impulse and show that the maximum efficiency is $\cos^2\alpha$, where α is the angle at which the steam enters the blades.
3. Define reheat factor and deduce the equation for the same.
4. What is the fundamental difference between the operation of impulse and reaction turbines? Explain the same with neat sketches.
5. What is compounding of steam turbine? What are the advantages of compounding? Illustrate with a diagram the pressure compounding of a steam turbine.
6. Define speed ratio, blade velocity coefficient, diagram or blade efficiency and stage efficiency in connection with steam turbines. Explain the importance of each in the design of steam turbines.
7. Arrive at an equation for maximum blade efficiency for a single-impulse turbine. Find the condition for maximum blade efficiency. In addition, determine the maximum work done/kg of steam and maximum blade efficiency.
8. Define 'degree of reaction' in connection with steam turbine and prove that $R = \frac{V_f}{2U}\left(\cot\beta_2 - \cot\beta_1\right)$ when $V_{m1} = V_{m2} = V_1$. Further prove that the moving and fixed blades should have the same shape for a 50 per cent reaction.
9. For a single stage reaction turbine prove that the diagram or blade efficiency is given by

$$\eta_b = 2 - \frac{2}{1 + 2\rho\cos\alpha_1 - \alpha_2}$$

when $R = 50\%$ and $V_{m1} = V_{m2}$

Also prove that blade efficiency is given by

$$(\eta_b)_{\max} = \frac{2\cos^2\alpha_1}{1+\cos^2\alpha_1}$$

10. For an impulse steam turbine, the following data are given:

Velocity of steam entering the wheel	= 450 m/s
Nozzle angle	= 18°
Moving blade exit angle	= 13°
Blade speed	= 180 m/s
Weight of steam supplied	= 3 kg/s
Velocity coefficient of blade	= 0.85

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Determine

(i) Power developed

(ii) Axial thrust

(iii) Diagram efficiency

11. A single-stage impulse steam turbine rotor has diameter 1.2 m and runs at 300 rpm. The nozzle angle is 18° and the blade speed ratio is 0.42. The ratio of relative velocity at outlet to relative velocity at the inlet is 0.9. The outlet of blade is 3° small than the inlet angle. The steam flow rate is 5 kg/s.

    Draw the velocity diagram and find velocity of whirl, axial thrust on bearing, blade angles and power developed.

    (**Ans**: 46.5 m/s; 13.5 N; 30°, 27°; 4.385 kW)

12. In an impulse turbine, the steam issues out of the nozzle at a velocity of 900 m/s. The mean blade velocity is 300 m/s. The nozzle angle is 20°. The mass flow rate of steam is 600 kg/min. Friction factor is 0.8. Determine the following:

    (i) Blade angles

    (ii) Power developed

    (iii) Blade efficiency

13. In a De Laval turbine, the steam issues from the nozzles with a velocity of 900 m/s. The nozzle angle is 20°. Mean blade velocity is 350 m/s. The blades are symmetrical. The mass flow rate is 600 kg/min. Friction factor is 0.8. Determine (i) blade angles (ii) power developed (iii) blade efficiency (iv) stage efficiency if nozzle efficiency is 93 per cent.

14. Steam issues from the nozzles of a De Laval turbine with a velocity of 914.4 m/s. The nozzle angle is 20°, the mean diameter of the blades is 254 mm, and the speed of rotation 20,000 rpm. The mass flow through the turbine nozzles and blading is 0.18 kg/s. If the friction loss in the blade channels is 33 per cent of the kinetic energy corresponding to the relative velocity at the inlet to the blades, draw the velocity diagram and calculate the following:

    (i) Velocity of whirl

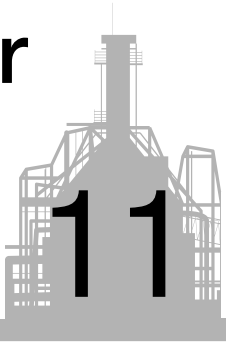
    (ii) Tangential force on blades

    (iii) Axial force on blades

    (iv) Work done on blades per second Author Queries:

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Steam Condenser and Circulating Water Systems



11

Contents

11.1 Ntroduction to condenser and its necessity
11.2 Classification and types of condensers
11.3 Surface condenser performance
11.4 Cooling tower
11.5 Cooling ponds
11.6 Corrosion in condensers and boilers
11.7 Corrosion in boilers
11.8 Questions

11.1 INTRODUCTION TO CONDENSER AND ITS NECESSITY

Condenser is an essential component of a modern steam condensing plant that increases the plant efficiency. It is a closed vessel in which high-pressure steam from the steam turbine condenses into low pressure. Different types of steam condensers are used in a steam condensing plant. When steam condenses, it forms hot water in the condenser. The heat contained in hot water coming from the condenser is rejected in either a cooling tower or cooling pond before re-circulating it in the water circulating system. This chapter deals with different types of condensers and cooling towers in brief.

When steam expands in a turbine, the pressure at the turbine exhaust will always be lower as shown in Figure 11.1 (see *T-s* diagram).

Figure 11.1 shows expansion of steam in a turbine from boiler pressure at point 1 to condenser pressure at point 2. The work due to expansion in the turbine is Δh_2. If the plant is working without a condenser, then expansion in the turbine takes place from point 1 to 2′. The resulting expansion work is Δh_1 which is lesser than Δh_2. Thus, a steam condensing plant results in higher turbine work and greater plant efficiency.

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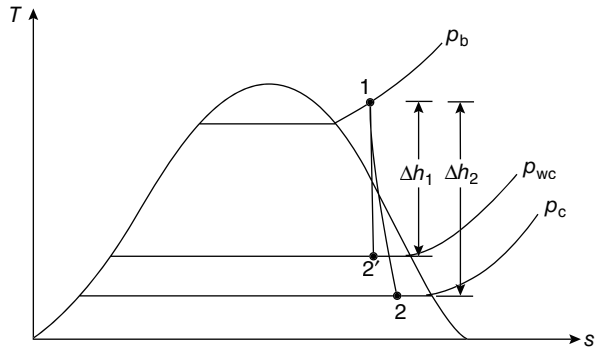


Fig. 11.1 *Expansion of Steam in Turbine*

11.1.1 Function of a Condenser

The primary function of the condenser is to condense the exhaust steam from the turbine. In doing so, high-quality feedwater can be reused in the cycle. In addition to this, a much more useful function is performed by the condenser. The steam exhausted in the turbine is condensed in the condenser at a low pressure (vacuum) by circulating cooling water. This pressure is equal to the saturation pressure corresponding to the condensing pressure of steam, and it is a function of cooling water temperature. Thus, the lower the exhaust pressure, the greater the turbine work. By lowering the turbine back pressure, the condenser increases the turbine work, increases the plant efficiency and reduces steam flow for a given plant output.

11.1.2 Elements of a Condensing Plant

Figure 11.2 shows a schematic diagram of a modern condensing plant. Elements of such a plant are condenser, condensate extraction pump, condenser cooling water recirculation pump, makeup water pump, air ejector pump, boiler feed pump and cooling tower.

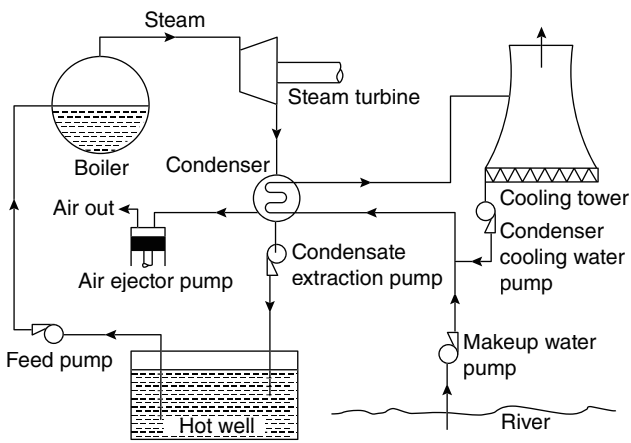


Fig. 11.2 *Steam Condensing Plant*

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1. *Condenser*

The primary function of the condenser is to condense steam that comes out of the turbine. It is a shell and tube heat exchanger in which heat exchange takes place between cold circulating water and hot steam. As the requirement of cooling water is extremely large, a steam condensing plant should be installed near a water source.

2. *Condensate extraction pump*

The function of the condensate extraction pump is to extract the condensate (at vacuum) from the condenser and feed it to the hot well.

3. *Cooling water re-circulating pump*

The function of the condensate cooling water re-circulating pump is to supply cooling water coming from the cooling tower back to the condenser.

4. *Makeup water pump*

The function of the makeup water pump is to supply makeup water to the condenser. Makeup water is necessary to account for water loss due to leakage and evaporation.

5. *Air ejector pump*

The function of the air ejector pump is to eject air that has leaked into the condenser through the sealings. Thus, it maintains constant vacuum in the condenser.

6. *Boiler feed pump*

The function of the boiler feed pump is to supply high-pressure feedwater to the boiler drum. It is a multi-stage centrifugal pump that operates above boiler pressure.

7. *Cooling tower*

Cooling tower is necessary whenever the source of water near the steam plant is not large. It is a hyperbolic structure in which condensate water coming from the condenser is cooled either naturally or artificially. In a cooling tower, hot condensate is cooled by evaporating water itself. Due to this, some portion of cooling water is lost to the atmosphere. This loss is compensated by using makeup water.

11.2 CLASSIFICATION AND TYPES OF CONDENSERS

Condensers are mainly classified into *direct-contact condensers* and *surface condensers*.

11.2.1 Direct-Contact Condensers

Direct-contact condensers are also known as *mixing*-type or *jet* condensers. In such type of condensers, exhaust steam coming from the turbine mixes directly with the circulating cooling water. Steam condenses into cooling water. These condensers are used where large quantity of clean cooling water is available. However, the drawback of using these condensers is that pure condensate is lost to the mixing water that cannot be re-circulated.

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The flow in these condensers may be either *parallel flow* or *counter flow*, where cooling water and steam move in same direction or in opposite direction, respectively.

Mainly three types of jet condensers are used.

11.2.1.1 Low-Level Jet Condenser

In a low-level jet condenser shown in Figure 11.3, water from the cooling pond enters the condenser due to the vacuum created in the condenser. Water, after passing over the perforated trays, breaks up into smaller particles, providing more surface area of contact. Hot steam from the turbine exhaust flowing in the opposite direction of water gets condensed while coming in contact with water sprays. The condensate collected at the bottom portion of the condenser is pumped to the hot well by the condensate extraction pump. Based on the quality of water, the condensate is either supplied to the boiler or returned to the pond.

The height of the condenser top should not exceed 10 m from the surface of the cooling pond. This ensures that proper vacuum be maintained in the condenser.

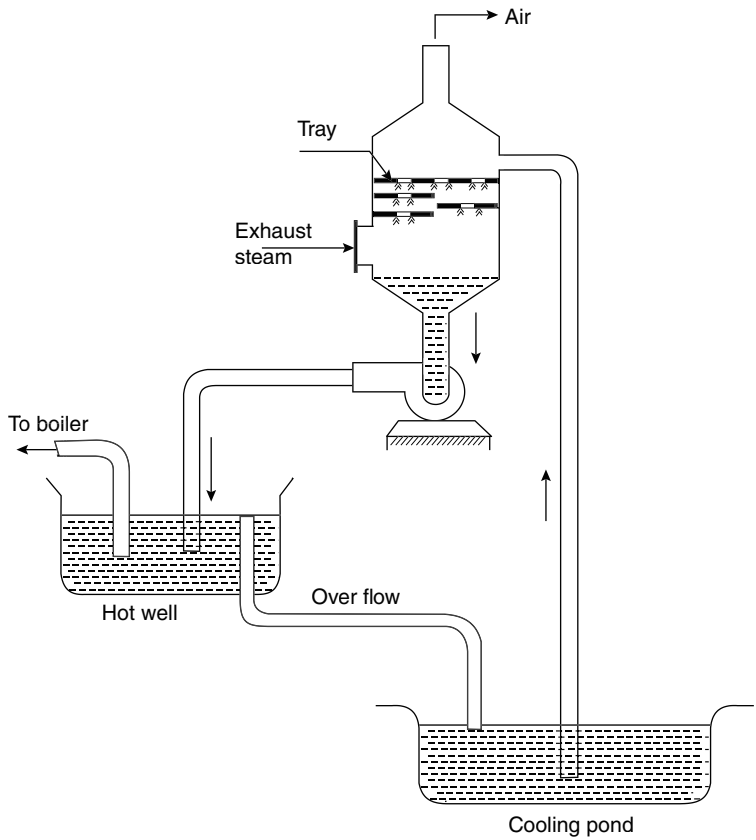


Fig. 11.3 *Low-Level Jet Condenser*

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11.2.1.2 High-Level Jet Condenser or Barometric Condenser

In a high-level jet condenser, a pipe of height more than 10.3 m connects the condenser and hot-well. This pipe is also known as *barometric leg*. Top end of the pipe connects the condenser whereas bottom end is immersed in the hot well opened to the atmosphere. Owing to this pressure difference, the condensate collected in the condenser flows to the hot well by gravity. Thus, barometric leg eliminates the use of condensate extraction pump, thereby ensuring downward flow of condensate towards the hot well.

A typical high-level jet condenser is shown in Figure 11.4. Water from the cooling pond is supplied to the condenser by using a circulating pump. Water entering the condenser passes over the trays and breaks up into smaller droplets. Water droplets mix with the exhaust steam that comes from the turbine. Steam gets condensed and gets collected at the bottom of the condenser from where it flows to the hot well purely by gravity. The use of *barometric leg* ensures that there will be no flow of condensate back into the steam line. Air released in the condenser can be removed by the air ejection pump that is fitted at the top of the condenser.

This condenser is not suitable under variable load conditions. At part load, it requires more quantity of water per kilo-watt of power generated.

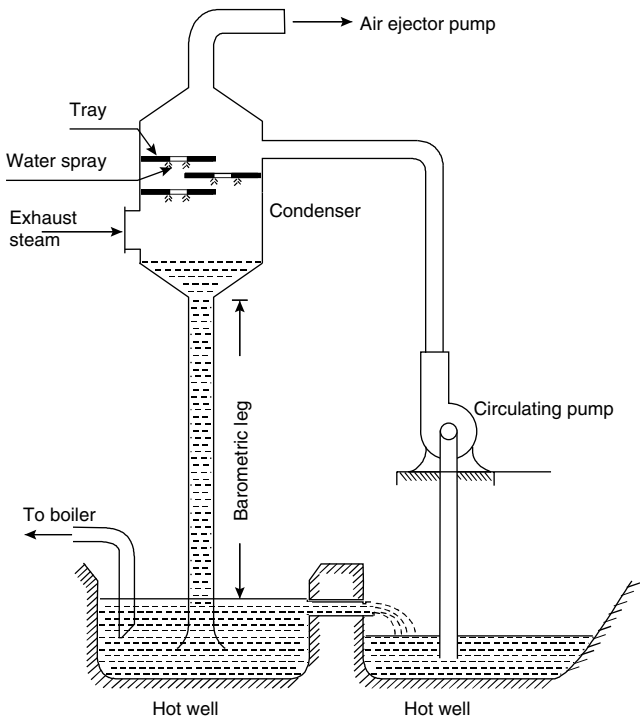


Fig. 11.4 *Barometric Condenser*

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11.2.1.3 Ejector Condenser

This is a low-level-type condenser that operates without the extraction pump. It consists of a venture-shaped conduit fitted with a series of convergent water nozzles as shown in Figure 11.5. Exhaust steam that comes from the turbine flows into conduit and mixes with water that comes from the water nozzles. Flow of cooling water through the nozzles creates partial vacuum that draws the exhaust steam coming from the turbine via a non-return valve. The mixture of condensate, evaporated air and cooling water is discharged through the diverging cone or diffuser into the hot well. The accidental rushing of water from the hot well into the turbine exhaust in the event of failure of cooling water supply is prevented by providing a non-return valve in the steam pipeline.

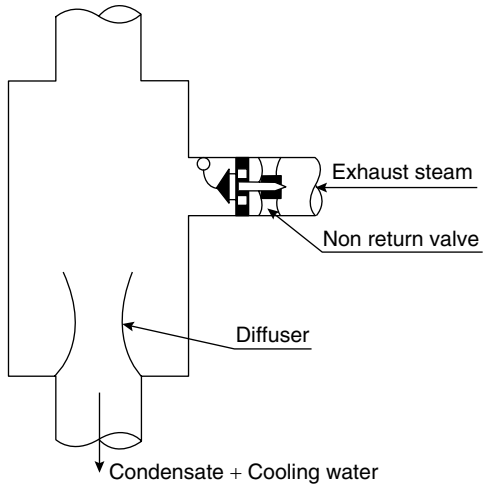


Fig. 11.5 *Ejector Condenser*

11.2.1.4 Advantages and Disadvantages of Jet Condenser

Advantages

(i) It occupies less space.
(ii) It requires less quantity of cooling water per kilogram of steam condensed.
(iii) It involves low initial cost and maintenance cost.

Disadvantages

(i) Due to direct mixing of the condensate with cooling water, the condensate water becomes impure, which cannot be directly used in the boiler.
(ii) It involves higher pumping cost of water. This increases the power consumed by the pump.
(iii) The power required to run the air pump is higher than that required for a surface condenser.
(iv) If the condensate extraction pump fails, then the turbine gets flooded with water.

11.2.2 Surface Condensers

Surface condensers are also known as non-mixing-type condensers. In such condensers, cooling water flows through the tubes surrounded by hot exhaust steam. Heat flows from hot stream to cold stream resulting in condensation of steam.

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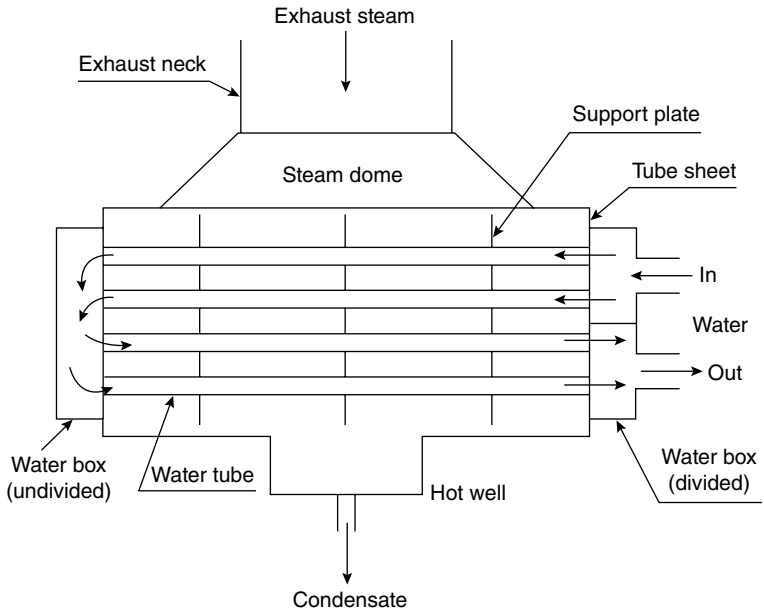


Fig. 11.6 *Two-Pass Surface Condenser*

Figure 11.6 shows a schematic diagram of a two-pass surface condenser. It consists of an air-tight cylindrical steel shell with water boxes on either side. The right-hand-side water box is divided to allow for the two water passes. The left-hand-side water box is undivided for ensuring free flow of water. The water tubes are rolled at each end into tube sheets. In between the tube sheets, steel support plates are provided at intermediate points to prevent tube vibration.

In a two-pass surface condenser, cooling water enters the upper half of the condenser tubes and leaves through the lower half. Exhaust steam condensed over the tubes in collected in the hot well. A single-pass surface condenser with same number of tubes and heat transfer area and with the same velocity of water requires twice as much water flow, but resulting in only half the temperature rise and lower condenser pressure.

Depending on the path of steam flow, surface condensers are broadly classified into two types as follows:

11.2.2.1 Down Flow Condenser

In this type of condenser, the air cooling section is separated from condenser tubes by a baffle plate. The air cooling section has denser tubes per unit area. Steam from the turbine enters from the top of the condenser and surrounds all over the tubes. Due to rapid heat transfer in the air cooling section, evaporated air cools faster which is removed by air extraction pump (see Figure 11.7).

11.2.2.2 Central Flow Condenser

It is a modification of downward flow condenser. In this condenser, steam flows over the tube periphery radially towards the centre. Air is extracted from the central cooling section (see Figure 11.8).

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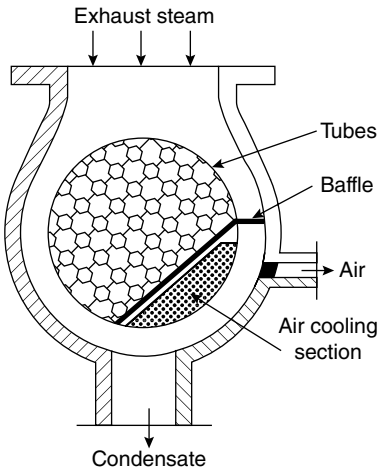


Fig. 11.7 *Down Flow-Type Surface Condenser*

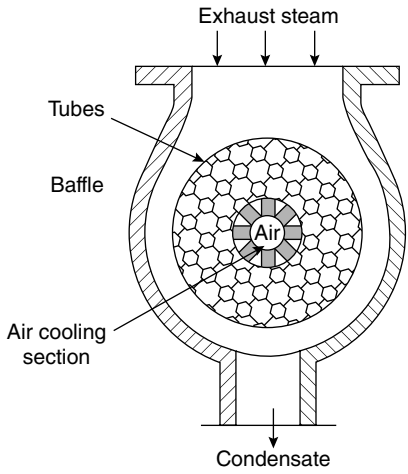


Fig. 11.8 *Central Flow-Type Surface Condenser*

11.2.2.3 Advantages and Disadvantages of Surface Condenser

Advantages

(i) Due to non-mixing of steam and cooling water, the condensate formed is pure, and hence can be used as a boiler feed.

(ii) As steam expands to the lowest possible pressure (due to higher vacuum maintained), the work done per kilogram of steam is high.

Disadvantages

(i) Chances of fouling of water tubes are more due to the accumulation of foreign matter. This ultimately results in loss of vacuum in the condenser.

(ii) Space requirement is large.

(iii) Higher installation, operation and maintenance cost.

(iv) Periodic cleaning of tubes is necessary.

11.2.3 Evaporative Condenser

In evaporative condenser, water is sprayed over the steam pipes carrying exhaust steam from the turbine using water nozzles. Such a typical arrangement is shown in Figure 11.9. The water molecules accumulated over the steam pipes form very thin film over them. This film comes in contact with air drawn over the surface by operating induced draft fan. Due to this action, thin film of water vaporizes continuously as it forms and is taken away by the air stream, resulting in rapid heat transfer from steam to water. This method of cooling is known as *evaporative cooling*. The water particles carried away by air stream are trapped using water eliminators. Such system requires makeup water arrangement as some water is always lost in the form of vapours to air stream.

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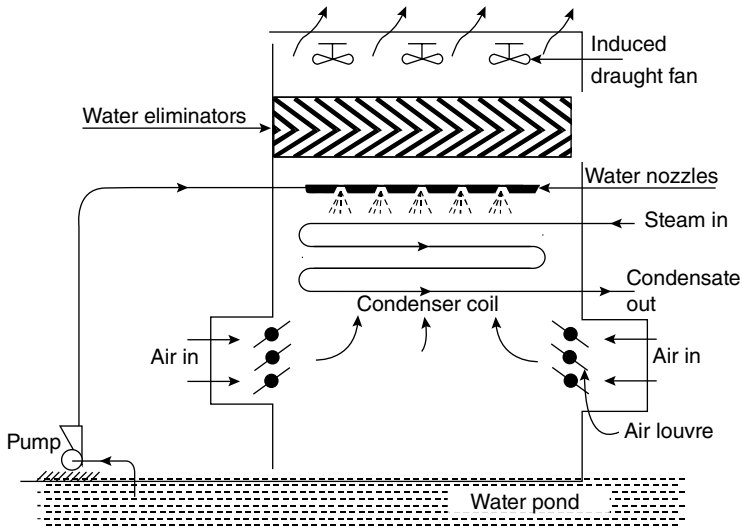


Fig. 11.9 *Evaporative Condenser*

Advantages

(i) It is simple in construction and lower installation cost.

(ii) It requires about 10 per cent less water as compared with surface condenser. Hence, it is preferred in places having fewer water supplies.

(iii) It is suitable for dry weather conditions.

Disadvantages

(i) Due to low vacuum maintained, the work done per kilogram of steam is not high.

(ii) It is used in small power plants.

11.3 SURFACE CONDENSER PERFORMANCE

11.3.3 Vacuum and Vacuum Efficiency

For maximum work done per kilogram of steam, it is very essential to maintain condenser vacuum. Condenser vacuum is destroyed due to leakage of air into it. Air leakage into the condenser may be due to the following reasons:

- Air may leak through the joints and sealings of the condenser.
- Air may enter in dissolved form in feedwater along with steam.
- In case of jet condensers, air leakage is possible along with injection water.
- Air leakage into the condenser has the following ill effects:
- It destroys condenser vacuum reducing work done per kilogram of steam.
- It forms a thin non-conducting layer over the tubes, reducing rate of heat transfer and ultimately condensation.

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11.3.3.1 Vacuum Efficiency

Vacuum in the condenser is measured in millimetre of mercury (Hg).

If H_b = barometric pressure head in millimetre of Hg
H_v = condenser vacuum in millimetre of Hg
Then, absolute pressure = $(H_b - H_v)$ millimetre of Hg
That is,

$$H_{abs} = (H_b - H_v) \tag{1}$$

In Eqn. (1), barometric head varies as atmospheric pressure. Thus for calculation, standard atmospheric pressure at sea level which is 760 mm of Hg is considered.

Therefore,

$$\text{Corrected vacuum} = 760 - (H_b - H_v) \tag{2}$$

Performance of the condenser depends on the vacuum maintained in it. The lowest possible pressure that can be maintained in the condenser is equal to the saturation pressure of steam corresponding to the temperature of water entering the condenser. But due to leakages of air, this pressure is not maintained. This difference is generally expressed in terms of vacuum efficiency.

Let p_a = partial pressure of air in the condenser

p_b = barometric pressure

p_s = saturation pressure of steam corresponding to cooling water inlet temperature

From Dalton's law of partial pressure,

Total pressure = $p_t = p_a + p_s$

Vacuum efficiency is defined as the ratio of actual vacuum to ideal vacuum.

$$\eta_{vac} = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}}$$

$$\eta_{vac} = \frac{p_b - p_t}{p_b - p_s}$$

11.3.3.2 Condenser Efficiency

Efficiency of the condenser is usually expressed in terms of the ratio of actual cooling water temperature rise to maximum possible temperature rise of cooling water.

If T_{wi} = cooling water temperature at inlet
T_{wo} = cooling water temperature at outlet
T_s = steam temperature at absolute condenser pressure
Then, condenser efficiency

$$\eta_{cond} = \frac{T_{wo} - T_{wi}}{T_s - T_{wi}}$$

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Example 11.1

A closed vessel of 0.7 m^3 capacity contains saturated water vapour and air at a temperature of 40°C and a pressure of 0.2 bar. Due to air leakage into the vessel, the pressure rises to 0.30 bar and the temperature falls to 38°C. Calculate the mass of air that has leaked in.

Solution:

From steam tables* partial pressure of steam at 40°C

$$p_s = 0.07384 \text{ bar}$$

From Dalton's law, partial pressure of air will be

$$p_a = 0.2 - 0.07384 = 0.12616 \text{ bar}$$

$$\text{Mass of air present, } m_a = \frac{p_a V}{RT}$$

$$= \frac{0.12616 \times 10^5 \times 0.7}{287 \times 313} = 0.098 \text{ kg}$$

From steam tables at 38°C

Partial pressure of steam = 0.06682 bar

Partial pressure of air, $p_a = 0.3 - 0.06682 = 0.233$ bar

$$\text{New mass of air present} = \frac{0.233 \times 10^5 \times 0.7}{287 \times (38 + 273)} = 0.183$$

Air leakage = 0.183 − 0.098 = 0.085 kg

Example 11.2

In a condenser test, the following observations were made.

Vacuum = 70.1 cm of mercury
Barometer = 76.3 cm of mercury
Mean temperature of condensation = 35°C
Hot well temperature = 29°C
Amount of cooling water = 46,270 kg/hr
Inlet temperature of cooling water = 16.75°C
Outlet temperature of cooling water = 31.15°C
Amount of condensate = 1,188 kg/h

Determine the following:

(1) The amount of air present per cubic metre of condenser volume
(2) The state of steam entering the condenser
(3) The vacuum efficiency

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Solution:

$$\text{Pressure in condenser} = (76.3 - 70.1) \times \frac{1.013}{76}$$

$$= 0.0826 \text{ bar (absolute)}$$

From steam tables* partial pressure of steam at 35°C = 0.0559 bar
Partial pressure of air, $p_a = 0.0826 - 0.0559 = 0.0267$ bar

(1) Amount of air $m_a = \dfrac{p_a V}{RT} = \dfrac{0.0267 \times 10^5 \times 1}{287 \times 308} = 0.03 \text{ kg}$

(2) State of steam

$$\text{Heat to coolant} = \frac{46270 \times (31.15 - 16.75) \times 4.187}{1188}$$

$$= 2348.27 \text{ kJ}$$

From steam tables* condenser pressure of 0.0827 bar

Latent heat of steam = 2399.57 kJ and $T_S = 42°C$

Steam gets undercooled from 35°C to 29°C. Neglecting losses

At 42°C, $h_{fw} = 175.4$ kJ/kg

At 29°C, $h_{fc} = 122.5$ kJ/kg

$$175.4 + x \times 2399.57 - 122.5 = 2348.27$$

Dryness fraction of steam,

$$x = 0.9566$$

(3) Vacuum efficiency

$$\text{Vacuum efficiency} = \frac{\text{Vacuum produced steam inlet}}{\text{Barometer pressure} - \text{Absolute pressure}}$$

Absolute pressure of steam at 35°C = 0.0559 bar

$$= \frac{76}{1.013} \times 0.0559$$

$$= 4.19 \text{ cm of mercury}$$

$$\eta_v = \frac{70.1}{76.3 - 4.19} = \mathbf{97.2\%}$$

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Example 11.3

In a condenser test, the following observations were made

Vacuum = 720 mm of mercury

Barometer = 765 mm of mercury

Mean temperature of condensation = 34°C

Hot well temperature = 29°C

Inlet temperature of cooling water = 15°C

Outlet temperature of cooling water = 25°C

Determine the following:

(1) Vacuum corrected to standard barometer of 760 mm
(2) Under-cooling efficiency
(3) Under-cooling condensate
(4) Condenser efficiency

Solution:

(1) Vacuum corrected to 760 mm barometer

$$= 760 - (\text{barometer height} - \text{vacuum})$$
$$= 760 - (765 - 720)$$
$$= 715 \text{ mm of Hg}$$

(2) Under-cooling efficiency

$$\eta_v = \frac{\text{Actual vacuum}}{\text{Barometer pressure} - \text{Absolute of steam}}$$

From steam tables, absolute pressure of steam at 34°C is 0.0533 bar

$$\eta_v = \frac{760}{1.013} \times 0.0533 = 39.99 \text{ mm of Hg}$$

$$\eta_v = \frac{720}{765 - 39.99} = 99.31\%$$

(3) Under-cooling of condenser

$$= \text{condenser temperature} - \text{hot well temperature}$$
$$= 34 - 29 = 5°C$$

(4) Condenser efficiency

$$\text{Condenser efficiency} = \frac{\text{Actual temperature rise of cooling water}}{\text{maximum possible temperature rise of cooling water}}$$

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Now condenser pressure = 765 − 720 = 45 mm of Hg (absolute)

$$= \frac{1.013}{760} \times 45 = 0.06 \text{ bar}$$

From tables saturation temperature of steam at 0.06 bar = 38°C

$$\text{Condenser efficiency} = \frac{25-15}{38-15} = 43.48\%$$

Example 11.4

In a condenser vacuum is 720 mm of Hg when the barometer reads 760 mm. Mean temperature of condensate is 32°C. Exhaust steam that comes to the condenser is 45,000 kg/h, and its dryness fraction is 0.9. Determine the quantity of cooling water required if the temperature rise of cooling water is 12°C.

Solution:

Condenser pressure = 760 − 720 = 40 mm of Hg

= 0.0533 bar

Total heat in exhaust steam = enthalpy at condenser inlet + dryness fraction × latent heat

$$= 129.8 + 0.9 \times 2421$$

$$= 2308.7 \text{ kJ/kg}$$

Sensible heat at 32°C = 134 kJ/kg

Mass of cooling water,

$$m\text{W} = \frac{m_s \left(xL + h_{fw} - h_{fc}\right)}{C_{pw}\left(T_2 - T_1\right)}$$

$$= \frac{45{,}000\left(2308.7 - 134\right)}{4.187 \times 12 \times 60}$$

$$= 32462.08 \text{ kg/min}$$

11.4 COOLING TOWER

In large power plants, cooling towers are used in place of cooling ponds. A cooling tower is a wooden or metallic rectangular structure, with packed baffling devices. The hot water is delivered to the top of tower and falls down through the tower and is broken into small particles while passing over the baffling devices. Air enters the towers at the bottom and flows upwards and cools the water. The hot air leaves the tower at the top. The cooled water falls down into a tank below the tower from where it can again be circulated to the condenser.

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11.4.1 Types

In modern power plants, condensate water is cooled by using a cooling tower. Cooling towers are basically classified as *wet cooling tower* and *dry cooling tower*. Further, they may either be forced draught (mechanical draft) or natural draft cooling towers depending on the mode of cooling.

11.4.1.1 Wet Cooling Towers

In wet cooling towers, water and air come in direct contact. Heat in water is dissipated by (i) the addition of sensible heat to air, (ii) evaporation of a portion of re-circulated water and (iii) addition of sensible heat to the water due to terminal temperature difference.

Figure 11.10 shows a wet cooling tower with natural draft. Hot water from the plant enters the distributor system where water is sprayed over horizontally set *fill* or *packing*. Air and water are thoroughly mixed by fill as water splashes down from one fill level to the next due to gravity. Outside air enters the tower through *louvers* from the sides of the tower. The heat and mass transfer (evaporation) between water and air is enhanced due to intimate mixing. Water gets cooled and is collected in a concrete basin at the bottom of the tower. Hot but moist air leaves the tower from the top. The cold water is re-circulated using a pump.

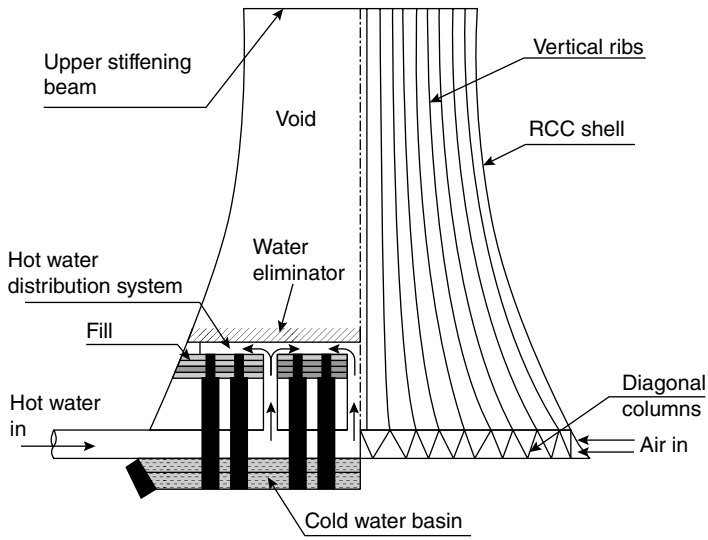


Fig. 11.10 *Natural Draft Counter-Flow Cooling Tower*

11.4.1.2 Dry Cooling Towers

In dry cooling tower, circulating water is passed through finned tubes over which cooling air is passed. Heat is rejected to the air in the form of sensible heat. A dry cooling tower may either be natural draft type or forced draft type. These types are cheaper than wet types of cooling towers. Figure 11.11 shows a direct, dry cooling tower.

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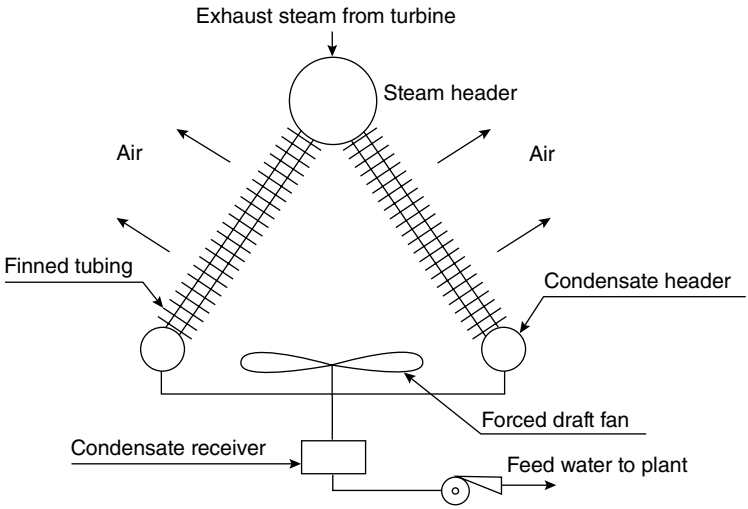


Fig. 11.11 *Direct Dry Cooling Tower*

Exhaust steam from turbine is admitted to a steam header (to minimize pressure drop). Steam gets condensed as it passes down through finned tubes arranged in parallel rows. Heat transfer is effected either by natural circulation of air or by forced circulation of air using a forced draft fan. The condensate collected in the receiver is re-circulated.

11.4.2 Principle of Operation and Performance

According to the nature of air draught, the cooling towers are classified as follows:

(i) Atmospheric cooling tower
(ii) Natural draught cooling tower
(iii) Mechanical draught cooling tower

Based on the type of draught, mechanical draught cooling tower can be further classified as follows:

(a) Induced draught cooling tower
(b) Forced draught cooling tower
(c) Combined induced and forced draught cooling towers (hybrid cooling tower)

The working principle of all the above types of cooling towers is described below.

(i) *Atmospheric cooling tower*

Figure 11.12 shows an atmospheric cooling tower. The hot water is delivered at the topmost tray, and it falls down from one tray to another until it reaches the tank below the tower. The water is cooled by air flowing across the tower. To increase the rate of cooling, water is sprayed through nozzles from the top. The number of decks of trays depends on the load of the plant. This type of cooling tower is used only for small capacity power plants.

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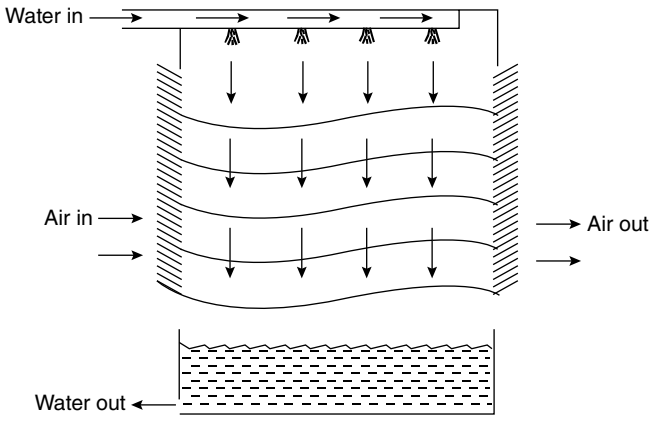


Fig. 11.12 *Atmospheric Cooling Tower*

(ii) *Natural draught cooling towers*

Natural draft towers are typically about 120 m high, depending on the differential pressure between the cold outside air and the hot humid air on the inside of the tower as the driving force. These towers do not use fans for draught generation. Depending on climatic and operating requirement conditions, selection on whether the natural or mechanical draft towers is done. Figure 11.13 shows a natural draught cooling tower.

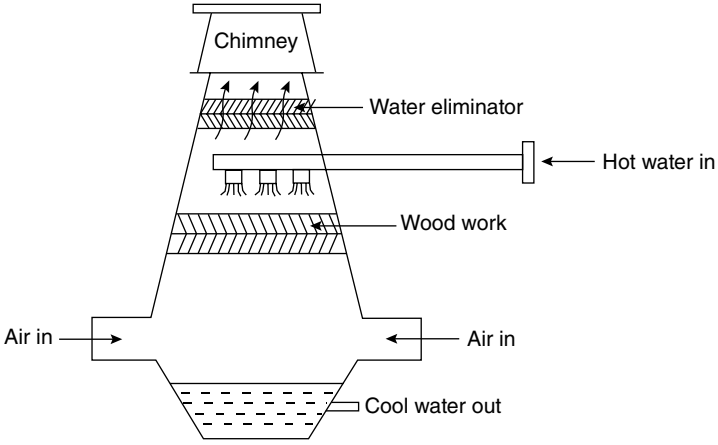


Fig. 11.13 *Natural Draught Cooling Tower*

In natural draught cooling towers, the flow of air occurs due the natural pressure head caused by density difference between the cold outside air and hot humid air inside.

In this tower, wooden hurdles and distributing trays are provided for spreading the water and for breaking it into small particles. The hot water from the condenser is pumped to a height of

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9–15 m and enters the tower and then gets distributed over the wood work and trays. The hot water then falls down, and the steam vapours that are lighter than air will rise upwards. This will create natural draught and air will enter from the bottom of the tower that is open to atmosphere. The rising air meets the falling spray of hot water and cools it. The hot air along with some vapour will leave the tower at the top and cold water will fall down into the pond below the tower and get circulated through the condenser again. Water eliminator prevents the escape of water particles by the air leaving the tower. The disadvantage of natural draught cooling tower is that to produce large draught, the tower should by very high.

11.4.2.1 Hyperbolic Natural Draft Cooling Tower

Hyperbolic natural draft cooling towers work best when the difference between cold water and WBT of air ($T_{wbt} - T_{wi}$) is equal to or is greater than the difference between hot water and cold water temperature, that is, when approach is equal to or greater than the cooling range. Therefore, they are preferred when operating conditions have low WBT and high relative humidity. Figure 11.14 shows a typical hyperbolic cooling tower.

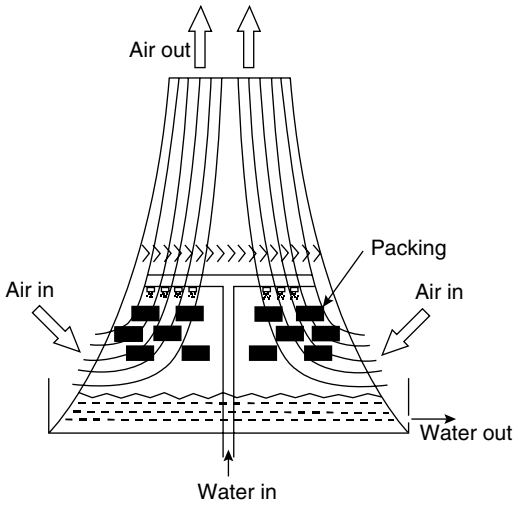


Fig. 11.14 *Hyperbolic Cooling Tower*

Although initial cost may be higher, the saving in fan power, longer life and less maintenance are favourable features of this tower.

Advantages

The advantages of hyperbolic tower over mechanical towers are listed below:

(i) The hyperbolic towers have a cooling capacity comparable with that of induced draft cooling towers, and they also require considerably less ground area.

(ii) Since no fans are needed, power cost and auxiliary equipments are eliminated; therefore, operating and maintenance costs at consequently reduced. It gives more or less trouble-free operation.

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(iii) Hyperbolic tower's chimney shape creates its own draft and ensures efficient operation even when there is no wind.

(iv) Ground fogging and warm air recirculation that are often the most common problems faced with mechanical draft installations are also avoided in hyperbolic towers.

(v) The towers may be as high as 125 m and 100 m in diameter at the base with the capability of withstanding winds of well over 100 mph. These are more or less self-supported structures.

(vi) The enlarged top of the tower allows water to fall out of suspension.

Drawbacks

The major drawbacks of this tower are listed below:

(i) Its cost is considerably high.

(ii) Its performance varies with the seasonal changes in DBT and RH of air.

The hyperbolic towers are almost often selected over the mechanical draft towers under the following operating conditions:

(a) A combination of low WBT and high inlet and outlet water temperature exists, that is, broad cooling range and long cooling approach.

(b) Heat load is heavy during winter.

(c) It is also more favourable over mechanical draft as central station size increases.

(iii) *Mechanical draught cooling tower*

In these towers, the draught of air is produced mechanically by means of propeller fans. They give higher efficiency, reduce spray and windage losses and require less ground area. Mechanical draft towers use fans (one or more) to move large quantities of air through the tower. The air flow in these towers may be either cross flow or counter flow with respect to the falling water. In cross flow towers, airflow is horizontal in the filled portion of the tower; whereas in counter flow, the air flow is in the opposite direction of the falling water.

The counter-flow tower requires less floor area than a cross flow tower but is taller for a given capacity. The cross flow towers have low-pressure drop in relation to their capacity and consume lower fan power, and hence have lower energy costs.

All mechanical towers must be properly located in order to ensure that the air diffuses freely without recirculation through the tower, and that the air intakes are not restricted. Cooling towers should be located as near as possible to the refrigeration systems they serve. If they are located below the refrigeration system, then the condenser water drains out of the system through the tower basin when the system is shut down.

11.4.2.2 Forced Draft Cooling Towers

The forced draft tower has the fan, basin and piping located within the tower structure. In this model, the fan is located at the base. There are no louvered exterior walls. Instead, the structural steel or wood framing is covered with panelling made of aluminium, galvanized steel or asbestos cement boards (see Figure 11.15).

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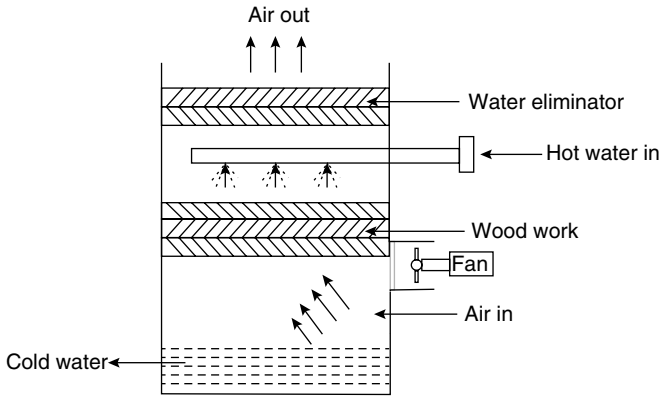


Fig. 11.15 *Mechanical Draught Cooling Tower*

In forced draught cooling, tower fan is located at the base of the tower and the air is blown by the fan up through the descending water. The entrained water is removed by water eliminator at the top.

11.4.2.3 Induced Draft Cooling Towers

In induced draught cooling towers, the fan is located at the top of the tower and draws the air in through the trays and discharges at the top through fan casing (see Figure 11.16).

Induced draught is considered to be better than forced draught because the power requirement is high for forced draught and the maintenance of fan is costlier. The induced draught occupies less space as the fan drives are placed at the top of the tower; the cooling effect is distributed across the entire cross section of the tower. Fans handle warm air, hence they are non-freezing. Since air leaves at a high speed, it is non-circulating.

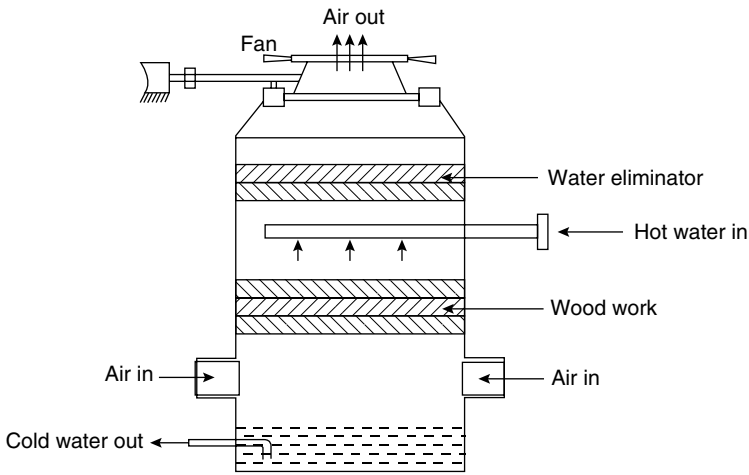


Fig. 11.16 *Induced Draught Cooling Tower*

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11.4.2.4 Hybrid Draft Cooling Towers

These towers utilize mechanical draft fans to augment airflow and are also referred to as fan-assisted natural draft towers. This minimizes the power required for the air movement, with least-possible stack cost impact. Properly designed fans may need to be operated only during periods of high ambient and peak loads (see Figure 11.17).

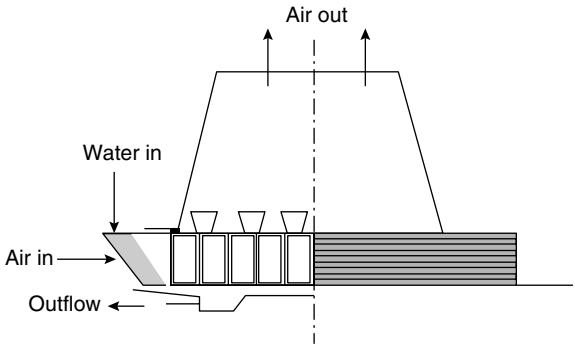


Fig. 11.17 *Hybrid Cooling Tower*

11.4.3 Location of Cooling Tower

The following points should be considered when locating the cooling tower (see Figure 11.18):

(i) The tower site should be such that it allows unrestricted air flow to the tower. Open space as far as possible should be allowed between the cooling tower louvers and nearby structures.

(ii) It is always advisable to place a cooling tower away from office building, laboratory and control room in the plant because of the possibility of the moist air and noise of the tower causing inconvenience to the working people.

(iii) Prevailing wind direction should be studied so as to minimize their circulation effect of hot air from the tower, which should be carried away from the tower and should not re-enter through the louvers.

(iv) The tower should be located in such a way that the piping running to and from it is minimum. The tower should be as near as possible to the source of makeup water.

(v) The placement of the tower should also be studied with reference to nearby chimney and effluent from processes, particularly if these are hot or oily.

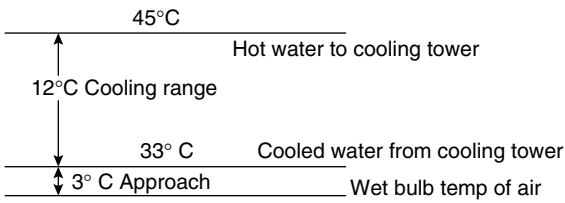


Fig. 11.18 *Location of Cooling Tower with Desired Temperature Distribution*

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11.5 COOLING PONDS

The cooling water system is one of the most important systems of power plant, and the availability of cooling water decides the plant site. Cooling system may be classified as follows (see Figure 11.19):

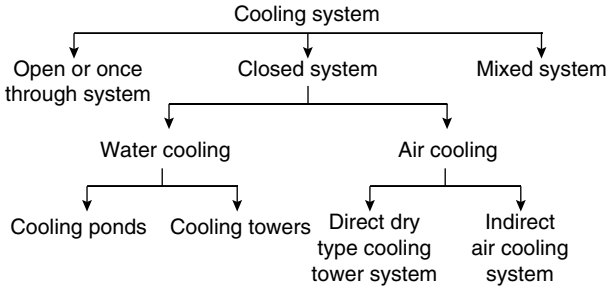


Fig. 11.19 *Classification of Cooling Ponds*

In open or once-through system, the water is drawn directly from the upstream side of river pumped through the condenser and is discharged to the downstream side of the river at temperature 5–10°C in excess of inlet temperature. The temperature of the discharged water should be kept within safe limits to prevent damage to fishes. In closed system, the hot water that comes out of condenser is cooled either by spraying in the pond or by passing through the cooling tower. In mixed system, river water and cooling towers are used simultaneously.

11.5.1 Cooling Pond

Spray pond or cooling pond works on the principle of evaporation of water sprayed through nozzles into the air above its surface. It consists of modules having electrically driven propeller-type pumps that distribute water through a diffuser system to spray nozzles. The modules are placed properly within the pond, and they may be fixed or floating. Each module consists of four spray nozzles. The surface area of contact of water is increased when it is sprayed in air. Cooling occurs due to the evaporation of water vapours as the spray propels upwards and falls down to the surface. Heat transfer is mainly by evaporation, conduction to air and partly by radiation. During warm weather, the rate of evaporation is rapid (see Figure 11.20).

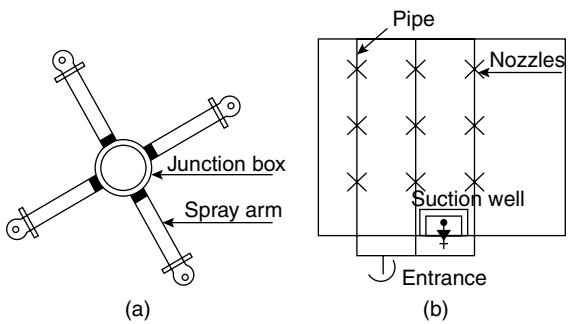


Fig. 11.20 *Layout of Spray Pond (a) Spray Module and (b) Spray Pond*

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Spray cooling pond is one of the simplest methods of cooling the condenser water. The hot water coming out of the condenser is sprayed through nozzles to expose maximum surface area of water to air for effective cooling. The spray ponds are surrounded with wooden walls to prevent the wind from carrying the water particles. Figure 11.21 shows a spray pond.

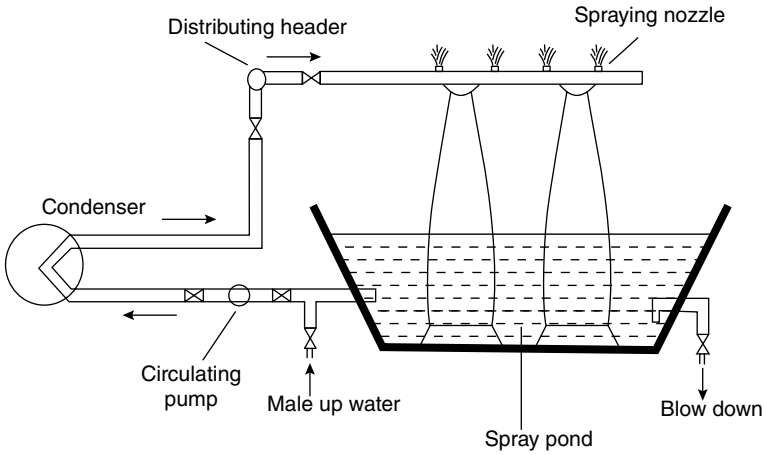


Fig. 11.21 *Spray Pond*

11.5.2 Types of Cooling Ponds

The common types of cooling ponds used in practice are explained below:

11.5.2.1 Single-Deck and Double-Deck Systems

In a single-deck system, spray nozzles are arranged at the same elevation as shown in Figure 11.22. Its effective cooling effect is less than double-deck system.

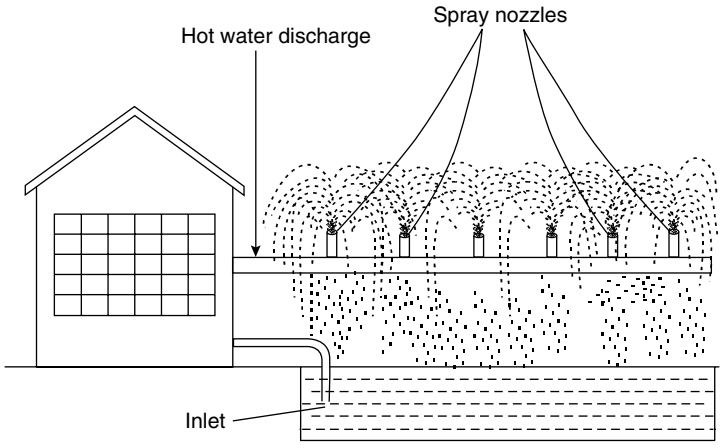


Fig. 11.22 *Single-Deck Spray Pond*

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11.5.2.2 Double-Deck Systems

In this system, spray nozzles are arranged at different elevations as shown in Figure 11.23. Its cooling effect is more than single-deck system as water comes in contact with air at lower temperature.

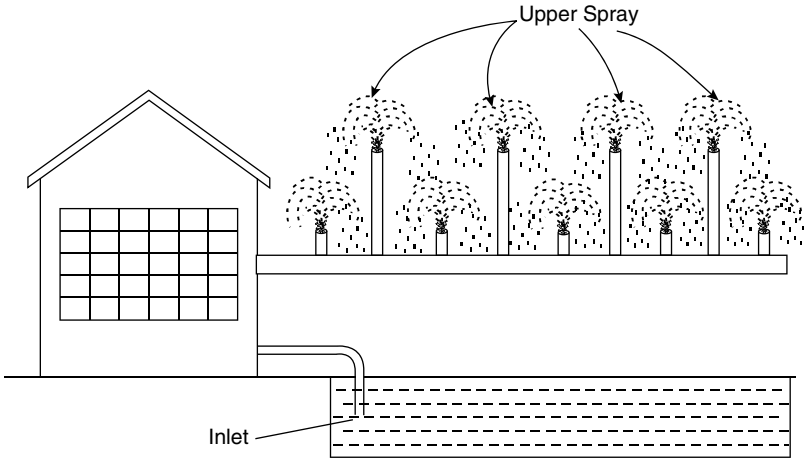


Fig. 11.23 *Double-Deck Spray Pond*

11.5.2.3 Natural and Directed Flow System

In natural flow system, water comes out from the condenser is just allowed to flow into the pond. This system is rarely used nowadays.

In directed flow system, the hot water that comes out of the condenser enters the middle channel. On reaching the far end, it is divided into two currents being directed by the baffle walls so as to traverse the pond several times before uniting at the intake point. The water gets more time and passes over a more surface. The cooling achieved is very effective.

11.5.2.4 Open and Louvre Fence

In an open pond, the drift losses will be more if the wind velocity is high. This can be avoided by providing louvre.

A few disadvantages of cooling ponds are listed below:

(i) The area required of cooling is considerably large.

(ii) Spray losses due to evaporation and windage run high.

(iii) There is no control over the temperature of cooled water. The cooling effect is reduced with the decrease in wind velocity; and if the load on the plant increases, the pond does not respond to the requirement. When the maximum cooling is required during increased load, it provides minimum cooling in the absence of wind flow.

(iv) The cooling efficiency is low compared with cooling tower. Spray cooling pond system is rarely used nowadays in modern thermal power plants of very high capacity as it requires large area that is rarely available.

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11.6 CORROSION IN CONDENSERS AND BOILERS

Boilers and condensers come in constant contact with water, and they are susceptible to corrosion. Boilers and condensers corrode due to the following reasons:

High temperatures that dissolve the oxygen in their environment.

Chemicals used in boiler rooms and cyclic moisture stresses involving boiling and condensation.

Interaction between carbon dioxide and oxygen can act as a contaminating agent and cause corrosion.

Copper alloys in the boilers and condensers can be attacked and degraded by the sulphur and ammonia-based vapours that are present in boiler rooms.

One of the ways in which corrosion damage in boilers and condensers can be prevented is to use thermal coating. Thermal coating in the form of a thermal spray helps to preserve the life of boiler tubes, which corrode very easily. The thermal spray will act as a barrier between the corrosive elements and the metal of the tubes. Spraying inside the boilers will also prevent the need for expensive tube replacement and welding.

11.7 CORROSION IN BOILERS

Corrosion is the reversion of a metal to its ore form. The process of corrosion is a complex electrochemical reaction and it takes many forms. Corrosion may either produce surface attachment over a large metal surface or it may result in pinpoint penetration of metal. Corrosion can be of widely varying origin and nature due to the action of dissolved oxygen, to corrosion currents set up as a result of heterogeneities on metal surfaces, or to the iron being directly attacked by the water. While basic corrosion in boilers may be primarily due to reaction of the metal with oxygen, other factors such as stresses, acid conditions, and specific chemicals may also contribute to significant influence and produce different forms of attack. It is necessary to consider the quantity of the various harmful substances that can be allowed in the boiler water without risk of damage to the boiler. Corrosion may occur in the feedwater system as a result of low pH water and the presence of dissolved oxygen and carbon dioxide. Few other types of corrosion found in boilers may be due to acidic, galvanic corrosion apart from oxygen and carbon dioxide attack.

11.7.1 Acidic Corrosion

Acidic corrosion in boilers may occur due to the following reasons:

(i) Low makeup feedwater pH can cause serious acid attack on metal surfaces in the preboiler and boiler system. Feed water can also become acidic from contamination of the system, namely process contamination of condensate or cooling water contamination from condensers.

(ii) Acidic corrosion can also be caused by chemical cleaning operations such as overheating of the cleaning solution, excessive exposure of metal to cleaning agent and high cleaning agent concentration.

(iii) In the boiler and feedwater system, acidic attack can take the form of general thinning, or it can be localized at areas of high stress.

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11.7.2 Galvanic Corrosion

Concentration of caustic (NaOH) can occur as a result of steam blanketing that allows salts to concentrate on boiler metal surface or by localized boiling beneath porous deposits on tube surface. Caustic corrosion occurs when caustic is concentrated and dissolves the protective magnetite (Fe_3O_4) layer, causing a loss of base metal and eventual failure due to cracking.

The following conditions favour the cracking of metal surface:

1. Excessive stressing of metal
2. Presence of caustic in the boiler water
3. Presence of traces of silica in the boiler water
4. Due to leakage of the boiler water on the stressed metal

11.7.3 Steam Blanketing

Steam blanketing is a condition wherein a steam layer is formed between the boiler water and the tube wall. Under this condition, insufficient water reaches the tube surface for efficient heat transfer. The water that reaches the overheated boiler wall is rapidly vaporized, leaving behind a concentrated corrosive caustic solution.

Boiler feedwater systems using de-mineralized or evaporated makeup or pure condensate may be protected from caustic attack through coordinated phosphate/pH control. Phosphate buffers the boiler water, reducing the chance of large pH changes, and hence the development of high caustic concentrations. Excess caustic combines with disodium phosphate and forms tri-sodium phosphate, by the following reaction:

$$Na_2HPO_4 + NaOH \rightarrow Na_3PO_4 + H_2O$$

This results in the prevention of caustic build up beneath deposits or within a crevice where leakage is occurring.

11.7.4 Oxygen Attack

When there is no proper mechanical and chemical deaeration, oxygen in the feedwater enters the boiler. Majority of oxygen is flashed off with the steam in the boiler, and the remainder of it can attack the boiler metal. Oxygen in water produces localized pitting. In the presence of oxygen, water if it contains ammonia, readily attacks copper and copper-bearing alloys, thereby corroding them. This leads to deposits on boiler heat transfer surfaces and reduces efficiency and reliability. Oxygen is highly corrosive in hot water. Even small concentrations can cause serious problems such as the formation of iron oxide producing iron deposits in the boiler. Oxygen corrosion may be highly localized or may cover an extensive area.

Oxygen attack is an electrochemical process that happens due to the following reactions:

$$\text{Anode: } Fe \rightarrow Fe^{2}+ + 2e^{-}$$

$$\text{Cathode: } \tfrac{1}{2}\, O_2 + H_2O + 2e^{-} \rightarrow 2\, OH^{-}$$

$$\text{Overall: } Fe + \tfrac{1}{2}\, O_2 + H_2O \rightarrow Fe(OH)_2$$

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The above reaction is endothermic resulting in temperature rise. This provides enough additional energy to accelerate reactions at the metal surfaces, resulting in rapid and severe corrosion. In a boiler-condenser system, the acceptable dissolved oxygen level depends on factors, such as feed water temperature, pH value, flow rate, dissolved solids content, and the metallurgy and physical condition of the system. In general, the limiting value of oxygen in makeup water is restricted to 0.10 mg/kg. Corrosion due to oxygen can be avoided using a chemical scavenger after mechanical deaeration. Membrane contractors can also be used to inhibit oxygen corrosion.

11.7.5 Carbon Dioxide Attack

Carbon dioxide exists in aqueous solutions as free carbon dioxide and in the combined forms of carbonate and bicarbonate ions. Dissolved carbon dioxide supports corrosion. The CO_2 gas dissolves in water, producing corrosive carbonic acid according to the following equation

$$H_2O + CO_2 \rightarrow H_2CO_3 \rightarrow H^+ + HCO^{3-}$$

This results in lower value of pH and also enhances the corrosive effect of oxygen.

In boiler systems, corrosion resulting from carbon dioxide is most commonly observed in the condensate system. This is because feedwater deaeration normally removes carbon dioxide from the boiler feedwater. The presence of the gas in condensate is typically due to carbonate and bicarbonate decomposition under boiler conditions. According to an estimate, feedwater with a total alkalinity of 100 mg/l as calcium carbonate could generate a carbon dioxide level of 79 mg/l* in the steam. Carbon dioxide of such a high concentration is corrosive if present in the condensate.

According to the type of corrosion encountered, many corrosion control techniques are used. The following are the major methods involved:

(i) Maintenance of the proper pH
(ii) Control of oxygen
(iii) Control of deposits
(iv) Reduction of stresses trough design and operational practices.
(v) Deaeration and use of membrane contractors are the best and most diffused ways of avoiding corrosion by removing the dissolved gasses (mainly O_2 and CO_2).

11.8 QUESTIONS

11.8.1 Objective Questions

1. The thermal efficiency of the engine with condenser as compared to without condenser, for the same given pressure and temperature of steam is
   (a) higher
   (b) lower
   (c) same as long as initial pressure and temperature is unchanged
   (d) none of the above

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2. In jet-type steam condensers,
   (a) cooling water passes through tubes and steam surrounds them
   (b) steam passes through tubes and cooling water surrounds them
   (c) steam and cooling water mix
   (d) steam and cooling water do not mix
3. In a shell and tube surface condenser,
   (a) steam and cooling water mix to give the condensate
   (b) cooling water passes through the tubes and steam surrounds them
   (c) steam passes through the cooling tubes and cooling water surrounds them
   (d) all of the above varying with situation
4. Air removal in a surface condenser if air is removed, there is
   (a) fall in absolute pressure maintained in condenser
   (b) rise in absolute pressure maintained in condenser
   (c) no change in absolute pressure in the condenser
   (d) rise in temperature of condensed steam
5. The cooling section in the surface condenser
   (a) increases the quantity of vapour extracted along with air
   (b) reduces the quantity of vapour extracted along with air
   (c) does not affect vapour quantity extracted but reduces pump capacity of air extraction pump
   (d) all of the above
6. Edward's air pump
   (a) removes air and also vapour from condenser
   (b) removes only air from condenser
   (c) removes only un-condensed vapour from condenser
   (d) removes air along with vapour and also the condensed water from condenser
7. In a steam power plant, the function of a condenser is
   (a) to maintain pressure below atmospheric to increase work output from the prime mover
   (b) to receive large volumes of steam exhausted from steam prime mover
   (c) to condense large volumes of steam to water that may be used again in boiler
   (d) all of the above
8. A regenerative type surface condenser uses
   (a) one pump to remove air and condensate
   (b) two pumps to remove air and condensate
   (c) three pumps to remove air, vapour and condensate
   (d) there is no pump, the condensate gets removed by gravity
9. In a evaporative type of condenser
   (a) steam flows in pipes surrounded by water
   (b) water flows in pipes surrounded by steam
   (c) either (a) or (b)
   (d) none of the above

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Answers

1. a 2. c 3. b 4. a 5. b 6. d 7. d 8. b 9. a

11.8.2 Review Questions

1. With a neat sketch, explain the operation of the following condensers:
   (a) jet condenser
   (b) surface condenser
   (c) evaporative condenser
2. Discuss the advantages and disadvantages of surface condensers over jet condensers.
3. Classify the condensers and explain the essential difference in each type.
4. Define 'vacuum efficiency' applied to a condenser.
5. What are the factors that affect the vacuum efficiency of a condenser?
6. What are the sources of air leakage into a condenser? Briefly state the effects of air leakage on the performance of a condenser.
7. With a neat sketch, explain hyperbolic cooling towers and discuss its merits and demerits.
8. What is the mechanism of evaporative cooling?
9. Explain the working of forced draft and 'induced draft' cooling towers with neat sketches.
10. What is necessity of cooling the surface condenser waters when a closed system is used?
11. Explain the different methods of water distribution system in cooling towers with neat sketches. Discuss its merits and demerits.
12. What is meant by mixed-type cooling system?
13. What is the difference between indirect-type dry cooling system and direct-type dry cooling system? Explain with neat sketches.
14. Why in a modern power plant air cooled cooling systems are preferred than water cooled cooling system?
15. The following reading were taken during a test on a condenser:
    - Vacuum in the condenser = 700 mm of Hg
    - Barometer reading = 762 mm of Hg
    - Temperature of steam entering into the condenser = 35°C
    - Cooling water circulated = 46,270 kg/hr
    - Inlet and outlet temperature of cooling water = 16.7°C and 31°C
    - Condensate collected = 1,120 kg/hr

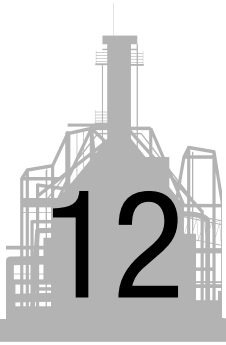
    Find out the following:
    (a) mass of air present per cubic metre of condenser volume
    (b) the condition of steam entering in to the condenser
    (c) vacuum efficiency of the condenser

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16. Discuss the function of cooling ponds and cooling towers in thermal power stations.
17. With a neat sketch, explain ‘once through cooling water system’.
18. What are the drawbacks of cooling ponds?
19. Differentiate dry and wet cooling towers.
20. Explain the necessity of circulating cooling water system.

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Gas Turbine Power Plant



12

Contents

12.1 Introduction
12.2 Classification of gas turbines
12.3 Construction and plant layout with auxiliaries
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12.9 Combined cycle power plants and comparison
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12.1 INTRODUCTION

Gas turbine is a rotary machine that uses gas as a working fluid. The basic turbine consists of a compressor, a combustor or heater and a turbine or expander as shown in the Figure 12.1. The compressor is supplied with air at ambient conditions and raises its pressure and temperature to the required limit for combustion. The compressed air enters the combustor or combustion

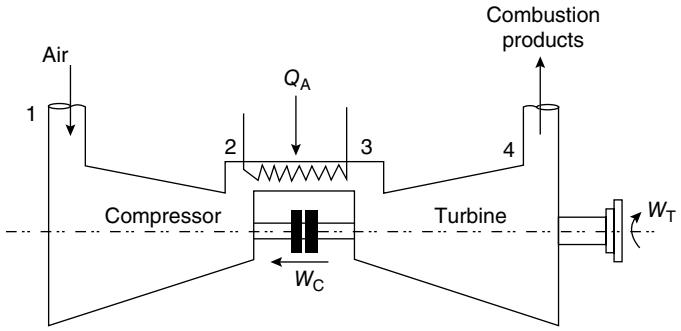


Fig. 12.1 *Gas Turbine: Working Principle*

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chamber where it gets heated from the fuel burning in the combustion chamber. The combustion products (gas and air) are then expanded in the turbine, thereby doing mechanical work. The gases are then exhausted to the atmosphere, as in an open-cycle gas turbine.

12.1.1 Joule or Brayton Cycle

Simple gas turbines can be analysed using Joule or Brayton cycle. Some assumptions are made before making any attempt to analyse an ideal cycle. These assumptions are as follows:

(i) The effect of friction upon the machine and the fluid is neglected.

(ii) The change in potential energy and kinetic energy of the medium as a result of overall process is neglected. Thus, pressure losses in the heat exchangers and the connecting passages are absent.

(iii) There is no increase in the mass flow rate due to any addition of fuel.

(iv) The working fluid is a perfect gas. The specific heat remains constant at all temperatures for turbines and compressors.

(v) Losses due to radiation are neglected.

Analysis of the cycle is carried out with the help of the above assumptions. The simple cycle with actual and ideal processes is shown in the *p-V* and *T-s* diagrams in Figure 12.2.

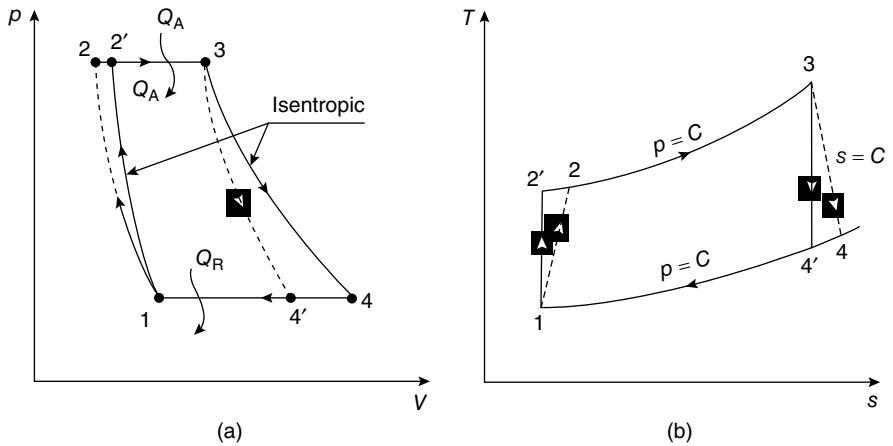


Fig. 12.2 *Joule Cycle: Ideal and Actual Processes*

12.1.1.1 Ideal Cycle Analysis

(i) 1–2′: Isentropic compression

Temperature increases from T_1 to $T_{2'}$

Pressure increases from p_1 to p_2.

Volume decreases from V_1 to $V_{2'}$.

Work done during the process,

$$W_C = C_p (T_{2'} - T_1) = h_{2'} - h_1$$

Heat transfer is zero.

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(ii) 2′–3: Constant pressure heat addition
Temperature increases from $T_{2'}$ to T_3.
Pressure remains constant at $p_{2'} = p_3$.
Volume increases from $V_{2'}$ to V_3.
Work done during the process

$$W_{2'-3} = p\,(V_3 - V_{2'})$$

Heat transfer during the process,

$$Q_A = C_p\,(T_3 - T_{2'}) = h_3 - h_{2'}$$

(iii) 3–4′: Isentropic expansion
Temperature decreases from T_3 to T_4.
Pressure decreases from $p_3 = p_2$ to $p_{4'} = p_1$.
Volume increases from V_3 to $V_{4'}$.
Work done during the process,

$$W_T = C_p\,(T_3 - T_{4'}) = h_3 - h_{4'}$$

Heat transfer is zero.

(iv) 4′–1: Constant pressure heat rejection
Temperature decreases from $T_{4'}$ to T_1.
Pressure remains constant at $p_1 = p_4$
Volume decreases from $V_{4'}$ to V_1.
Work done during the process,

$$W_{4-1} = p(V_{4'} - V_1)$$

Heat transfer during the process,

$$Q_R = C_p\,(T_{4'} - T_1) = h_{4'} - h_1$$

1. *Net work done/cycle*

Net work done by the cycle is the difference of the turbine work and the compressor work.

$$\text{Net work done} = \text{turbine work} - \text{compressor work}$$

$$\oint W = \text{heat added} - \text{heat rejected}$$

$$= Q_A - Q_R$$

$$= C_p\,(T_3 - T_{2'}) - C_p\,(T_{4'} - T_1)$$

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$$= C_p (T_3 - T_{4'}) - C_p (T_{2'} - T_1) \tag{1}$$

$$= (h_3 - h_{4'}) - (h_{2'} - h_1) \tag{2}$$

where, h = enthalpy of the working fluid, kJ/kg

2. *Work ratio*

Work ratio is defined as the ratio of net work done to the turbine work (positive work).

$$W_R = \frac{\oint W}{W_T} = \frac{\text{Net work done}}{\text{Turbine work}}$$

$$= \frac{(h_3 - h_{4'}) + (h_{2'} - h_1)}{(h_3 - h_{4'})}$$

$$= \frac{C_p (T_3 - T_{4'}) + C_p (T_{2'} - T_1)}{C_p (T_3 - T_{4'})}$$

$$W_R = 1 - \frac{T_1 \left(\frac{T_{2'}}{T_1} - 1 \right)}{T_3 \left(1 - \frac{T_{4'}}{T_3} \right)} \tag{3}$$

Again, considering the isentropic processes 1–2′ and 3–4′:

$$W_R = 1 - \frac{T_1 \left(\frac{T_{2'}}{T_1} - 1 \right)}{T_3 \left(1 - \frac{T_{4'}}{T_3} \right)}$$

Let

$$\left(\frac{p_2}{p_1} \right)^{\frac{(\gamma - 1)}{\gamma}} = \frac{T_{2'}}{T_1} = \frac{T_3}{T_{4'}} = R, \text{ say} \tag{4}$$

then,

$$W_R = 1 - \frac{T_1 (R - 1)}{T_3 \left(1 - \frac{1}{R} \right)}$$

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$$W_R = 1 - \frac{T_1}{T_3} R \tag{5}$$

3. *Thermal efficiency*

Thermal efficiency of the cycle is given by the ratio of net work done to the heat supplied.

$$\eta = \frac{\text{Net work done}}{\text{Heat added}}$$

$$= \frac{W_T - W_C}{Q_A}$$

$$= \frac{C_p (T_3 - T_{4'}) + C_p (T_{2'} - T_1)}{C_p (T_3 - T_{2'})}$$

$$= \frac{(T_3 - T_{2'}) + (T_{4'} - T_1)}{(T_3 - T_{2'})}$$

$$= 1 - \left(\frac{T_{4'} - T_1}{T_3 - T_{2'}} \right) \tag{6}$$

From the isentropic processes, 1–2′ and 3–4′,

$$R = \frac{T_{2'}}{T_1} = \frac{T_3}{T_{4'}} = \left(\frac{p_2}{p_1} \right)^{\frac{(\gamma-1)}{\gamma}} = \left(\frac{p_3}{p_4} \right)^{\frac{(\gamma-1)}{\gamma}} = (\beta)^{\frac{(\gamma-1)}{\gamma}}$$

where β = pressure ratio = $\frac{p_2}{p_1} = \frac{p_3}{p_4}$

Equation (6) can be modified as follows:

$$\eta = 1 - \frac{T_{4'} - T_1}{\left(\frac{T_3}{T_{2'}} - 1 \right) T_{2'}} = 1 - \frac{T_{4'} - T_1}{\left(\frac{T_{4'}}{T_1} - 1 \right) T_{2'}} = 1 - \frac{T_1}{T_{2'}}$$

$$\eta = 1 - \frac{1}{R} \tag{7}$$

$$= 1 - \frac{1}{(\beta)^{\frac{(\gamma-1)}{\gamma}}} \tag{8}$$

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12.1.1.2 Actual Cycle Analysis

In the actual cycle, expansion and compression in the turbine are not isentropic. This means that compressor and turbine efficiencies are to be considered for the analysis of the cycle as shown in Figure 12.3.

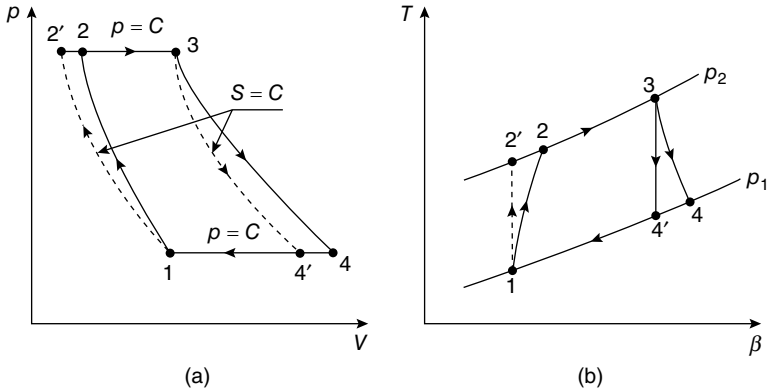


Fig. 12.3 *Joule Cycle: Actual Cycle Analysis*

Considering processes 1 and 2,
Compressor efficiency,

$$\eta_C = \frac{\text{Isentropic work}}{\text{Actual work}}$$

$$\eta_C = \frac{h_{2'} - h_1}{h_2 - h_1} \tag{9}$$

$$\eta_C = \frac{T_{2'} - T_1}{T_2 - T_1} \tag{10}$$

$$\therefore \quad (T_{2'} - T_1) = \eta_C (T_2 - T_1) \tag{11}$$

Considering processes 3 and 4,
Turbine efficiency,

$$\eta_t = \frac{\text{Actual work}}{\text{Isentropic work}}$$

$$\eta_t = \frac{h_3 - h_1}{h_3 - h_{4'}} \tag{12}$$

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$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4'}} \tag{13}$$

$$\therefore \quad (T_3 - T_{4'}) = \frac{T_3 - T_4}{\eta_t} \tag{14}$$

From Equation (11),

$$(T_2 - T_1) = \frac{1}{\eta_C}(T_{2'} - T_1)$$

$$T_2 = T_1 + \frac{1}{\eta_C}(T_{2'} - T_1)$$

$$\frac{T_2}{T_1} = 1 + \frac{1}{\eta_C}\left(\frac{T_{2'}}{T_1} - 1\right)$$

$$\frac{T_2}{T_1} = 1 + \frac{1}{\eta_C}(R - 1) \tag{15}$$

From Equation (14),

$$T_4 = T_3 - \eta_t (T_3 - T_{4'})$$

$$\frac{T_4}{T_3} = 1 - \eta_t\left(1 - \frac{1}{R}\right) \tag{16}$$

1. *Net work done/cycle*

$$\oint W = W_T - W_C$$

$$= C_p (T_3 - T_4) - C_p (T_2 - T_1)$$

$$= (h_3 - h_4) - (h_2 - h_1) \tag{17}$$

In the absence of pressure losses, the pressure and ideal temperature ratios for the compressor and the turbine are the same.

2. *Work ratio*

Work ratio is given by the following equation.

$$W_R = \frac{\oint W}{W_T} \tag{18}$$

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3. *Thermal efficiency*

$$\eta = \frac{\text{New work done}}{\text{Heat supplied}} \tag{19}$$

$$= \frac{Q_A - Q_R}{Q_A}$$

$$= 1 - \frac{Q_R}{Q_A}$$

$$= \frac{C_p(T_3 - T_4) + C_p(T_2 - T_1)}{C_p(T_3 - T_2)}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{20}$$

$$\eta = \frac{T_3\left(1 - \frac{T_4}{T_3}\right) + T_1\left(\frac{T_2}{T_1} - 1\right)}{T_3 - T_1\left(\frac{T_2}{T_1}\right)}$$

$$= \frac{T_3\left(\eta_t\left(1 - \frac{1}{R}\right)\right) + T_1\left(1 + \frac{1}{\eta_C}(R-1) - 1\right)}{T_3 - T_1\left(1 + \frac{1}{\eta_C}(R-1)\right)}$$

$$= \frac{T_3\eta_t\left(1 - \frac{1}{R}\right) + T_1\frac{1}{\eta_C}(R-1)}{T_3 - T_1\left(1 + \frac{1}{\eta_C}(R-1)\right)} \tag{21}$$

$$= \frac{\left[\frac{T_3}{T_1}\eta_t\left(\frac{R-1}{R}\right) + \frac{1}{\eta_C}(R-1)\right]T_1}{\left[\frac{T_3}{T_1} - \left(1 + \frac{1}{\eta_C}(R-1)\right)\right]T_1}$$

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Assuming maximum temperature ratio,

$\frac{T_3}{T_1} = T_R$ and substituting in the above equation,

$$\eta = \frac{\left[T_R \eta_t \left(\frac{R-1}{R}\right) + \frac{1}{\eta_C}(R-1)\right]}{\left[T_R - \left(1 + \frac{1}{\eta_C}(R-1)\right)\right]}$$

Dividing throughout by $(R-1)$

$$\eta = \frac{\left[\left(\frac{T_R \eta_t}{R}\right) - \frac{1}{\eta_C}\right]}{\left[\frac{T_R}{R-1} - \left(\frac{1}{R-1} + \frac{1}{\eta_C}\right)\right]} \tag{22}$$

$$\eta = \frac{\left[\left(\frac{T_R \eta_t}{R}\right) - \frac{1}{\eta_C}\right]}{\left[\frac{T_R - 1}{R-1} - \frac{1}{\eta_C}\right]} \tag{23}$$

12.1.1.3 Maximum Pressure Ratio

Considering Equation (19):
Work ratio,

$$W_R = 1 - \frac{T_1(R-1)}{T_3 \times \eta_c \times \eta_T \left(1 - \frac{1}{R}\right)}$$

where $R = \beta^{\frac{\gamma-1}{\gamma}}$

For fixed temperature, T_1 and T_3, the maximum pressure ratio is given by the following equation:

$$(\beta)_{max} = \frac{T_3}{T_1}^{\frac{\gamma}{\gamma-1}}$$

$$= T_R^{\frac{\gamma}{\gamma-1}} \tag{24}$$

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12.1.1.4 Optimum Pressure Ratio for Maximum Specific Output

For maximum power output for a given plant size, the pressure ratio that gives maximum available work per kilogram of working fluid is necessary.

Net work done/kg assuming $\eta_t = \eta_i = 1$

$$\oint W = W_T - W_C$$

$$= C_p (T_3 - T_4) - C_p (T_2 - T_1)$$

$$= C_p T_3 \left(1 - \frac{1}{R}\right) - C_p T_1 (R - 1)$$

where $$R = \beta^{\frac{\gamma-1}{\gamma}}$$

As the lower and upper temperatures are fixed, for maximum specific output,

$$\frac{d}{dR} \oint W = 0$$

i.e., $$C_p T_3 R_{opt}^{-2} - C_p T_1 = 0$$

or $$\frac{T_3}{R_{opt}^2} = T_1$$

$\therefore$ $$R_{opt} = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2}} = T_R^{\frac{1}{2}}$$

But, $$R = \beta^{\frac{\gamma-1}{\gamma}}$$

$\therefore$ $$\beta_{opt} = T_R^{\frac{\gamma}{2(\gamma-1)}} \quad (25)$$

Considering compressor and turbine efficiency,

$$\beta_{opt}^{*} = \eta_c \times \eta_T \times T_R^{\frac{\gamma}{2(\gamma-1)}} \quad (26)$$

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12.1.1.5 Optimum Pressure Ratio for Maximum Cycle Thermal Efficiency

Actual heat supplied per kilogram of working fluid is given by the following equation:

$$Q_A = C_p (T_3 - T_2) \approx C_p (T_3 - T_{2'})$$

Now, thermal efficiency is

$$\eta = \frac{(T_3 - T_4) + (T_2 - T_1)}{(T_3 - T_{2'})}$$

$$= \frac{\eta_t (T_3 - T_{4'}) + \frac{1}{\eta_c}(T_{2'} - T_1)}{(T_3 - T_{2'})} = \frac{\eta_t T_3 \left(1 - \frac{T_{4'}}{T_3}\right) + \frac{T_1}{\eta_c}\left(\frac{T_{2'}}{T_1} - 1\right)}{T_1 \left(\frac{T_3}{T_1} - \frac{T_{2'}}{T_1}\right)}$$

As $\frac{T_{2'}}{T_1} = \frac{T_3}{T_{4'}} = \beta^{\frac{\gamma-1}{\gamma}}$,

$$\eta = \frac{\eta_t T_3 \left(1 - \frac{1}{\beta^{\frac{\gamma-1}{\gamma}}}\right) - \frac{T_1}{\eta_c}\left(\beta^{\frac{\gamma-1}{\gamma}} - 1\right)}{T_1 \left(\frac{T_3}{T_1} - \beta^{\frac{\gamma-1}{\gamma}}\right)} \tag{27}$$

As T_1 and T_3 are fixed, the only variable is pressure ratio. Hence, for maximum thermal efficiency, the above equation has to be differentiated with respect to zero.

Substituting $T_R = \frac{T_3}{T_1}$ and $R = \beta^{\frac{\gamma-1}{\gamma}}$ we obtain,

$$\frac{d\eta}{dR} = 0 = \frac{d}{dR} \frac{\eta_t T_R \left[\left(1 - \frac{1}{R}\right) + \frac{1}{\eta_c}(R-1)\right]}{(T_R - R)}$$

Simplifying the equation,

$$0 = \eta_t T_R \frac{1}{R^2}(T_R - R) - \frac{1}{\eta_c}(T_R - R) - \left\{\eta_t T_R \left(1 - \frac{1}{R}\right) - \frac{1}{\eta_c}(R-1)\right\}(-1)$$

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$$0 = \left\{\frac{\eta_t T_R}{R^2}(T_R - R) - \frac{1}{\eta_c}(T_R - R)\right\} + \left\{\eta_t T_R\left(\frac{R-1}{R}\right) - \frac{1}{\eta_c}(R-1)\right\}$$

$$0 = \left\{\frac{\eta_t \eta_c T_R - R^2}{\eta_c R^2}\right\}(T_R - R) + (R-1)\left\{\frac{\eta_t \eta_c T_R - R}{\eta_c R}\right\}$$

$$0 = \left\{\eta_t \eta_c T_R - R^2\right\}(T_R - R) + \left\{\frac{R(R-1)}{\eta_t \eta_c T_R - R}\right\}$$

i.e., $$\left\{\eta_t \eta_c T_R - R^2\right\}(T_R - R) = (R - \eta_t \eta_c T_R) R(R-1)$$

i.e., $$\eta_t \eta_c T_R^2 - T_R R^2 - \eta_t \eta_c T_R R + R^3 = R^3 - \eta_c \eta_t T_R R^2 - R^2 + \eta_c \eta_t T_R R$$

i.e., $$R^2\left\{\eta_t \eta_c T_R + 1 - T_R\right\} - 2\eta_t \eta_c T_R R + \eta_t \eta_c {T_R}^2 = 0$$

i.e., $$R^2\left\{(\eta_t \eta_c - 1)T_R + 1\right\} - 2\eta_t \eta_c T_R R + \eta_t \eta_c {T_R}^2 = 0$$

Multiplying the above equation throughout by $\frac{1}{R^2}$

$$\left\{(\eta_t \eta_c - 1)T_R + 1\right\} - \frac{2\eta_t \eta_c T_R}{R} + \frac{\eta_t \eta_c {T_R}^2}{R^2} = 0$$

i.e., $$\frac{\eta_t \eta_c {T_R}^2}{R^2} - \frac{2\eta_t \eta_c T_R}{R} + \left\{(\eta_t \eta_c - 1)T_R + 1\right\} = 0$$

This is a quadratic equation, and hence it can be solved as follows:

$$\frac{1}{R} = \frac{2\eta_t \eta_c T_R + \sqrt{(2\eta_t \eta_c T_R)^2 - 4(\eta_t \eta_c T_R)\left\{(\eta_t \eta_c - 1)T_R + 1\right\} + 1}}{\left(2\eta_t \eta_c {T_R}^2\right)}$$

$$R = \frac{\left(2\eta_t \eta_c {T_R}^2\right)}{2\eta_t \eta_c T_R + 2\sqrt{(\eta_t \eta_c T_R)^2 - (\eta_t \eta_c T_R)\left\{(\eta_t \eta_c - 1)T_R + 1\right\} + 1}}$$

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$$R = \frac{\left(2\eta_t \eta_c T_R{}^2\right)}{2\eta_t \eta_c T_R \left[1 + \sqrt{(T_R - 1)\left(\frac{1}{\eta_t \eta_c} - 1\right)}\right]}$$

$$R = \frac{T_R}{\left[1 + \sqrt{(T_R - 1)\left(\frac{1}{\eta_t \eta_c} - 1\right)}\right]} \tag{28}$$

Hence, the optimum pressure ratio is given as follows:

$$\beta_{opt} = \left[\frac{T_R}{\left\{1 + \sqrt{(T_R - 1)\left(\frac{1}{\eta_t \eta_c} - 1\right)}\right\}}\right]$$

The variation of specific output and thermal efficiency with pressure ratio can be plotted on the graph as shown in Figures 12.4 and 12.5. The optimum pressure ratio values for maximum specific output and maximum thermal efficiency are not same.

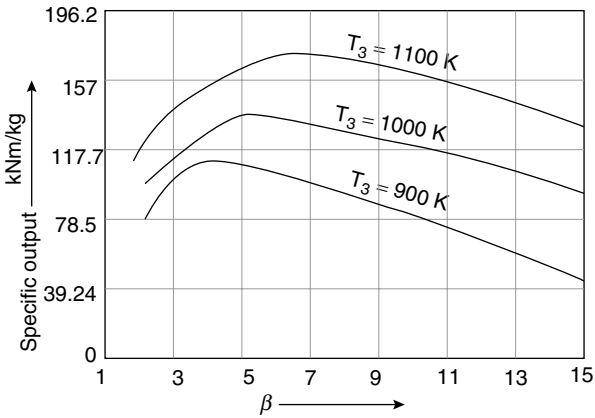


Fig. 12.4 *Variation of Specific Output with Pressure Ratio*

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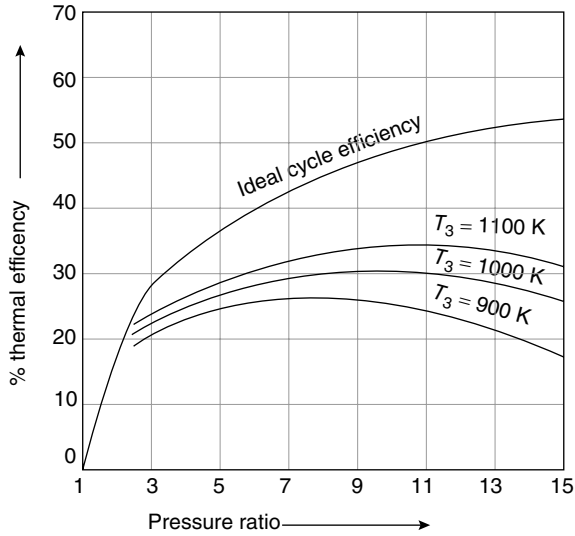


Fig. 12.5 *Thermal Efficiency v/s Pressure Ratio*

For practical applications, optimum pressure ratio for maximum specific output is recommended as this factor decides the plant size.

12.2 CLASSIFICATION OF GAS TURBINES

Gas turbines are classified as (i) constant pressure combustion gas turbine and (ii) constant volume combustion gas turbine.

In the case of constant volume combustion, gas turbine heat addition takes place at constant volume in the combustion chamber. As it entails valves and consequent intermittent operation, it become obsolete and is replaced by constant pressure gas turbine.

12.2.1 Principle of Working of Open- and Closed-Cycle Gas Turbines

The constant pressure gas turbines work based on the following principles:

1. *Direct open cycle*

The direct open-cycle gas turbine is shown in Figure 12.6. In the direct open-cycle gas turbine, air enters the compressor at point 1, where it is compressed to point 2. The air then enters the combustion chamber or the reactor where it receives heat at constant pressure (ideal cycle) and emerges hot at point 3. The gas that comes out of the combustion chamber expands in the turbine to point 4. The hot exhaust mixes with the atmosphere outside the cycle and a fresh cool supply is taken in at point 1. Part of the turbine power is used to run the compressor and auxiliaries.

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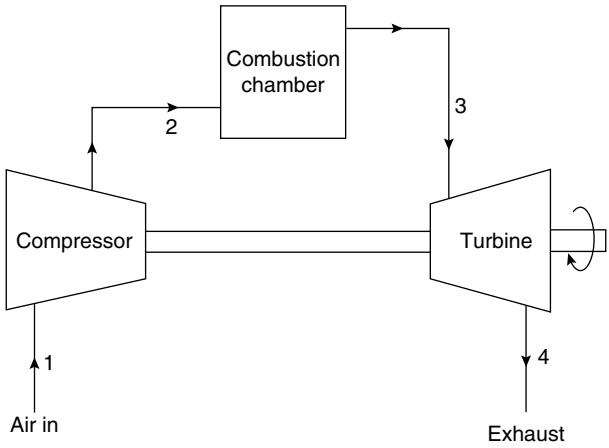


Fig. 12.6 *Direct Open Cycle Gas Turbine*

2. *Indirect open cycle*

An indirect open-cycle gas turbine is shown in Figure 12.7. The elements of the indirect cycle gas turbine are similar to those in the direct open cycle except that here the air is a secondary fluid that receives heat from a primary coolant in a heat exchanger. This cycle is suitable where environment concerns restrict direct air heating as in nuclear reactors, where radioactivity releases may spread to the atmosphere.

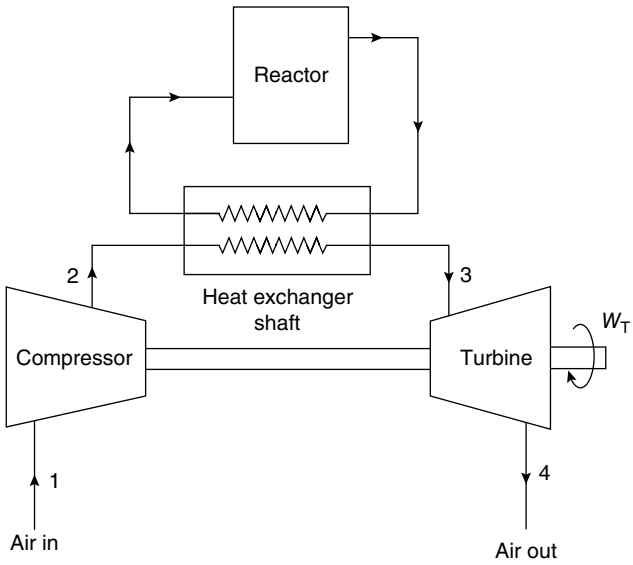


Fig. 12.7 *Indirect Open Cycle Gas Turbine*

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3. *Direct closed cycle*

A direct closed-cycle turbine is shown in Figure 12.8. In the direct closed cycle, the gas coolant is heated in the reactor, expanded through the turbine, cooled in a heat exchanger and compressed back to the reactor. In this cycle, a gas other than air may be used.

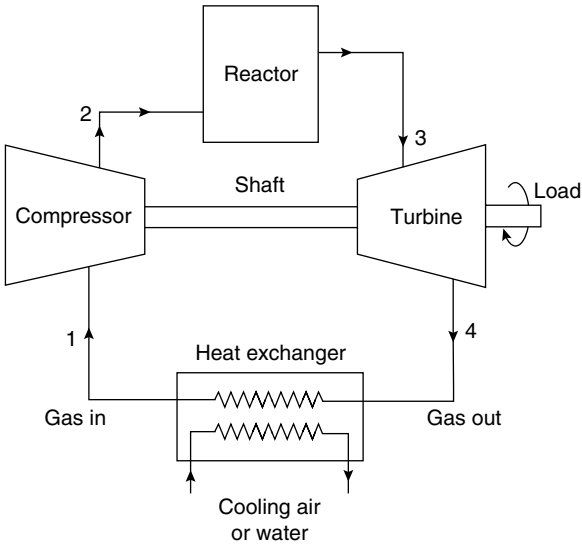


Fig. 12.8 *A Direct Closed-Cycle Turbine*

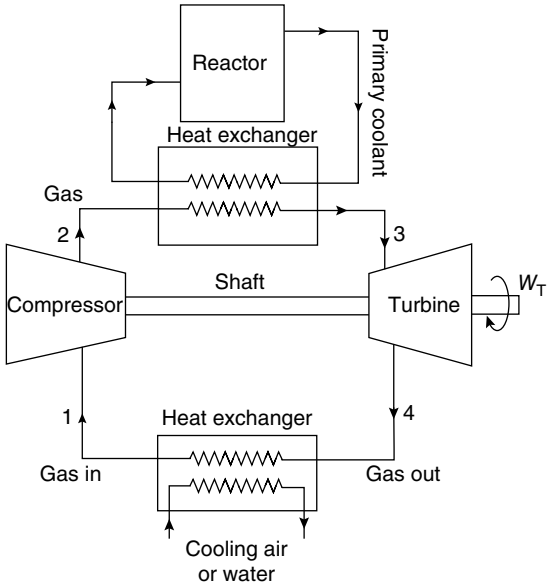


Fig. 12.9 *Indirect Closed-Gas Turbine Cycle*

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No effluent of radioactive gases passes into the atmosphere under normal working conditions. This cycle permits pressurization of the working fluid with consequent reduction in plant size. Helium is the most suitable working fluid in this case.

4. *Indirect closed cycle*

The indirect closed cycle combines the indirect open cycle and direct closed cycle in that the reactor is separated from the working fluid by a heat exchanger. The working gas rejects heat to the atmosphere via a heat exchanger as shown in Figure 12.9. The primary coolant may be water, a liquid metal or a gas such as helium.

12.2.2 Comparison of Open and Closed-Cycle Turbines

S. no.	Open cycle	Closed cycle
1.	Working fluid is usually air.	It uses any fluid as working fluid.
2.	Direct heating of air in the combustion chamber results in erosion and corrosion of turbine blades due to contaminated gases.	Indirect heating eliminates erosion and corrosion problem; hence, efficiency as well as the life of blades is more.
3.	Working pressure is atmospheric, and hence plant size is larger.	Working pressure is above atmospheric, which reduces plant size.
4.	Heat transfer coefficient is small due to lower working pressure.	Heat transfer coefficient is larger due to high working pressure.
5.	Part load efficiency is poor as load regulation is done by varying the turbine inlet temperature.	Better part load efficiency as the load is regulated by changing the fluid pressure by varying the quantity of the working fluid at constant temperature.
6.	Cost of power developed is more as good quality fuel has to be used.	Power developed is economical as inferior quality fuel can be used due to indirect or external heating.
7.	Filtering of incoming air is necessary and which is laborious.	Filtering of air is eliminated due to indirect heating.
8.	Plant efficiency and power output are less due to low working pressure of the working fluid.	Plant efficiency and power output are higher due to higher working pressure.
9.	System is simple.	System is complicated and needs bulky air heaters due to longer heat transfer coefficient of air heaters.
10.	It requires no coolant, and atmosphere itself acts as a coolant.	It requires coolant for pre-cooling turbine exhaust before it enters the compressor.
11.	The system need not be rigid due to lower working pressure.	The system should be rigid and strong to withstand higher working pressure.
12.	System is cheaper.	System must be air tight, and hence costlier.

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12.3 CONSTRUCTION AND PLANT LAYOUT WITH AUXILIARIES

In the gas turbine, a fraction of power developed is utilized to run compressor along with other auxiliaries. In closed-type gas turbines, the working fluid may not be air and the pressure and temperature of the working fluid are raised by means of external heating. Here, the products of combustion do not come in contact with the working fluid.

12.3.1 Components of a Gas Turbine

1. *Turbine*

Turbine drives the compressor and the generator. Both impulse and reaction-type turbines are used but with a few stages. The turbine used should be light in weight and should have high efficiency, reliability and long life. The most commonly used turbines are axial flow types.

2. *Combustion chamber*

Combustion of the fuel takes place in the combustion chamber. The main components of a typical gas turbine combustion chamber are shown in Figure 12.10.

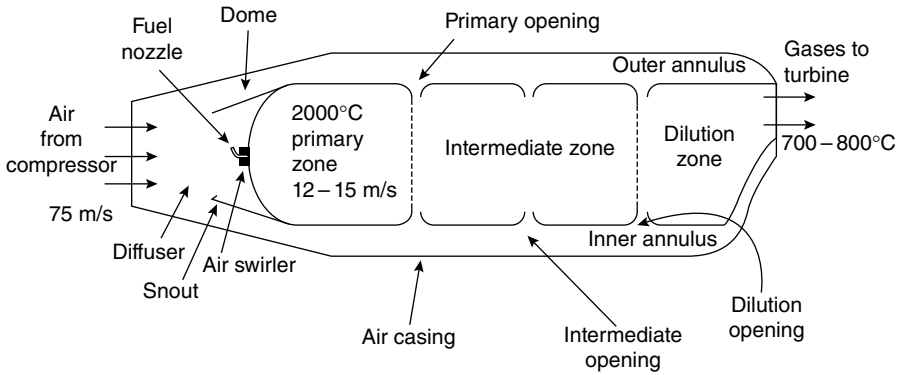


Fig. 12.10 *Main Components of a Gas Turbine Combustion Chamber*

The air–fuel ratio used varies from 60:1 to 120:1. The fuel used may be refined kerosene. Air enters the combustion chamber at a velocity of 75 m/s. About 25 per cent of the air from the compressor is introduced directly to the fuel burner. This is known as primary air. It mixes with the atomized fuel and forms a turbulent mixture for ignition. It prevents the formation of carbon deposits over the burner tips and produces an envelope around the flame, preventing the flame from touching the walls. As the primary air is passed through swirl vanes, it causes reversal of gas flow, and hence stabilizes the flame; that is, it prevents the extinction of flame by high velocity air. The remaining air, known as secondary air, enters the annulus around the flame tube, cooling the upper portion before entering intermediate and dilution holes or openings. Any imperfectly mixed fuel-rich pockets of the gas burns in the intermediate zone which might have formed due to dissociation. The dilution zone further brings down the temperature so as to suit the operating temperature of the turbine. About 20–40 per cent of the total combustor air flow is used for this purpose. A traditional gas turbine combustor showing air and gas passages in all the three zones can be seen in Figure 12.11.

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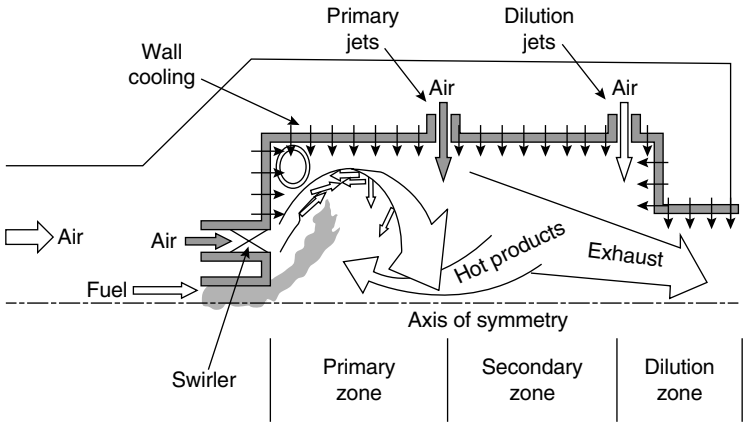


Fig. 12.11 *Air and Gas Flow in a Traditional Gas Turbine Chamber*

3. *Compressor*

Generally used pressure ratios of a gas turbine plant of high efficiency and output is 10:1 or more. For a pressure ratio of 4:1, a single-stage centrifugal compressor is generally preferred. This is because a centrifugal compressor can operate over a wide range of mass flow rate than an axial flow compressor. For higher pressure ratios instead of using multi-stage centrifugal compressors, axial flow compressors are found most suitable. For pressures ratios, more than 6:1, more than one compressor can be used. Axial flow compressors are heavier than centrifugal flow compressors but have higher efficiency. Reciprocating compressors are generally not preferred because they are heavy, slow, have more sliding parts and cannot handle large volumes of air.

12.3.2 Centrifugal Compressor

A centrifugal compressor is a power-absorbing turbomachine in which the angular momentum of the fluid flowing through the impeller is increased. Fans, blowers, superchargers, boosters, etc., can be treated as centrifugal type depending on pressures and field of applications. A pressure ratio ranging from 4:1 to 6:1 can be obtained using a single-stage compressor. Even though it has high specific fuel consumption, a centrifugal compressor has the following advantages over an axial flow–type compressor:

(i) It occupies less space.

(ii) It operates over a wide range of mass flow rate at any rotational speed.

(iii) It is robust in construction and cheaper to produce.

(iv) Its performance is unaffected by build-up of deposits on the surfaces of air channels.

(v) Its performance is not affected by atmospheric contamination.

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12.3.3 Main Parts of a Centrifugal Compressor

Figure 12.12 shows centrifugal compressor stage consisting mainly of four parts:

1. *Inlet casing* – The function of the inlet casing is to accelerate the fluid to the impeller inlet using a convergent nozzle. The outlet of the inlet casing is known as eye.
2. *Impeller* – The function of the impeller is to increase the kinetic energy of the fluid and the static pressure.
3. *Diffuser* – It transforms the high kinetic energy of the fluid at the impeller outlet into static pressure. The diffuser may be with vanes or may be vaneless.
4. *Outlet casing* – It is also known as volute or scroll casing. It collects the fluid from the diffuser.

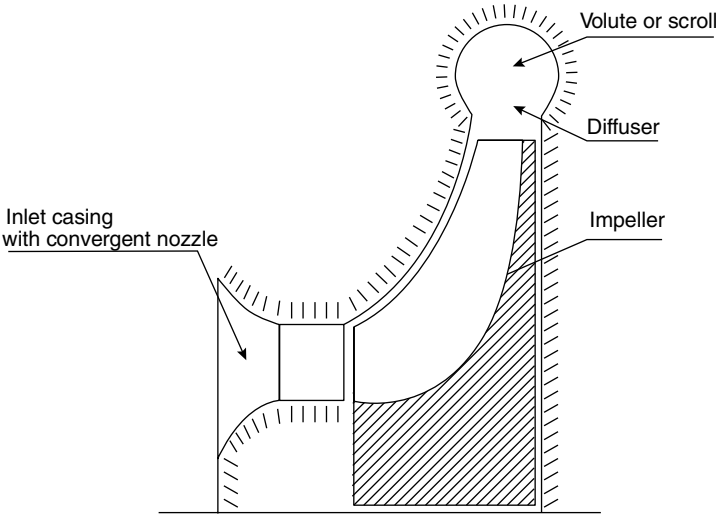


Fig. 12.12 *Section of a Centrifugal Compressor Stage*

12.3.4 Impeller and Diffuser

1. *Impeller*

Figure 12.13 shows different components of an impeller. It consists of impeller vanes, hub, shroud and an inducer.

(i) Impeller vanes are used to transfer energy from the impeller to flowing fluid.

(ii) The hub corresponds to surface AB. Air is drawn near the hub known as impeller eye.

(iii) The shroud corresponding to surface CD. If the impeller is enclosed by a shroud, it is known as shrouded impeller.

(iv) The inducer corresponding to section EF in impeller increases the angular momentum of the fluid without increasing its radius of rotation.

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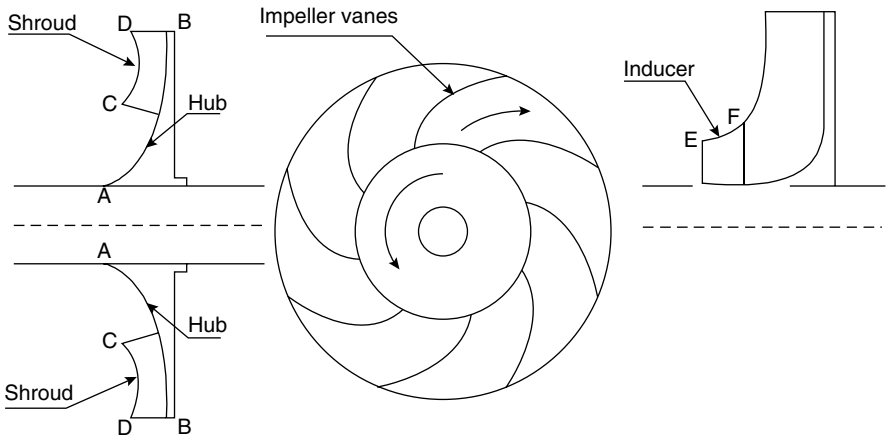


Fig. 12.13 *Impeller Components*

The impeller may be either single sided or double sided. In the latter case, the fluid is drawn on both sides and the impeller is subjected to approximately equal forces in an axial direction.

2. *Diffuser*

The diffuser may be either vaned diffuser or vaneless diffuser. The vaneless diffuser has only the annual space without having any guide vanes. The diffuser increases the static pressure of the fluid by reducing kinetic energy.

Figure 12.14 shows the variation of pressure in a centrifugal compressor.

When air is accelerated in the nozzle next to the inlet casing, its velocity increases, while entering the impeller with corresponding drop in static pressure ($a - b$). In the impeller, blades impart swirling motion to air resulting in high velocity at impeller tip. Energy transfer takes place resulting in increased static pressure of air ($b - c$) due to centripetal acceleration. Air further enters the diffuser wherein high velocity air leaving the impeller is converted into pressure ($c - d$). This high-pressure air is collected in the collector or volute that delivers it.

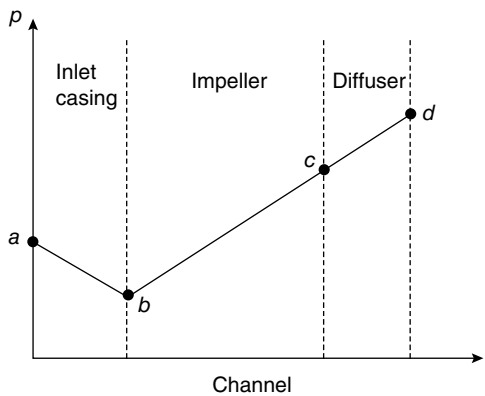


Fig. 12.14 *Variation of Pressure in a Centrifugal Compressor*

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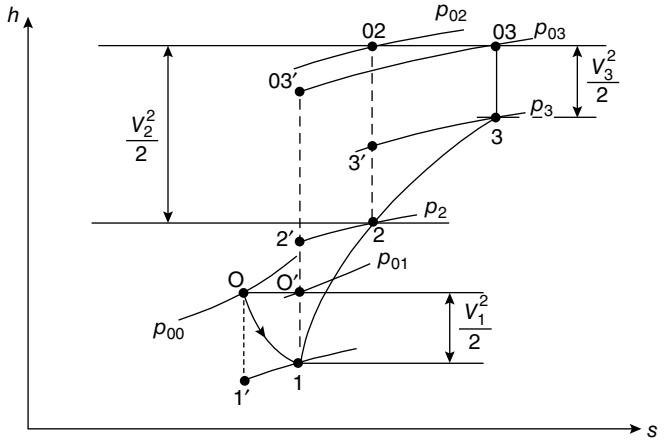


Fig. 12.15 *Compression Process in a Centrifugal Compressor Stage*

Figure 12.15 shows the compression process in a centrifugal compressor stage. State point 01 corresponds to ambient air conditions.

As air is accelerated in the inlet casing, it causes a pressure drop from p_{01} to p_1 or p_{00} to p_1 assuming losses in inlet casing. The enthalpy change during this process is equal to a change in kinetic energy of processes 01–03′ shows isentropic compression corresponding to delivery stagnation pressure. Processes 1–2 and 1–2′ represent actual compression and isentropic compression in the impeller, respectively.

Pressure p_{02} corresponds to delivery pressure if the total kinetic energy leaving the impeller is converted into pressure. Since the diffusion process is not isentropic, the final delivery pressure is corresponding to static pressure p_3. Energy corresponds to the kinetic energy in the collector. $p_{03} < p_{02}$ as the diffusion process is irreversible.

12.3.5 Diffuser

Diffusers used in centrifugal compressors may be either vaned or vaneless. Since a considerable amount of energy is transformed in the diffuser, its efficient transformation is to be taken care of Design of diffuser is very important as losses due to flow of fluid against the adverse pressure gradient may be high.

12.3.5.1 Vaneless Diffuser

In a vaneless diffuser, the fluid is diffused around the impeller before leaving the stage in a vaneless space. The diffuser gains static pressure rise due to diffusing action of the fluid from a small diameter to a larger diameter (d_3). The corresponding area of cross section in the radial direction are as follows:

$$A_2 = \pi d_2 b_2$$

and

$$A_3 = \pi d_3 b_3$$

This type of flow in a vaneless space is a free vortex flow, the angular momentum remaining constant.

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Figure 12.16 shows a vaneless diffuser. It consists of an annular passage without vanes surrounding the impeller. The width of the annulus may be constant or may be tapered outwards to provide more rapid diffusion. The taper angle θ varies from 4° to 8°. Some of the disadvantages of vaneless diffuser are as follows.

(i) Large-sized diffusers with larger diameter are required to reduce the velocity.

(ii) It has lower efficiency owing to long fluid flow path.

(iii) It can be used only for small a pressure rise.

However, it is used for industrial applications where large-sized compressors are needed. It is more economical and gives wider range of operation.

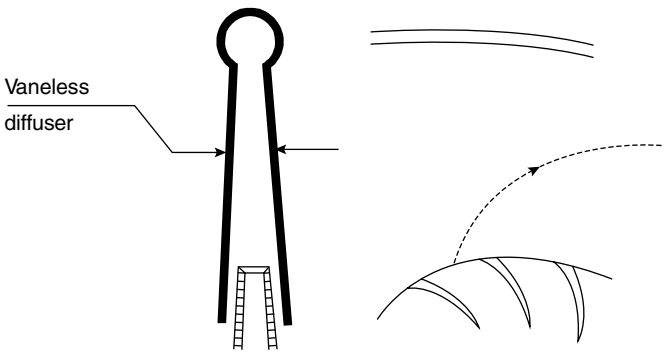


Fig. 12.16 *Vaneless Diffuser*

12.3.6 Axial Centrifugal Compressor

A schematic diagram of an axial flow compressor is shown in Figure 12.17. It consists of a number of fixed blades attached to the casing, and alternate rows of moving blades are fixed

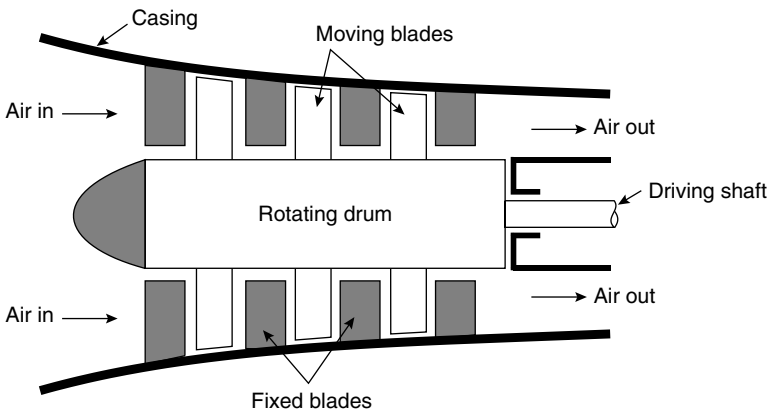


Fig. 12.17 *Axial Flow Compressor with Drum-Type Rotor*

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on a central drum, which rotates along with the shaft. Air progresses from one blade row to the next and is guided by the fixed blades. Fixed blades serve the function of a diffuser, and hence the pressure of air increases when it comes out of the duct. As air passes parallel to the axis of rotation of the shaft, the flow is axial, and hence the name axial flow compressor. These compressors usually run at high speed (10,000–30,000 rpm) and are used to supply large quantity of air as in aircraft gas turbines. The pressure ratio of compression may be about 10:1.

Due to rapid compression, the time available for heat transfer is less and hence compression is nearly adiabatic. However, due to generation of friction and turbulence, internal energy within air generates temperature that is higher than the adiabatic temperature. Thus, the index 'n' is greater than adiabatic index 'γ' for a rotary compressor.

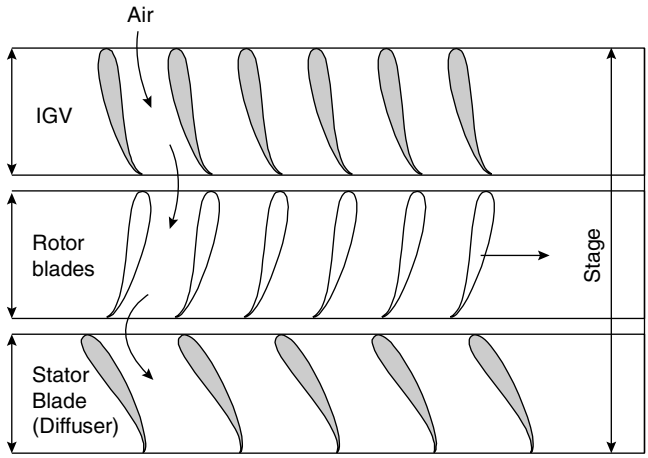


Fig. 12.18 *An Axial Compressor Stage*

As shown in Figure 12.18, the compressor consists of alternate sets of moving and fixed blades. The sets of fixed blades that are spaced around an outer stationary casing are called stator blades, and the sets of moving blades fixed to a spindle are called rotor blades. A very small tip clearance at the end of the stator and rotor blades is provided to ensure smooth and efficient flow. A compressor stage consists of one set of stator blades and one set of rotor blades. Depending on the pressure ratio required, stage may be either single or multiple. In multistage arrangement, the successive sets of blades are reduced in length to compensate for reduction in volume resulting due to pressure increase from stage to stage.

An axial compressor may have either disc-type rotor or drum-type rotor. For the same overall weight, centrifugal stresses are lower in the disc type than in the drum type.

The basic principle of operation of an axial flow-type compressor is similar to that of a centrifugal type. Here, the rotating blades impart kinetic energy to air that is further converted into pressure rise. Stator recovers part of kinetic energy imparted to the working fluid and redirects the fluid to the rotating blades of next stage at a convenient angle. Air is guided to the rotor blades using inlet guide vanes (IGVs) as shown in Figure 12.19. Air enters the combustion chamber through one to three rows of diffuser blades or straightener blades installed after the last stage. This is to ensure that air is straightened and skewed down before it enters the combustion chamber.

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12.3.7 Stage Velocity Triangles

Figure 12.19 shows the velocity triangles for an axial compressor stage. Subscripts 1, 2, and 3 correspond to condition of air at entry to rotor, exit from rotor and diffuser blades respectively.

Let α_1, α_2, α_3 and β_1, β_2 be the air angles in the absolute and relative systems, respectively. Subscripts *a* and *u* represent axial and tangential component of absolute velocity, respectively, and *ru* represents tangential component of relative velocity V_r.

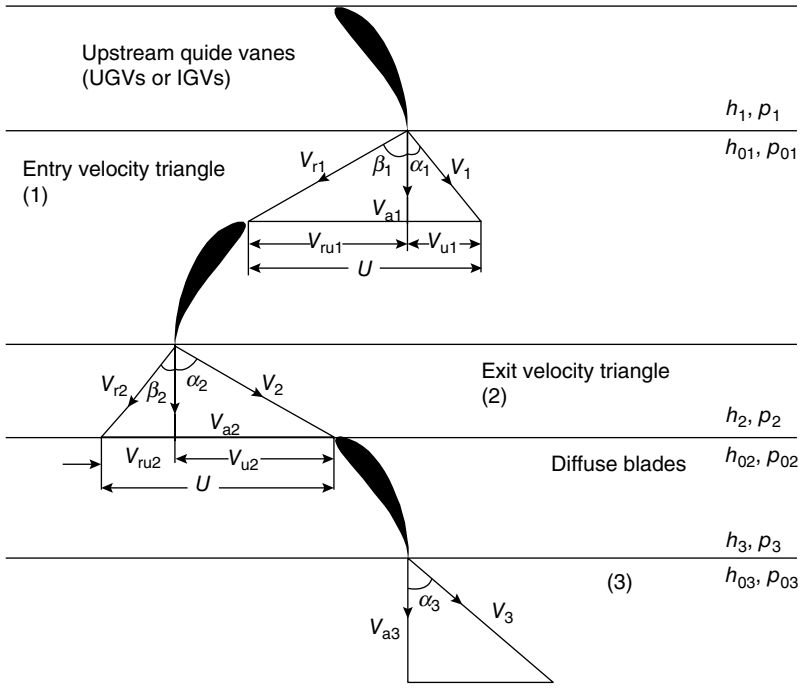


Fig. 12.19 *The Velocity Triangles*

12.3.8 Work Input to Compressor

In the case of an axial flow compressor, it is assumed that the axial velocity V_a is constant throughout. In order to have this condition, it is necessary to maintain a converging area of flow as pressure increases in every stage.

Work input:

$$W = U(V_{u2} - V_{u1})$$
$$= U(V_a \tan \alpha_2 - V_a \tan \alpha_1)$$
$$W = UV_a(\tan \alpha_2 - \tan \alpha_1)$$
$$W = UV_a(\tan \beta_1 - \tan \beta_2)$$

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From Euler's equation,

$$E = \frac{1}{2}\,(V_2^2 - V_1^2) + \frac{1}{2}\,(V_{r1}^2 - V_{r2}^2)$$

as U = constant throughout.

12.3.9 Work Done Factor

It can be noted that due to secondary flows and growth of the boundary layers on the hub and the casing of the compressor annulus, the axial velocity is not uniform. In a multistage compressor, this effect is considerable after the first stage and the degree of distortion depends on the number of stages. Owing to this effect, the axial velocity in the hub and tip regions is much lesser than the mean value; whereas in the central zone, it is much higher than mean value.

The work-absorbing capacity decreases with an increase in axial velocity and vice versa. Hence, the work-absorbing capacity is decreased in the central region as compared to hub and tip regions. However, the increase in work at the hub and tip regions is not expected in the actual practice due to higher losses. The net result is the work-absorbing capacity in a stage is less than that given by Euler's equation. This reduction in the work-absorbing capacity of the stage is taken into account by a factor known as work done factor, denoted by Ω. Its value ranges from 0.98 to 0.85 depending on number of stages.

Considering work done factor, work done on air is given by

$$W = \Omega\, UV_a\,(\tan\beta_1 - \tan\beta_2)$$

Since,

$$W = C_P\,\Delta T_s = C_P\,(To_2 - To_1),$$

$$\Delta T_s = \frac{\Omega U V_a}{C_P}\,(\tan\beta_1 - \tan\beta_2)$$

Where ΔT_s is the stagnation temperature rise in stage.

Therefore, work done factor is a measure of the ratio of the actual work-absorbing capacity of the stage to its ideal value.

12.3.10 Degree of Reaction

The degree of reaction for axial compressors is given by

$$R = \frac{\text{Actual change of enthalpy in the rotor}}{\text{Actual change of enthalpy in the stage}}$$

$$R = \frac{h_2 - h_1}{h_3 - h_1} = \frac{T_2 - T_1}{T_3 - T_1}$$

If $V_1 = V_3$, then

$$R = \frac{h_2 - h_1}{h_{03} - h_{01}}$$

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$$= \frac{V_1^2 - V_{r2}^2}{U\left(V_{u2} - V_{u1}\right)}$$

In terms of air angles,

$$R = \frac{V_a^2\left(\tan^2\beta_1 - \tan^2\beta_2\right)}{2UV_a\left(\tan\beta_1 - \tan\beta_2\right)}$$

$$R = \frac{1}{2}\frac{V_a}{U}\left(\tan\beta_1 + \tan\beta_2\right)$$

Substituting $\frac{V_a}{U} = \phi$ = flow coefficient

$$\tan\beta_m = 1/2\ (\tan\beta_1 + \tan\beta_2)$$

= mean relative air angle

then,

$$R = \phi\ \tan\beta_m$$

$\therefore$

$$R = \frac{1}{2} - \frac{1}{2}\frac{V_a}{U}\left(\tan\alpha_1 - \tan\beta_2\right)$$

12.4 FUEL AND FUEL SYSTEMS

Fuel supply is one of the most important factors that govern the operation of gas turbines. Fuel that mismatches with the gas turbine combustion requirements may seriously affect the operation and maintenance apart from load shedding and turbine tripping. Fuels used in gas turbines can be broadly classified as gaseous and liquid fuels:

The most commonly used gaseous fuels are the following:

(i) Natural gas
(ii) Liquefied natural gas (LNG)
(iii) Liquefied petroleum gas (LPG)
(iv) Refinery gas
(v) Coke oven gas
(vi) Hydrogen
(vii) Coal gas

The most commonly used liquid fuels are the following:

(i) No. 2 diesel
(ii) Kerosene
(iii) Jet A

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(iv) Naphtha

(v) Condensates

(vi) Ethanol and methanol

(vii) Heavy residual grade oils and crude oils

12.4.1 Natural Gas

Selection of fuel influences the performance of gas turbines. In case of a simple gas turbine, natural gas yields 2–3 per cent more power when compared to distillate oil, as it has higher specific heat of combustion products due to higher hydrogen to carbon ratio (H/C ratio). Higher H/C ratio results in more water vapour formation.

Natural gas mainly contains methane and sometimes traces of ethane, propane and pentanes, which increase the heating value. Presence of inert gases such as nitrogen and carbon dioxide generally decreases the heating value. Table 12.1 list down the key properties of methane and propane.

12.4.2 Liquefied Natural Gas

Owing to rapid growth in liquefaction trains and worldwide shipment, LNG has become an important fuel for gas turbine operation. This is despite of LNG having traces of inert gases that reduce its heating value.

Table 12.1 *Key Properties of Methane and Propane*

Hydrocarbon	Methane (CH4)	Propane (C3H8)
Molecular weight	16	44
Carbon (% weight)	75	82
Hydrogen (% weight)	25	18
C/H ratio	3	4.5
Lower heating value, kJ/kg	50,044	45,227

As can be seen from Table 12.1, methane has higher heat value compared to propane towing to which it produces more water vapour and carbon dioxide, resulting in higher specific heat and hence turbine power. It can be noticed that fuels with higher hydrogen content tend to generate more power. Some gaseous fuels used for gas turbines and their heating values are given in Table 12.2.

12.4.3 Liquid Fuels

For the land-based turbines, natural gas offers convenient delivery and handling and prefers to be continued as the fuel, by offering clean and efficient combustion properties. Due to the 'location-specific' availability of this fuel around the globe or due to scarcity, liquid fuels

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ranging from light distillates to residual grade oils or unrefined crude oils are gaining importance as alternative gas turbine fuels. Liquid distillates such as diesel oil and kerosene are also very widely used as secondary backup for gas turbine fuels in dual fuel applications. The following Table 12.3 gives the information on liquid fuels for gas turbine and their heating values.

Table 12.2 *Gaseous Fuels for Gas Turbine and their Heating Values**

Gaseous fuel	Typical LHV (MJ/Nm^3)
Natural gas, LNG	35.5
LPG typical blend	104.8
Propane	91.2
Butane	118.5
Blast furnace gas	3.6
Coke oven gas	11.8
Hydrogen	10.8
Refinery gas	Varies

* *Source:* Beareau of Energy Efficienry.

Table 12.3 *Liquid Fuels for Gas Turbine and their Heating Values [1]*

Liquid fuel	Typical LHV (MJ/kg)
No.2 Distillate	42.7
Kerosene	43
Jet A	43.2
Naphtha	44.2
Condensate fuel	45.2
Methanol	19.9
Ethanol	26.8
Crude oil	41.2
Heavy fuel oil	39.6

12.5 GAS TURBINE MATERIAL

Advancements made in the field of materials have contributed in a major way in building gas turbine engines with higher power ratings and efficiency levels. Improvements in design of the gas turbine engines over the years have importantly been due to development of materials with enhanced performance levels. Gas turbines have been widely utilized in power generation, industrial sector, marine and aircraft engines. In gas turbine plant major components, turbine blades in particular are critically exposed to very high temperatures continuously unlike in an IC engine. In this context, gas turbine materials should posses the following properties:

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(i) Higher capability to withstand elevated temperature service and more engine efficiency
(ii) High temperature to weight ratio for weight reduction
(iii) Ability to withstand to fatigue- and/or creep-loading and thermal shocks.
(iv) High level of mechanical properties and excellent resistance to oxidation/hot corrosion resistance
(v) Easy machinability, good castability and forgeability
(vi) Ability to maintain structural stability at elevated temperature

The following Table 12.4 provides the component characteristics and materials used in gas turbines.

Table 12.4 *Component Characteristics and Materials Used in Gas Turbines*

Component	Characteristics	Materials
Turbine rotor discs	• Ability to withstand very high temperature • Ability to withstand very high centrifugal stresses	• Austenitic steels with traces of chromium (12–18%), nickel (8–12%), molybdenum and titanium • Ferritic steels for low-temperature applications
Compressor	• Impeller experiences very high pressure and thermal stresses. • High centrifugal stresses	• Titanium and nickel alloys (Inconel 625) for blades • Aluminium alloys to minimize centrifugal stresses
Combustion chamber	• Ability to withstand very high temperature • Higher creep resistance and resistance to hot corrosion	• Nickel and titanium alloys with ceramic coatings. • Nickel chromium alloy – Nimonic-75
Turbine blades	• Must endure the intense heat of the engine • Resistance to oxidation, scaling and deformation • Good fatigue and creep strength	• Stainless steel alloys • Nickel chromium alloy – Nimonic 8–20 with/without ceramic coatings

The following Table 12.5 provides the component characteristics and materials used in gas turbines.

Table 12.5 *Component Characteristics and Materials Used in Aeroengines*

Component	Characteristics	Materials
Suction fan	• Should be extremely strong so that it does not fracture when birds or other debris are sucked into the blades	• Titanium alloys
Intermediate compressor	• Experiences very high pressure	• Titanium and nickel alloys (Inconel 625)

(*Continued*)

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Table 12.5 (*Continued*)

Component	Characteristics	Materials
Combustion chamber	• Ability to withstand very high temperature	• Nickel and titanium alloys with ceramic coatings
Turbine blades	• Must endure the intense heat of the engine	• Titanium and single crystal castings of nickel alloys with ceramic coatings
Exhaust systems (i) Inner duct (ii) Outer duct	• Should posses very high strength and resistant to high temperature	(i) Titanium alloys (ii) Composites synthetic fibres held together with resins

12.6 ADVANTAGES AND DISADVANTAGES OF THE GAS TURBINE PLANT

Advantages over diesel plants

(i) Due to perfect balancing of gas turbine unit, the gas turbine plants are subjected to less vibrations.

(ii) The mechanical efficiency of gas turbine plant is as high as 95 per cent, whereas the mechanical efficiency of diesel plant is less.

(iii) The torque characteristics of turbine plants are far better than diesel plants as it is a continuous power developing system.

(iv) The work developed per kilogram of air is large compared with diesel plant as the expansion of gases to atmospheric pressure is possible.

(v) The weight of gas turbine plant is hardly 0.15 kg/H.P, whereas the weight of diesel plant is 2.5 kg/H.P. Therefore, the space required and capital cost of the gas turbine plant is considerably less than diesel plant.

(vi) All the parts of gas turbine (compressor, combustion chamber and turbine) can be designed and tested individually, and they can be arranged as per requirements as the compression, combustion and expansion take place in different units.

(vii) The running speed of the turbine (4000–100,000 rpm) is considerably large compared with diesel engine (1000–2000 rpm).

(viii) The lubrication and ignition systems are more simplified compared with diesel plant.

(ix) The specific fuel consumption does not increase with time in gas turbine plant as rapidly as in diesel plants.

(x) The installation and maintenance costs are less than diesel plants.

(xi) The exhaust of gas turbine is free from smoke as the quantity of air supplied is four times greater than theoretically required for complete combustion.

Smokeless combustion has been achieved by the gas turbine with proper control of primary zone mixture ratios and methods of secondary air introduction, and

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recirculation could produce reduced pollution conditions. It is claimed that the exhaust gases that come out are smokeless without increasing the production of nitric oxide emissions.

(xii) Any poor-quality fuels can be used in gas turbine plant while special grade fuels are required for diesel engines to avoid knocking. This is a great advantage of gas turbine plant.

The disadvantages of gas turbine plants over diesel plants can be summarized as easier maintenance, improved reliability, lower initial cost, smaller plant dimensions and less space requirements for same output, less vibrations, absence of cyclic variations and greater starting torque.

The disadvantages of gas turbine plants over diesel power plants include poor part load efficiency, requirement of special metals and alloys for different components of the plants, special cooling methods for cooling the turbine blades, low starting torque and less life.

Advantages over steam power plant

(i) The handling of ash is a major problem in steam plants, which is completely eliminated in open-cycle gas turbine plants using gas or liquid as fuel.

(ii) The cubic capacity of the buildings required for gas turbine plant is about 50 per cent of steam plant. The total weight of the materials required for gas turbine plant is also 50–60 per cent of steam plant. Therefore, considerable saving in capital cost is possibly having the same efficiency as steam plant. The gas turbine plants can be installed at selected load centres as space requirement is considerably less where steam plant could not be accommodated.

(iii) The site of the steam power plant is dictated by the availability of large cooling water, whereas an open-cycle gas turbine plant can be located near load centres as no cooling water is required. The cooling water required for closed-cycle gas turbine plant is hardly 10 per cent of the steam plant.

(iv) The comparison of thermal efficiency is shown in Figure 12.5. It is obvious from the figure that the gas turbine plant should be operated above 550°C to compete with the steam plants. Above 375°C, the gap between the steam cycle and gas turbine cycle efficiency widens. Above 550°C, the efficiency of the gas turbine plant increases three times as fast as the steam cycle efficiency for a given temperature increase.

(v) The ratio of exhaust to inlet volume for the same pressure and initial temperature conditions would be only 3.95 in case of gas turbine plant as against 250 for steam plant.

(vi) Steam plant developing power at 17 per cent efficiency costs equal to the gas turbine plant developing power at 34 per cent. This being the case, the gas turbine plant as a base load plant has little future. This is economical as base load plant only where the fuel oil is available at a considerably cheaper rate.

(vii) The gas turbine plant uses fewer auxiliaries than the steam turbine plant; therefore, small size of the gas turbine components enables complete work-tested units to be transported to the site.

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(viii) The ease and rapidity with which the output of gas turbine plants can be made available from a cold start permit a reduction in the capacity of steam plant held in reserve in a hot condition and needed to cover for short load prediction errors and random load fluctuations. The gas turbine plants peak-up the load hardly within 15 minutes. This loading response can be put to good effect in the event of an unscheduled disconnection of a large generator.

(ix) The gas turbine plants are always desirable as peak load plants irrespective of the cost due to their good response, but its adoption as base load is justified on the economic ground. It can be used as a base load plant only where the gas turbine fuel costs are considerably less.

(x) In general, gas turbines can be built relatively quicker and require much less space and civil engineering works and water supplies.

(xi) The components and circuits of a gas turbine plant can be arranged to give the most economic results in any given circumstances, which is not possible in the case of steam power plants.

(xii) The gas turbine becomes more economical for operating conditions below a given load factor as saving on the capital charges outweighs the additional cost of fuel.

(xiii) The gas turbine plant as peak load plant is more preferable as it can be brought on load quickly and surely. The fuel consumption is of secondary importance, since the time of operation is limited.

12.6.1 Advantages and Disadvantages of Gas Turbine Plant

Advantages

(i) It is smaller in size and weight compared to an equivalent steam power plant. For large capacities, it is smaller in size than a comparable diesel engine plant. Hence, it is most suitable where size and weight are the main considerations, such as in ships, aircraft engines, and locomotives.

(ii) It has low initial cost and low operating costs compared to steam power plant of same capacity.

(iii) It is available at relatively short delivery times and is quick to install and put to use.

(iv) It is quick starting and smooth running.

(v) It offers flexibility in supplying process needs such as compressed air in addition to electric power and in using a range of liquid and gaseous fuels and even low-calorific-value fuels.

(vi) Water requirement is less compared to a condensing steam power plant.

(vii) No stand by losses as in a steam power plant in which boilers are operated even when turbine in not loaded.

(viii) Easy to lubricate the compressor, turbine bearings and maintenance cost in less.

(ix) Plant does not require heavy foundations and building and subject to fewer environmental restrictions.

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(x) Due to higher operating temperature, thermal efficiency is more.

Disadvantages

(i) Overall cycle efficiency is less as a portion of power generated by the turbine is consumed by the compressors.

(ii) The plant is incompatible with solid fuels.

(iii) It is costlier due to the use of high heat-resistant materials and sophisticated manufacturing processes for blade manufacture.

(iv) It runs at comparatively higher speeds, and hence requires a reduction gear for normal industrial applications.

12.7 METHOD OF IMPROVING OUTPUT AND PERFORMANCE

The following methods are used to improve the output and performance of a gas turbine.

12.7.1 Reheating

Plant efficiency can be improved by reheating of the gas after it expands in the high-pressure turbine as shown in Figure 12.20.

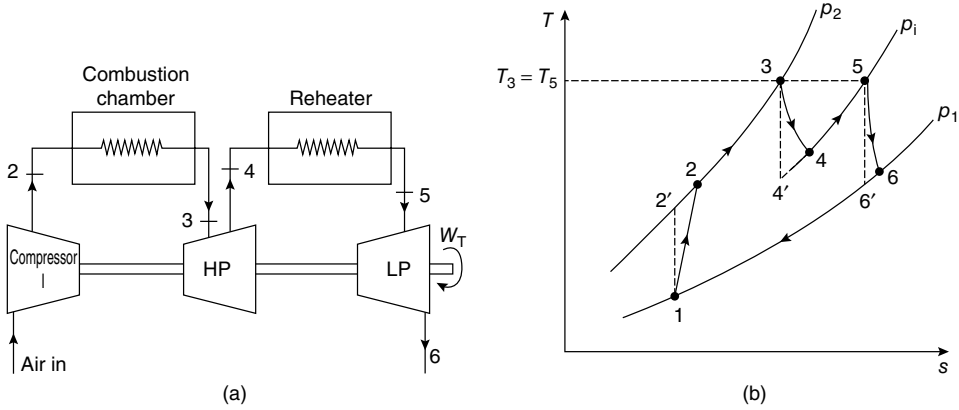


Fig. 12.20 *(a) Gas Turbine Plant with Reheater (b) T-s Diagram with Reheater*

For the same expansion ratio with reheating in two stages as compared to single-stage expansion, the work output will be increased as seen on the *T-s* diagram. Usually, the upper limit for reheating is $T_3 = T_5$.

Now, turbine work output from LP and HP turbines is

$$W_T = C_{pg}(T_3 - T_4) + C_{pg}(T_5 - T_6)$$

$$= C_{pg}\,\eta_{t1}(T_3 - T_4') + C_{pg}\,\eta_{t2}(T_5 - T_6')$$

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$$W_T = \eta_{t1} C_{pg} T_3 \left(1 - \frac{T_{4'}}{T_3}\right) + \eta_{t2} C_{pg} T_5 \left(1 - \frac{T_{6'}}{T_5}\right) \tag{1}$$

where η_{t1}, η_{t2} = HP and LP turbine efficiencies
C_{pg} = specific heat of the gas

For the isentropic processes 3–4′ and 5–6′,

$$\frac{T_3}{T_{4'}} = \left(\frac{p_2}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \text{ and } \frac{T_5}{T_{6'}} = \left(\frac{p_i}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

Substituting in Equation (1) above with $\frac{\gamma-1}{\gamma} = x$

$$W_T = \eta_{t1} C_{pg} T_3 \left(1 - \left(\frac{p_i}{p_2}\right)^x\right) + \eta_{t2} C_{pg} T_5 \left(1 - \left(\frac{p_1}{p_i}\right)^x\right) \tag{2}$$

In the above equation with T_3, p_1 and p_2 fixed, only variable is p_i. Hence for maximum work output,

$$\frac{dW_T}{dp_i} = 0$$

i. e.,
$$-\frac{d}{dp_i}\eta_{t1} T_3 \left(\frac{p_i}{p_2}\right)^x - \frac{d}{dp_i}\eta_{t1} T_5 \left(\frac{p_1}{p_i}\right)^x = 0$$

$$-\eta_{t1} T_3 x p_2^{-x} p_i^{x-1} - \eta_{t2} T_5 (-x) p_1^x p_i^{-(x-1)} = 0$$

$$\eta_{t1} T_3 p_2^{-x} p_i^{x-1} = \eta_{t2} T_5 p_i^{-(x-1)} = 0$$

$$\frac{T_3}{5^{\frac{0.4}{1.4}}} = \frac{\eta_{t2}}{\eta_{t1}} \times \frac{T_5}{T_3} \times (p_1 p_2)^x$$

Hence,
$$p_i^{2x} = \frac{\eta_{t2}}{\eta_{t1}} \times \frac{T_5}{T_3} \times (p_1 p_2)^x$$

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$$\therefore \quad p_i = \sqrt{\left(\frac{\eta_{t2}}{\eta_{t1}} \times \frac{T_5}{T_3}\right)^{\frac{1}{x}} \times (p_1 p_2)}$$

i. e.,
$$p_i = \sqrt{\left(\frac{\eta_{t2}}{\eta_{t1}} \times T_R\right)^{\frac{\gamma}{\gamma-1}} \times (p_1 p_2)}$$

Hence intermediate pressure

$$p_i = \sqrt{(p_1 p_2)\left(\frac{\eta_{t2}}{\eta_{t1}} \times T_R\right)^{\frac{\gamma}{\gamma-1}}} \tag{3}$$

When, $T_5 = T_3$, $T_R = 1$ and if $\eta_{t2} = \eta_{t1}$,

$$p_i = \sqrt{p_1 p_2} \tag{4}$$

Thermal efficiency of the cycle is given by the following equation:

$$\eta = \frac{W_T - W_C}{Q_A}$$

$$= \frac{(T_3 - T_4) + (T_5 - T_6) - (T_2 - T_1)}{(T_3 - T_2) + (T_5 - T_4)} \tag{5}$$

12.7.2 Regeneration

The thermal efficiency and specific output of the gas turbine cycle can be increased by using regeneration method. In this method, the heat in the exhaust gases is used to increase the temperature of air that comes out of the compressor. This increases the temperature of air and reduces the fuel consumption in the combustion chamber. A gas turbine plant fitted with a regenerator or heat exchanger is shown in Figure 12.21.

In perfect regeneration, under ideal conditions, temperature of compressed air that comes out of the compressor at temperature T_2 is increased to T_5 and temperature of air after expansion in the turbine is reduced from T_4 to T_6 as shown on the *T-s* diagram in Figure 12.22.

Thus, $T_5 = T_4$ and $T_6 = T_2$, and it can be noticed that the turbine and compressor work are unchanged.

Hence, the thermal efficiency of the cycle is modified as follows:

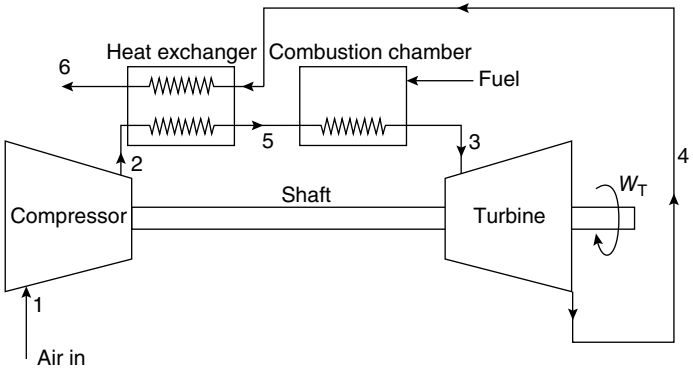
$$\eta = \frac{C_p (T_3 - T_4) + C_p (T_2 - T_1)}{C_p (T_3 - T_5)} \tag{1}$$

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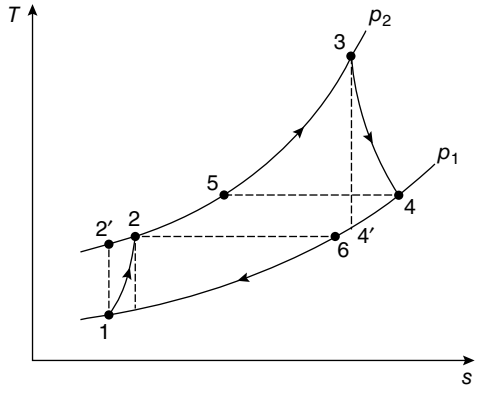
$$\eta = \frac{(T_3 - T_5) + (T_2 - T_1)}{(T_3 - T_5)} \text{ as } T_4 = T_5$$

$$= 1 - \frac{(T_2 - T_1)}{(T_5 - T_4)} \tag{2}$$

$$= 1 - \frac{\frac{1}{\eta_c}(T_{2'} - T_1)}{\eta_t (T_3 - T_{4'})}$$



(a)



(b)

Fig. 12.21 *(a) Gas Turbine with Regenerator (b) T-s Diagram*

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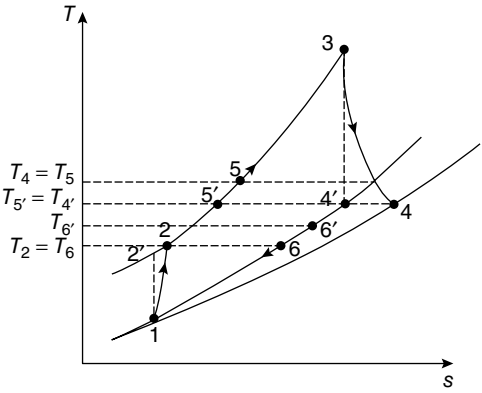


Fig. 12.22 *T-s Diagram with Regenerator*

$$= 1 - \frac{T_1\left(\frac{T_{2'}}{T_1} - 1\right)}{\eta_t \times \eta_c \left(1 - \frac{T_{4'}}{T_3}\right) T_3}$$

$$= 1 - \frac{T_1\left(\beta^{\frac{\gamma-1}{\gamma}} - 1\right)}{\eta_t \times \eta_c \left(1 - \frac{1}{\beta^{\frac{\gamma-1}{\gamma}}}\right) T_R} \text{; as } \frac{T_3}{T_1} = T_R$$

$$= 1 - \frac{\beta^{\frac{\gamma-1}{\gamma}}}{\eta_t \times \eta_c T_R}$$

$$\eta = 1 - \frac{R}{\eta_C \eta_t T_R} \text{; as } R = \beta^{\frac{\gamma-1}{\gamma}} \tag{3}$$

From Equation (3), it is evident that as temperature ratio increases (T_R), thermal efficiency of the cycle increases. Again, decreasing the pressure ratio term (R), thermal efficiency increases.

In an actual cycle, it is not possible to attain the condition $T_5 = T_4$. The actual temperature of air that comes out of the regenerator is lesser than T_5 and is equal to T_5' as shown on the *T-s* diagram in Figure 12.22.

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Hence, cycle efficiency

$$\eta = \frac{(T_3 - T_4) + (T_2 - T_1)}{(T_3 - T_{5'})} \tag{4}$$

Heat exchanger efficiency or effectiveness,

$$\eta = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}}$$

$$\eta = \frac{m_a C_{pa}(T_{5'} - T_2)}{m_a C_{pg}(T_5 - T_2)} = \frac{m_a C_{pa}(T_{5'} - T_2)}{m_a C_{pg}(T_4 - T_2)} \tag{5}$$

where m_a, m_g = mass flow rate of air and gas, respectively, with corresponding specific heats C_{pa} and C_{pg}.

In case, $m_a = m_g$ and $C_{pa} = C_{pg}$, then the above equation reduces to

$$\eta = \frac{(T_{5'} - T_2)}{(T_4 - T_2)} \tag{6}$$

Even though regenerator efficiency increases with increased heat-transfer surface area, the space and pressure loss consideration limits its use.

12.7.3 Intercooling

Another method of improving the plant efficiency is to reduce the work absorbed by the compressor stages, by adopting an intercooler in between the stages. Such an arrangement is shown in Figure 12.23.

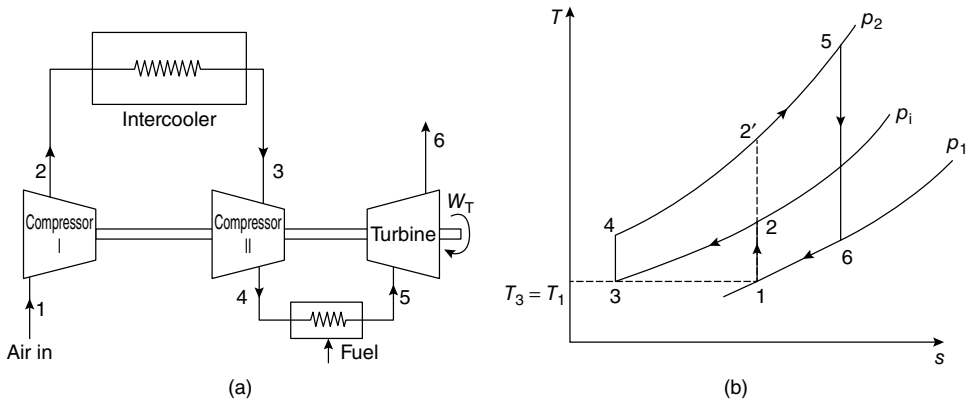


Fig. 12.23 *(a) Gas Turbine with Intercooler and (b) T-s Diagram for Perfect Intercooling*

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For single-stage compression, compressor work input is given by the following equation:

$$W_C = C_p\,(T_2' - T_1) = C_p\,[(T_2 - T_1) + (T_2' - T_2)] \tag{1}$$

For two-stage compression, compressor work input is given by the following equation:

$$W_C = C_p\,[(T_2 - T_1) + (T_4 - T_3)] \tag{2}$$

Comparing the above two equations, it can be noted that in the *T-s* diagram that

$$(T_2' - T_2) > (T_4 - T_3)$$

Hence, work input is reduced in two-stage compression. For perfect intercooling, $T_3 = T_1$.

But in actual practice, intercooling is not perfect, and hence $T_3 > T_1$ as shown in the *T-s* diagram in Figure 12.24.

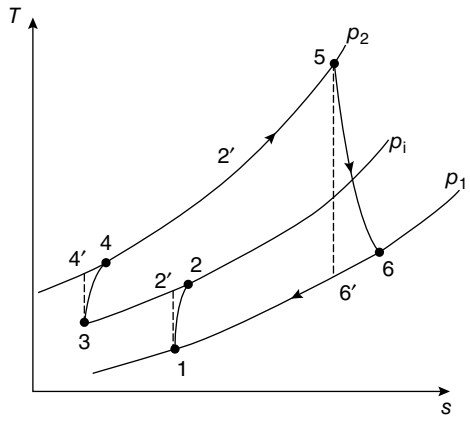


Fig. 12.24 *Gas Turbine with Imperfect Intercooling*

Considering a two stage compression, compressor work is given by

$$W_C = C_p\,(T_2 - T_1) + C_p\,(T_4 - T_3)$$

$$= \frac{C_p}{\eta_{c1}}\left(T_{2'} - T_1\right) + \frac{C_p}{C_2}\left(T_{4'} - T_3\right)$$

$$= \frac{C_p T_1}{\eta_{c1}}\left(\frac{T_{2'}}{T_1} - 1\right) + \frac{C_p T_3}{\eta_{c2}}\left(\frac{T_{4'}}{T_3} - 1\right)$$

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$$W_C = \frac{C_p T_1}{\eta_{c1}}\left(\left(\frac{p_i}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) + \frac{C_p T_3}{\eta_{c2}}\left(\left(\frac{p_2}{p_i}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{3}$$

where η_{C1}, η_{C2} = isentropic efficiency of first and second compressors, respectively,
p_i = intermediate pressure between two stages,

Substituting $\frac{\gamma-1}{\gamma} = x$ and $\frac{T_3}{T_1} = T_R$, we get

$$W_C = \frac{C_p T_1}{\eta_{c1}}\left(\left(\frac{p_i}{p_1}\right)^{x} - 1\right) + \frac{C_p T_R T_1}{\eta_{c2}}\left(\left(\frac{p_2}{p_i}\right)^{x} - 1\right) \tag{4}$$

For minimum compressor work, with pressures p_1 and p_2 and temperatures T_1 and T_3 fixed, we have,

$$\frac{dW_C}{dp_i} = 0$$

$$\frac{d}{dp_i}\frac{C_p T_1}{\eta_{c1}}\left(\frac{p_i}{p_1}\right)^{x} + \frac{d}{dp_i}\frac{C_p T_R T_1}{\eta_{c2}}\left(\frac{p_2}{p_i}\right)^{x} = 0$$

i.e.,
$$\frac{C_p T_1}{\eta_{c1}}\left(\frac{p_i^{x-1}}{p_1^{x}}\right) + \frac{C_p T_R T_1}{\eta_{c2}} p_2^{x}(-x) p_i^{-x-1} = 0$$

i.e.,
$$\frac{x}{\eta_{c1}}\left(\frac{p_i^{x-1}}{p_1^{x}}\right) + \frac{T_R x}{\eta_{c2}} p_2^{x}(-x) p_i^{-x-1} = 0$$

$$\left(\frac{p_i^{x-1}}{p_i^{x}}\right)\frac{1}{\eta_{c1}} = \frac{\left(T_R\ p^{x}\right)}{\eta_{c2}} \times \left(p_i\right)^{-(x+1)}$$

i.e.,
$$\left(p_i^{x-1} p_i^{x+1}\right) = \frac{\eta_{c1}}{\eta_{c2}} \times T_R^{\frac{1}{x}} \times \left(p_1 p_2\right)^{x}$$

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Hence, $$p_i^{2x} = \frac{\eta_{c1}}{\eta_{c2}} \times T_R \times (p_1 p_2)^x$$

$\therefore$ $$p_i = \sqrt{\frac{\eta_{c1}}{\eta_{c2}} \times T_R^{\frac{1}{x}} \times p_1 p_2}$$

i. e., $$p_i = \sqrt{\frac{\eta_{c1}}{\eta_{c2}} \times T_R^{\frac{\gamma-1}{\gamma}} \times p_1 p_2}$$

In other words, for imperfect intercooling,

$$p_i = \sqrt{\frac{\eta_{c1}}{\eta_{c2}} \times T_R^{\frac{\gamma-1}{\gamma}} \times p_1 p_2} \tag{5}$$

For perfect intercooling, $T_R = 1$ and if $\eta_{C1} = \eta_{C2}$, then

$$p_i = \sqrt{p_1 p_2} \tag{6}$$

Substituting this value in Equation (4) above,
For perfect intercooling,

$$W_C = \frac{C_p T_1}{\eta_c}\left(\left(\frac{p_i}{p_1}\right)^x + \left(\frac{p_2}{p_i}\right)^x - 2\right) = \frac{C_p T_1}{\eta_c}\left(\left(\frac{p_2}{p_1}\right)^{x/2} + \left(\frac{p_2}{p_1}\right)^{x/2} - 2\right)$$

$$W_C = \frac{2C_p T_1}{\eta_c}\left(\left(\frac{p_2}{p_1}\right)^{x/2} - 1\right)$$

Thermal efficiency of the turbine is given by the following equation:

$$\eta_{th} = \frac{(T_5 - T_6) - \left[(T_2 - T_1) - (T_4 - T_3)\right]_1}{(T_5 - T_4)}$$

In the above equation as $W_C = (T_2 - T_1) + (T_4 - T_3)$ reduces and turbine work $W_T = (T_5 - T_6)$ remains same, thermal efficiency of the cycle decreases as $(T_5 - T_4)$ = heat addition increases in case of imperfect intercooling, as compared to perfect intercooling.

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Example 12.1

Air is drawn in a gas turbine at 18°C and 1 Bar and leaves the compressor at 5 Bar. Data observed are the following:

Temperature of gases entering the turbine = 678°C.

Pressure loss in combustion chamber = 0.1 Bar.

$\eta_{compressor} = 85\%$ $\gamma = 1.4$ for air

$\eta_{combustion} = 85\%$ $C_p = 1.024$ kJ/kg K for gas

$\eta_{turbine} = 80\%$

Find the (i) quantity of air, if plant develops 1065 kW; (ii) heat supplied per kg of air circulated and (iii) thermal efficiency of cycle.

Solution: The cycle is shown on the *T-s* diagram below.

Consider processes 1 and 2, $\dfrac{T_2}{T_1} = \left(\dfrac{5}{1}\right)^{\frac{0.4}{1.4}} = \left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 1.584$

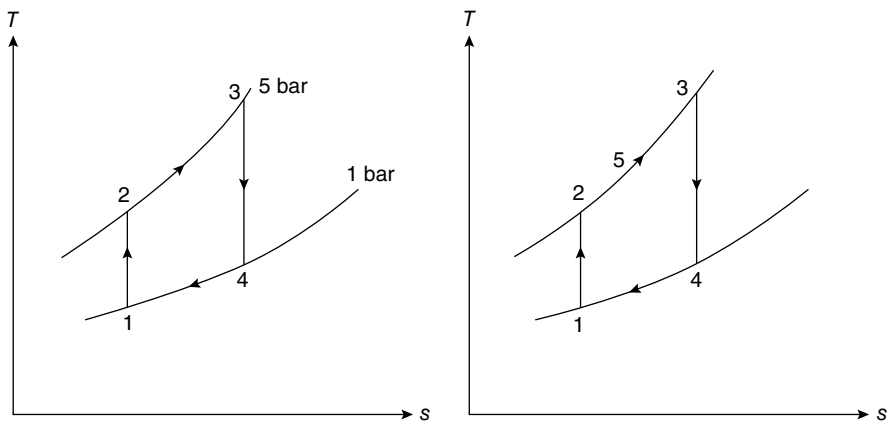
$$T_2 = T_1 \times 1.584 = 300 \times 1.584 = 475.2 \text{ K.}$$

$$\frac{T_3}{T_4} = 5^{\frac{0.4}{1.4}}$$

$$T_4 = \frac{T_3}{5^{\frac{0.4}{1.4}}}$$

$$= \frac{823}{1.584}$$

$\therefore$ $$= 519.57 \text{ K}$$



(a) Without regenerator (b) With regenerator

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1. Efficiency of the turbine without regenerator,

$$\eta = \frac{W_T - W_C}{Q_A}$$

For unit mass of air,

$$= \frac{(T_3 - T_4) + (T_2 - T_1)}{(T_3 - T_2)}$$

$$= \frac{(823 - 519.57) - (475.2 - 300)}{823 - 475.2}$$

$$= 36.87\%$$

2. Efficiency of the turbine with regenerator,

Regenerator efficiency

$$\in = \frac{T_{5'} - T_2}{T_5 - T_2}$$

$$T_5' = \in (T_5 - T_2) + T_2$$

$$= 0.60\ (519.57 - 475.2) + 475.2$$

$$= 26.62 + 475.2$$

$$= 501.822\ \text{K}$$

$$\eta = \frac{(T_3 - T_4) - (T_2 - T_1)}{T_3 - T_{5'}}$$

$$= \frac{(823 - 519.57)(475.2 - 300)}{(823 - 501.822)}$$

$$= 39.93\%$$

Example 12.2

In an open-cycle constant pressure gas turbine unit air enters compressor at 1 bar and 300 K. The pressure of air after compression is 4 bar, the isentropic efficiencies of compressor and turbine are 80 per cent and 85 per cent, respectively. The air fuel ratio is 90:1. Calculate the power developed and thermal efficiency of the cycle if the flow rate of air is 3 kg/s. Take C_p = 1.005 kJ/kgK and $\gamma = 1.4$ for air and gases $CV = 42{,}000$ kJ/kg

Solution:

(i) Power developed:

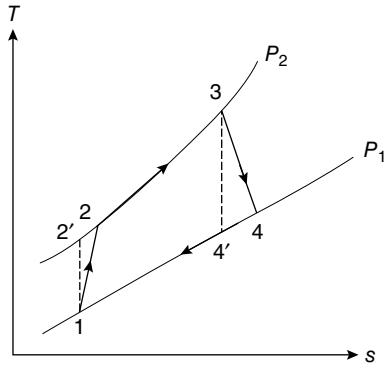
Temperature at the end of isentropic compression is given by

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$$\frac{T_{2'}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2' = (300)\left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2' = 445.8 \text{ K}$$



Actual temperature rise during compression is

$$T_2 - T_1 = \frac{T_{2'} - T_1}{h_C} = \frac{445.8 - 293}{0.8}$$

$$= 191$$

$$T_2 = T_1 + 178 = 300 + 191$$

$$T_2 = 491 \text{ K}$$

Heat supplied by the fuel = heat taken by the gases:

$$m_f \, CV = (m_a + m_f) \, C_p \, (T_3 - T_2)$$

$$CV = \left(\frac{m_a}{m_f} + 1\right) \times C_p \times (T_3 - T_2)$$

$$42{,}000 = (90 + 1) \times 1(T_3 - 491)$$

$$T_3 = \frac{42{,}000}{91 \times 1} + 491$$

$$T_3 = 952.54 \text{ K}$$

The work required to drive the compressor is

$$W_C = C_P \, (T_2 - T_1)$$

$$W_C = 1.005 \, (491 - 300)$$

$$W_C = 191.96 \text{ kJ/kg of air per sec.}$$

Temperature at the end of isentropic expansion is

$$T_4' = \left(\frac{T_3}{\beta^{\frac{\gamma-1}{\gamma}}}\right) = \left(\frac{952.54}{4^{\frac{1.4-1}{1.4}}}\right) = 641.01 \text{ K}$$

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$\therefore$ Actual temperature drop $= T_3 - T_4 = \eta_t\,(T_3 - T_4')$

$= 0.85\,(952.54 - 641.01) = 264.8°C$

$T_4 = T_3 - 264.8 = 952.54 - 264.8$

$T_4 = 687.74$ K

Total turbine work is

$$W_t = m\,C_p\,(T_3 - T_4)$$

$$= \left(\frac{91}{90}\right) \times 1 \times (952.54 - 687.74)$$

$W_t = 267.74$ kJ/kg of air per sec

$\therefore$ Net work output $= W_T - W_C = 267.74 - 191.96$

$= 75.78$ kW/kg of air

Net power developed by the turbine $= 75.78 \times 3 = 227.34$ kW

(ii) Thermal efficiency

Heat supplied per kilogram of air $= \frac{1}{90} \times 42000 = 466.67$ kJ/kg of air

$$\text{Thermal efficiency, } \eta_t = \frac{75.78}{466.67} = 0.1624 = 16.24\%$$

Example 12.3

In a regenerative gas turbine cycle, air enters the compressor at 1 bar 15°C. Pressure ratio = 6. The isentropic efficiencies of compressor and turbine are 0.8 and 0.85, respectively. The maximum temperature in the cycle is 800°C. The regenerator efficiency = 0.78, Assume $C_p = 1.1$ kJ/kgK and $\gamma = 1.32$ for the combustion products and find the cycle efficiency.

Solution: The gas turbine cycle is shown as the *T-s* diagram.

Pressure ratio $\beta = \frac{p_2}{p_1} = 6$

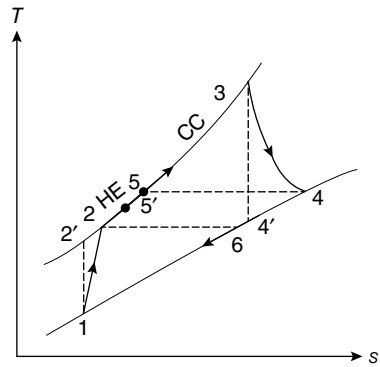
$p_2 = 6$ bar

For the actual cycle, temperature of air at the compressor exit is given by

$$T_2 = T_1 + \frac{T_1}{\eta_C}\left(\left(\frac{p_{2'}}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)$$

$$= 288 + \frac{288}{0.8}\left(\left(\frac{p_{2'}}{p_1}\right)^{\frac{0.4}{1.4}} - 1\right)$$

$= 528.66$ K



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Compressor work is given by

$$W_C = C_P (T_2 - T_1) = 1.005 (528.66 - 288)$$
$$= 241.86 \text{ kJ/kg}$$

Considering processes 3 and 4, the actual temperature of gases at turbine exit is given by

$$T_4 = T_3 - \eta_t T_3 \left[\frac{1}{(\beta)^{\frac{\gamma-1}{\gamma}}} \right] = 1073 - 0.85 \times 1073_3 \left[\frac{1}{(6)^{\frac{0.4}{1.4}}} \right]$$

$$= 751.66 \text{ K}$$

Turbine work is given by

$$W_T = m_g C_{pg} (T_3 - T_4); \qquad \text{(ignoring fuel mass, } m_a = m_g)$$
$$= 1 \times 1.1 (1073 - 751.66)$$
$$= 353.47 \text{ kJ/kg}$$

Net work output

$$= W_T - W_C = 353.47 - 241.86$$
$$= 111.6 \text{ kJ/kg}$$

Now, efficiency of the regenerator is given by

$$\in = \frac{T_{5'} - T_2}{T_5 - T_2}; \; T_5 = T_4 \text{ for ideal conditions}$$

$$0.78 = \frac{T_{5'} - 528.66}{751.66 - 528.66}$$

$$T_5' = 528.66 + 0.78 (751.66 - 528.66)$$
$$= 702.8 \text{ K}$$

Thermal efficiency of the cycle, assuming $m_a = m_g$ and $C_{pa} = C_{pg}$,

$$\eta = \frac{(T_3 - T_4)(T_2 - T_1)}{(T_3 - T_{5'})}$$

$$= \frac{(1073 - 751.66) + (528.66 - 288)}{(1073 - 702.8)}$$

$$= \frac{321.34 - 230.68}{370.2}$$
$$= 24.49\%$$

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Considering C_{pg} and C_{pa}

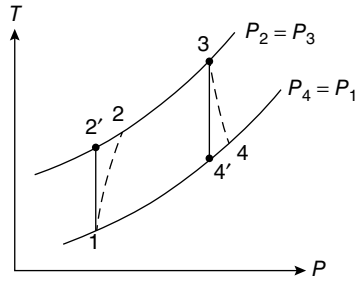
$$\eta = \frac{\oint W}{1.1(1073 - 702.8)} = \frac{111.6}{407.22}$$
$$= 27.4\%$$

Example 12.4

In an open-cycle gas turbine plant, air enters the compressor at 1 bar and 27°C. The pressure after compression is 4 bar. The isentropic efficiencies of the turbine and the compressor are 85 per cent and 80 per cent, respectively. Air fuel ratio is 80:1. The calorific value of the fuel used is 42,000 kJ/kg and mass flow rate of air is 2.5 kg/s. Determine the power output from the plant and the cycle efficiency. Assume that C_p and γ to are same for both air and products of combustion.

Solution:

Given data: $p_1 = 1$ bar, $T_1 = 300$ K
$p_2 = p_3 = 4$ bar, $\eta_T = 0.85$, $\eta_c = 0.80$
$m_a = 2.5$ kg/s; $m_a/m_f = 80$
$CV = 42{,}000$ kJ/kg
$C_p = 1.005$ kJ/kgK (assumed)
$\gamma = 1.4$ (assumed)



For processes 1 and 2′,

$$\frac{T_{2'}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2' = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= 300\left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2' = 446 \text{ K}$$

Compressor efficiency $\eta_c = \dfrac{T_{2'} - T_1}{T_2 - T_1}$

$$T_2 = T_1 + \frac{T_2^1 - T_1}{h_c}$$

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$$\therefore \qquad T_2 = 300 + \frac{446 - 300}{0.80}$$

$$T_2 = 482.5 \text{ K}$$

For processes 3 and 4′,

$$\frac{T_{4'}}{T_3} = \left(\frac{p_{4'}}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{4'} = T_3\left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \qquad (1)$$

Heat supplied per unit time:

$$Q_{2-3} = m_f CV$$

$$= \frac{2.5}{80} \times 42000$$

$$= 1312.5 \text{ kJ/s}$$

$$\text{But } Q_{2-3} = (m_a + m_f)h_3 - m_a h_2$$

$$h_3 = \frac{Q_{2-3} + m_a h_2}{m_a + m_f}$$

$$T_3 = \frac{Q_{2-3} + m_a C_p h_2}{C_p\left(m_a + m_f\right)}$$

$$= \frac{1312.5 + 2.5 \times 1.005 \times 482.5}{1.005\left(1 + \frac{2.5}{80}\right)}$$

$$T_3 = 992.3 \text{ K}$$

From Equation (1),

$$T_{4'} = T_3\left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = 992.3\left(\frac{1}{4}\right)^{\frac{0.4}{1.4}}$$

$$= 667.5 \text{ K}$$

Turbine efficiency is given by

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4'}}$$

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$$\therefore T_4 = T_3 - \eta_T(T_3 - T_4')$$

$$T_4 = 992.3 - 0.85\ (992.3 - 667.5)$$

$$T_4 = 716.22\ \text{K}$$

Compressor work per unit time $= W_C = m_a C_p\ (T_2 - T_1)$

$$= 2.5 \times 1.005\ (482.5 - 300) = 458.53\ \text{kJ/s}$$

$$= 458.53\ \text{kW}$$

Turbine work per unit time $= W_T = (m_a + m_f)\ C_p\ (T_3 - T_4)$

$$W_T = (2.5 + 2.5/80) \times 1.005 \times (992.3 - 716.22)$$

$$= 702.32\ \text{kW}$$

Net work output $\oint W = 702.32 - 458.53$

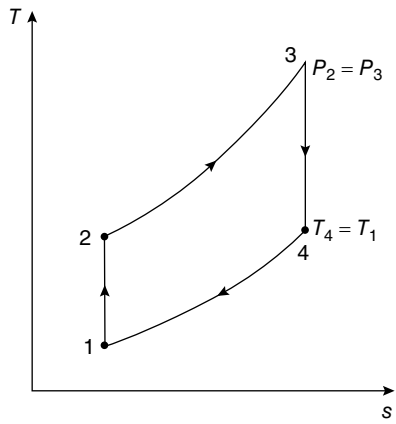
$$= 243.79\ \text{kW}$$

Cycle efficiency $\eta_{cycle} = \left(\dfrac{\oint W}{Q_{2-3}}\right) = \dfrac{243.79}{1312}$

$$= 18.58\%$$

Example 12.5

A gas turbine power plant operates on ideal Brayton cycle. The minimum and maximum cycle temperature are T_1 = 300 K and T_3 = 800 K, respectively. Find the value of the optimum pressure ratio for maximum specific work output and the cycle efficiency for this condition. Is it possible to improve this cycle efficiency by including regenerator? Substantiate.



Solution: Given data

$$T_{min} = T_1 = 300\ \text{K}$$

$$T_{max} = T_3 = 800\ \text{K}$$

Optimum pressure ratio for maximum output is given by

$$\beta_{opt} = \frac{p_2}{p_1} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

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$$= \left(\frac{800}{300}\right)^{\frac{1.4}{2\times 0.4}} = 5.565$$

$$\text{Cycle efficiency, } \eta = 1 - \frac{1}{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}} = 1 - \sqrt{\frac{T_1}{T_3}}$$

$$\eta = 1 - \sqrt{\frac{300}{800}} = 38.76\%$$

Condition for maximum work output is satisfied for

$$\frac{T_3}{T_1} = \left(\frac{p_2}{p_1}\right)^{2\left(\frac{\gamma-1}{\gamma}\right)}$$

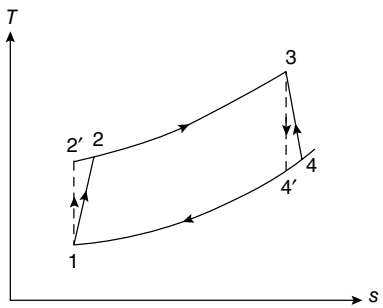
But $$\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_3}{T_1} = \left(\frac{T_2}{T_1}\right)^2 = \left(\frac{T_3}{T_4}\right)^2$$

$$T_2 = T_4 = \sqrt{T_1 T_3}$$

As $T_2 = T_4$, regenerator is not required.

Example 12.6



A simple gas turbine plant operating on the Brayton cycle has air entering the compressor at 10 kPa and 27°C. The pressure ratio = 9 and maximum cycle temperature = 727°C. What will be the percentage change in cycle efficiency and net work output if the expansion in the turbine is divided into two stages each of pressure ratio 3, with intermediate reheating to 727°C? Assume compression and expansion are ideal isentropic.

Solution:

(i) Single-stage expansion

Given that

$p_2 = 1$ bar

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$T_1 = 300$ K

$= 9$

$T_3 = T_5 = 1000$ K

For the processes 1 and 2,

$$= \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9}{1}\right)^{\frac{0.4}{1.4}} = 1.875$$

$$\therefore \quad T_2 = T_1 \times 1.875 = 300 \times 1.875 = 562.5 \text{ K}$$

For the processes 3 and 4,

$$\frac{T_3}{T_4} = \frac{T_2}{T_1}$$

$$T_4 = \frac{T_3 T_1}{T_2} = \frac{1000 \times 300}{562.5} = 533.3 \text{ K}$$

Compression work, $W_C = C_P(T_2 - T_1)$

$= 1.005\ (562.5 - 300)$

$W_C = 263.8$ kJ/kg

Expansion work, $W_T = C_P(T_3 - T_4)$

$= 1.005\ (1000 - 533.3)$

$W_T = 469.03$ kJ/kg

Net work output $\oint W_1 = W_T - W_C$

$= 469.03 - 263.81 = 205.22$ kJ/kg

Heat supplied $Q_S = C_P(T_3 - T_2)$

$= 1.005\ (1000 - 562.5)$

$= 439.68$ kJ/kg

Cycle efficiency $(\eta)_1 = \left(\frac{\oint W_1}{Q_S}\right)$

$$= \frac{205.22}{439.68} = 0.4667 = 46.67\%$$

(ii) Expansion in two stages with pressure ratio in each stage = 3

$$= (3)^{0.286} = 1.369$$

$$T_{4a} = \frac{1000}{1.369} = 730.4 \text{ K}$$

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As the pressure ratio in the second stage is the same and air is reheated to 1000 K,

$$T_6 = T_{4a} = 730.4 \text{ K}$$

Work of expansion in each stage will be the same.
∴ Total work of expansion

$$\begin{aligned} W_T &= 2C_P (T_3 - T_{4a}) \\ &= 2 \times 1.005 (1000 - 730.4) \\ &= 541.89 \text{ kJ/kg} \end{aligned}$$

Work of compression will remain the same, that is, 263.81 kJ/kg.

Net work output

$$\begin{aligned} \oint W_2 &= W_T - W_C \\ &= 541.89 - 263.81 \\ &= 278.08 \text{ kJ/kg} \end{aligned}$$

Heat supplied

$$\begin{aligned} Q_{S2} &= \text{heat supplied in combustion chamber} \\ &\quad + \text{heat supplied in the reheater} \\ &= C_P(T_3 - T_2) + C_P(T_5 - T_{4a}) \\ &= 1.005 (1000 - 562.5) + 1.005 (1000 - 730.4) \\ &= 439.68 + 270.94 \\ &= 710.62 \text{ kJ/kg} \end{aligned}$$

Cycle efficiency

$$\eta_2 = \left(\frac{\oint W_2}{Q_{S2}} \right) = \frac{278.08}{710.62} = 0.3913 = 39.13\%$$

Percentage change in efficiency

$$\begin{aligned} \eta &= \frac{\eta_2 - \eta_1}{\eta_1} = \frac{39.13 - 46.67}{46.67} \\ &= -16.15\% \end{aligned}$$

Percentage change in work output

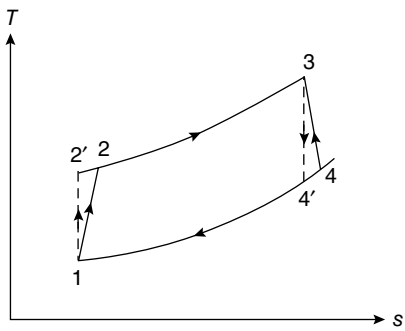
$$= \left(\frac{\oint W_2 - \oint W_1}{\oint W_2} \right)$$

$$= \frac{278.08 - 205.22}{205.22} = 0.355 = 35.5\%$$

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Example 12.7

A gas turbine power plant operates on the simple Brayton cycle with air as the working fluid and delivers 32 MW of power. The minimum and maximum temperatures in the cycle are 310 K and 900 K, respectively, and the pressure of air at the compressor exit is eight times the value at the compressor inlet. Assuming an isentropic efficiency of 80 per cent for the compressor and 86 per cent for the turbine, determine the mass flow rate of air through the cycle.



Solution: Given data

$$T_{max} = T_3 = 900 \text{ K}$$

$$T_{max} = T_1 = 310 \text{ K}$$

$$r_P = 8, = 0.8, = 0.86$$

For processes 1and 2, $T_2' = T_1 \, r_P^{\frac{\gamma-1}{\gamma}}$

$$T_2' = 310 \times 8^{\frac{0.4}{1.4}} = 561.55 \text{ K}$$

Considering compressor efficiency,

$$= \frac{T_{2'} - T_1}{T_2 - T_1}$$

$$\therefore = T_1 + \frac{T_2 - T_1}{\eta_C}$$

$$= 561.55 + \frac{561.55 - 310}{0.8}$$

$$= 624.44 \text{ K}$$

For process 3 and 4,

$$T_4' = \frac{T_3}{r_P^{\frac{\gamma-1}{\gamma}}}$$

$$= \frac{90}{8^{(0.4/1.4)}}$$

$$= 496.84 \text{ K}$$

Considering turbine efficiency,

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4'}}$$

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$$\therefore T_4 = T_3 - \eta_T (T_3 - T_{4'})$$
$$= 900 - 0.86\ (900 - 496.84)$$
$$= 553.3\ \text{K}$$

Compressor work is given by

$$W_C = C_P(T_2 - T_1) = 1.005\ (624.44 - 310)$$
$$= 316.01\ \text{kJ/kg}$$

Turbine work is given by

$$W_T = C_P(T_3 - T_4)$$
$$= 1.005\ (900 - 553.3)$$
$$= 348.43\ \text{kJ/kg}$$

Net work,

$$\oint W = W_T - W_C = 32.42\ \text{kJ/kg}$$

$$\text{Power } P = m \times \oint W$$

$$\therefore m = \left(\frac{P}{\oint W} \right)$$

$$m = \frac{32 \times 10^3}{32.42} = 987.05\ \text{kg/s}$$

Example 12.8

A gas turbine plant draws in air at 1.013 bar, 10°C and has a pressure ratio of 5.5. The maximum temperature in the cycle is limited to 750°C. Compression is conducted in an uncooled rotary compressor having an isentropic efficiency of 82 per cent and expansion takes place in a turbine with an isentropic efficiency of 85 per cent. A heat exchanger with an efficiency of 70 per cent is fitted between the compressor outlet and combustion chamber. For an air flow of 40 kg/s, find the (i) overall efficiency of cycle, (ii) turbine output, (iii) air–fuel ratio if the calorific value of fuel used is 45.22 MJ/kg.

Solution: All the processes are shown on the *T-s* diagram below.

Considering processes 1 and 2,

$$T_2' = T_1\ r_P^{\frac{\sqrt{r}-1}{\sqrt{r}}} = 283 \times (5.5)^{\frac{0.4}{1.4}} = 460\ \text{K}$$

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$$\eta_c = \frac{T_{2'} - T_1}{T_2 - T_1}$$

$$0.82 = \frac{460 - 283}{T_{2'} - 283}$$

$$= 498.85 \text{ K}$$

Considering expansion in turbine,

$$T_4 = \frac{T_3}{r_P^{\frac{\gamma-1}{\gamma}}} = \frac{1023}{(5.5)^{\frac{0.4}{1.4}}} = 629.3 \text{ K}$$

Considering turbine efficiency,

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4'}}$$

$$0.85 = \frac{1023 - T_4}{1023 - 629.3}$$

$$T_4 = 688.36 \text{ K}$$

Considering heat exchanger efficiency,

$$\eta_{HE} = \frac{T_5 - T_2}{T_4 - T_2}$$

$$0.70 = \frac{T_5 - 498.85}{688.36 - 498.85}$$

$$\therefore T_5 = 640.98 \text{ K}$$

(i) Overall efficiency of cycle
Cycle efficiency is given by

$$\eta = \frac{W_T - W_C}{Q_A} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_5)}$$

$$= \frac{(1023 - 688.36) + (498.85 - 688.36)}{(1023 - 640.98)}$$

$$= 31\%$$

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(ii) Turbine output

$$\text{Net work done, } \oint W = W_T - W_C = 4751.6 \text{ kJ/kg}$$

$$\text{Output power} = 1 \times 4751.6$$

$$= 4751.6 \text{ kW}$$

(iii) Air–fuel ratio

Making heat balance in the combustion chamber,

$$m_f \times CV = (m_a + m_f)\, C_{pa}$$

$$m_f \times 45.22 \times 10^3 = (40 + m_f) \times 1.005 \times (1023 - 640.98)$$

$$m_f \times 117.78 = 40 + m_f$$

$$m_f = \frac{40}{116.78}$$

$$\therefore \quad \frac{m_a}{m_f} = 116.78$$

12.8 FREE-PISTON ENGINE PLANT – LIMITATION, APPLICATION AND ADVANTAGES

Free-piston engines are typically diesel powered, opposed piston engines with mechanical synchronization of the two pistons. The synchronization mechanism did, in addition to evening out possible force differences, also drive the accessories such as fuel injection pump, oil pump and water pump. Free-piston gas generators were considered for use in large-scale power plants, and at a later stage for vehicle propulsion in the late 1940s. General Motors and Ford Motor Company had working prototypes of these engines aimed for vehicle propulsion. Larger free-piston gas generators were installed in a number of stationary power plants and in some marine installations.

In a free-piston engine, the piston is 'free' and its motion is not restricted by the position of a rotating crankshaft, unlike in a conventional engine. The motion is determined by the interaction between the gas and load forces acting upon it. Because of this, a free-piston engine offers (i) variable stroke length, (ii) the need for active control of piston motion, (iii) potential reductions in frictional losses and (iv) possibilities to optimize engine operation by varying compression ratio.

R. P. Pescara [2] was the first to invent a free-piston engine with his patent dating from 1928. Free-piston engines may be classified as single, dual, opposed piston and gas generators. A brief account of these engines is discussed further.

1. *Single piston*

A single piston free-piston engine is shown in Figure 12.25. This engine essentially consists of three parts, namely a combustion cylinder, a load devic, and a rebound device to store the

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energy required to compress the next cylinder charge. The hydraulic cylinder serves the purpose of both load and rebound device. The rebound device is used to accurately control the amount of energy put into the compression process, and hence to regulate the compression ratio and stroke length. The system is simple in design but offers high controllability compared to the other free-piston engine configurations.

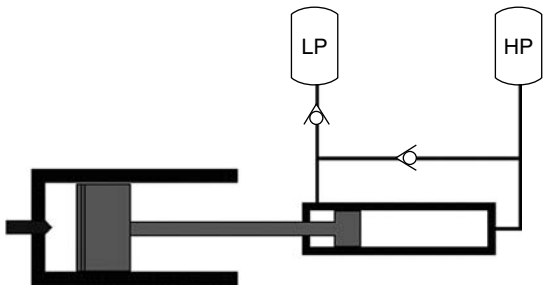


Fig. 12.25 *Single-Piston Hydraulic Free-Piston Engine [3]*

2. *Dual piston*

Figure 12.26 shows a hydraulic dual piston free-piston engine [4]. The dual piston engine configuration eliminates the need for a rebound device, as the (at any time) working piston provides the work to drive the compression process in the other cylinder. This allows a simple and more compact device with higher power to weight ratio.

The major drawback of this engine is that it is difficult to control of piston motion, in particular stroke length and compression ratio. This is because the combustion process in one cylinder

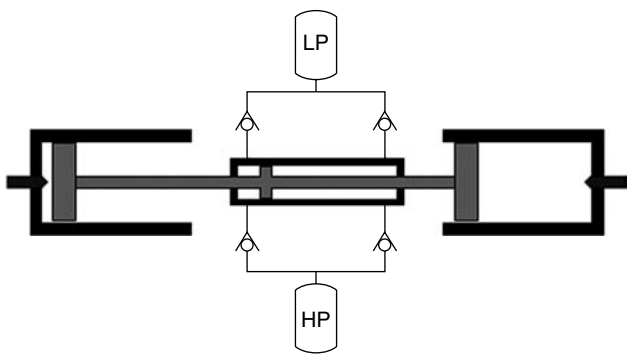


Fig. 12.26 *A Hydraulic Dual Piston Free-Piston Engine [4]*

drives the compression in the other, and small variations in the combustion will have high influence on the next compression. Experimental work with dual piston engines has reported high sensitivity to load nuances and high cycle-to-cycle variations [5, 6].

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3. *Opposed piston*

Figure 12.27 shows an opposed piston free-piston engine, with a mechanical piston synchronization mechanism. An opposed piston free-piston engine consists of two single piston units connected with a common combustion chamber. Each piston requires a rebound device, and a load device may be coupled to one or both of the pistons.

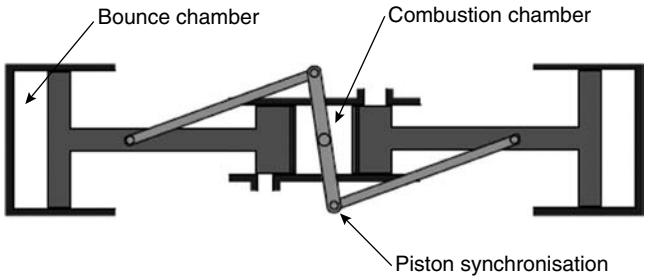


Fig. 12.27 *A Hydraulic Dual Piston Free-Piston Engine*

In the earlier days, these engines were used as air compressors and later as gas generators in large-scale plants, using a number of units feeding a single-power turbine.

The advantages of the opposed piston are the following:

(i) Perfectly balanced and vibration-free design.

(ii) Reduced heat transfer losses due to the opposed piston cylinder as cylinder head is eliminated

(iii) Allows uniflow scavenging resulting in high scavenging efficiency.

The disadvantages of the opposed piston are the following:

(i) Piston synchronization mechanism is a must.

(ii) Usage of dual set of the main components makes the engine complicated and bulky.

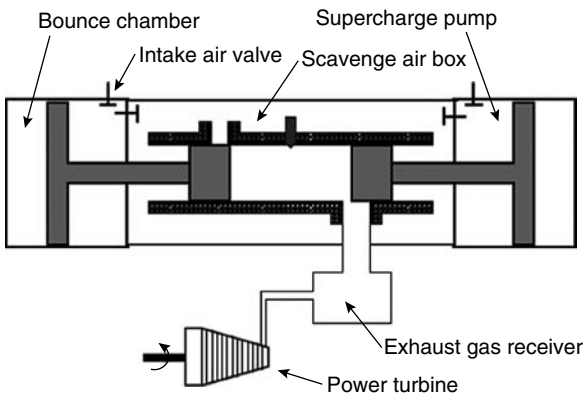


Fig. 12.28 *An Opposed Piston Free-Piston Gas Generator Plant*

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4. *Gas generators*

Figure 12.28 illustrates an opposed piston free-piston gas generator plant. Free-piston gas generators use free-piston engines that feed hot gases to a power turbine. The engine is used to supercharge the intake air, and the entire output work is utilized by the power turbine. In the mid-twentieth century, free-piston gas generators were used in some large-scale marine and stationary power plants. Free piston engines are started by rapidly introducing compressed air into the bounce chambers as shown in Figure 12.28. These engines were highly supercharged and operated on higher mean effective pressures than conventional diesel engines [3].

The most important advantages of free-piston gas generators are the following:

(i) Compared to a conventional gas turbine, the free-piston gas generator offers high compression and pressure ratios.

(ii) As the work needed to compress the intake air is already extracted from the gas when supplied to the power turbine, the gas fed to the turbine is at a lower temperature, which reduces the materials requirements and allows the turbine to be placed further away from the combustor without extensive heat transfer losses.

(iii) Low fuel quality requirements,

(iv) Vibration-free design,

(v) Good dynamic response and low turbine material requirements due to lower inlet temperatures.

(vi) The fuel economy of the free-piston gas generator power plant was competitive with conventional diesel engines and 80–100 per cent better than conventional gas turbine plants [3]. The specific weight was somewhere between the low specific weight of the gas turbine and the high specific weight of the conventional diesel engine.

(vii) Free-piston gas generator efficiency is higher compared to the conventional gas turbine due to the higher compression pressures. Free-piston engines could achieve an end-of-compression pressure of 100 times the atmospheric pressure as against six in the simple-type gas turbine [7].

Some of the limitations of these engines are the following:

(i) High development efforts were put into both conventional diesel engines and gas turbine technology, giving rapid performance increases.

(ii) Due to the difficulty in matching of a pulsating-flow compressor with a continuous-flow turbine, part-load efficiency is lower. Due to this, the gas generator is limited to constant power applications [8, 9].

(iii) High failure rates and low lifetime and availability were reported, related to high pressure and temperature operation [9].

(iv) The engines did not provide large advantages in weight or fuel economy compared to conventional engines and could not compete with the power to weight ratio of the gas turbine.

With the maturing of conventional gas turbine technology, development of the free-piston gas generator was largely abandoned in the early 1960s. [8, 9, 10]

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12.9 COMBINED CYCLE POWER PLANTS AND COMPARISON

Simple cycle gas turbine (GTs) plants using natural gas and fuel oil operate at around 33 percent and 25 per cent efficiency, respectively, major part of the heat being wasted as thermal energy in the hot exhaust gases.

Combining of multiple thermodynamic cycles to generate power, overall plant efficiency can be increased up to 60 per cent by using a heat recovery steam generator (HRSG). The HSRG captures heat from high-temperature exhaust gases to generate steam, which is then supplied to a steam turbine to generate additional electric power. The cycle working under this principle is known as combined cycle. In most of the instance a combined cycle power plant utilizes gas turbines in conjunction with a steam turbine and is called a combined cycle gas turbine (CCGT) plant. Different configurations of CCGT power plants are in use where each GT has its own associated HRSG, and multiple HRSGs supplying steam to one or more steam turbines. For example, in a plant with a 3×1 configuration, three GT/HRSG trains supply to one steam turbine. Similarly, there can be 1×1, 2×1 or 4×1 arrangements also. The steam turbine capacity is decided to match the number and capacity of supplying GTs/HRSGs. These types of power plants are being installed in increasing numbers round the world where there is access to substantial quantities of natural gas.

12.9.1 Combined Cycle Principles of Operation

The HRSG is basically a heat exchanger popularly known as a boiler, or comprises a series of heat exchangers. It generates steam for the steam turbine exchanging heat from the hot exhaust gas flow from a gas turbine through heat exchanger tube banks. The tubes are arranged in sections, or modules also known as economizers, evaporators, superheater/reheater and preheaters. The fluid circulation in HRSG could be either natural or forced circulation using pumps. Saturated steam from the steam drums or once-through system is passed through superheater or reheater tubes to superheat the steam. The superheated steam produced by the HRSG is supplied to the steam turbine where it expands through the turbine blades, imparting rotation to the turbine shaft. The energy delivered to the generator coupled to the drive shaft is converted into electricity. After exiting the steam turbine, the steam condenses in a condenser and fed back to the HRSG. Figure 12.29 shows a schematic diagram of a combined cycle gas turbine plant.

12.9.1.1 CCGT Design Considerations

Depending on the exhaust gas characteristics HSRGs are designed and configurations, steam requirements, etc., are decided later. Due to the high temperature of exhaust gases available at the gas turbine exit (600°C), GTs are designed to produce steam at multiple pressure levels (high-pressure steam in a large CCGT plant can reach up to 110 bar) to optimize energy recovery. Generally, three sets of heat exchanger modules – one each for high pressure (HP) steam, intermediate pressure (IP) steam, and low pressure (LP) steam – are used for this purpose.

The HRSGs present operational constraints on the CCGT power plant owing to their location, directly downstream of the gas turbines. Due to the changes in temperature and pressure of the exhaust gases thermal and mechanical stresses are set up.

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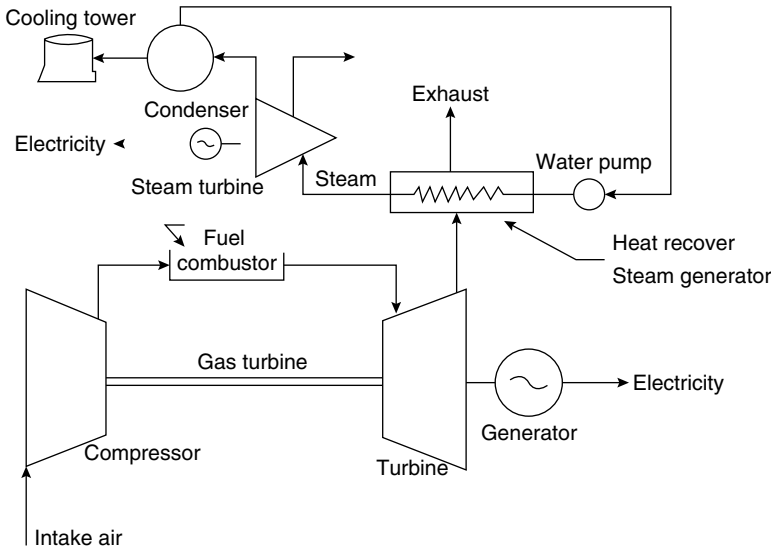


Fig. 12.29 *Combined Cycle Power Plant*

When CCGT power plants are operated under fluctuating load conditions, characterized by frequent starts up and shut downs or when operating under part-load conditions, thermal stresses developed could cause damage to some components of the HRSG. The HP steam drum and superheater headers are subjected to the highest exhaust gas temperatures, and hence are more prone to reduced mechanical life. Some important design and operating considerations are the following:

(i) Temperatures of the gas and steam that the module materials can withstand

(ii) Mechanical stability for turbulent exhaust flow

(iii) Corrosion of HRSG tubes

(iv) Steam pressures to decide drum thickness

12.9.2 Coupled Cycle – GT–ST Plant Operation

In this system, an open-circuit gas turbine has a compressor, a combustor and a turbine. For this type of cycle, the input temperature to turbine and the output temperature of flue gases are very high. This high-temperature flue gases have heat energy high enough to provide heat for a second cycle that uses steam as the working medium, that is, thermal power plant. Figure 12.30 shows the working principle of combined cycle gas turbine (CCGT) plant. Different components of the plant are discussed further in detail.

1. *Air Inlet system*

Air is drawn though the large air inlet section where it is cleaned, cooled and controlled. Majority of all heavy-duty gas turbines are designed to operate under a wide variety of climatic and environmental conditions due to inlet air filtration systems. These filtration systems are specifically designed to suit the plant location. Under normal operating conditions, the inlet

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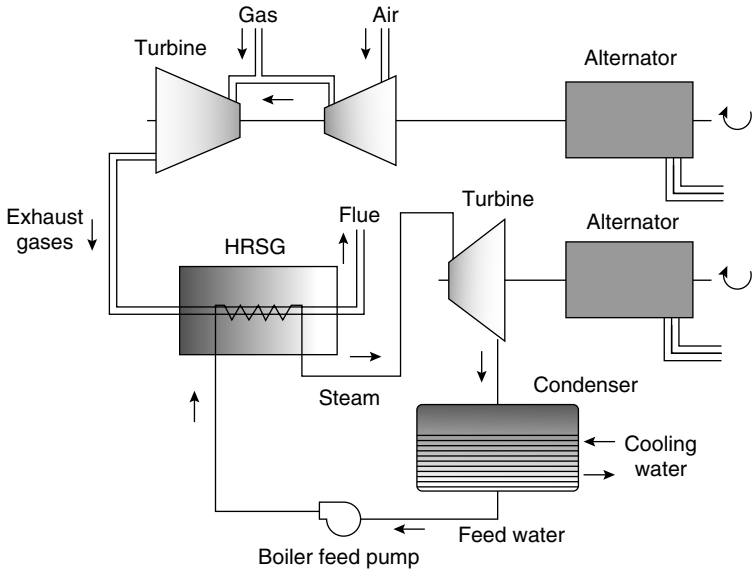


Fig. 12.30 *Working Principle of Combined Cycle Gas Turbine Plant*

system has the capability to process the air by removing contaminants to levels below those that are harmful to the compressor and turbine. In general, the incoming air may have contaminants in solid, liquid and gaseous states in addition to corrosive components. *Gaseous contaminants include* ammonia, chlorine, hydrocarbon gases, sulfur in the form of H_2S, SO_2, discharge from oil cooler vents, etc. Some of the liquid contaminants include chloride salts dissolved in water (sodium, potassium), nitrates, sulfates and hydrocarbons, etc. *Solid contaminants may include* sand, alumina and silica, rust, dust particles, alumina and silica, calcium sulfate, ammonia compounds from fertilizer and animal feed operations, airborne seeds, etc. In addition to these, some *corrosive agents* such as chlorides, nitrates and sulfates may also deposit on compressor blades inducing stress corrosion attack and/or cause corrosion pitting. Alkali metals such as sodium and potassium may also combine with sulfur to form a highly corrosive agent that may attack portions of the hot gas path. These contaminants are removed by passing through various types of filters. Gas phase contaminants such as ammonia or sulfur that cannot be removed by filtration are removed based on special methods.

2. *Turbine system*

The purified air is then compressed and mixed with natural gas and ignited in the combustion chamber after injecting the fuel. The high-pressure gas stream generated in the combustion chamber expands in the turbine and spins the turbine rotor and a generator, producing electricity. Heat of the gas turbine's exhaust is further passed through the heat recovery steam generator (HRSG), where live steam at temperature between 420°C and 580°C is generated, which is used as a working fluid in the secondary circuit.

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3. *Heat recovery steam generator*

In heat recovery steam generator (HRSG), heat exchange takes place between the highly purified water flowing in tubes and the hot flue gases surrounding them, generating steam. As explained earlier, steam expands in the turbine to run the turbine rotor and hence a coupled generator, to produce electricity. The hot gases leave the HRSG at around 140°C before being discharged into the atmosphere. The steam condensing and water system in this circuit are similar to that in a typical thermal power plant. The HRSG takes more time to warm up from cold conditions compared to hot conditions.

12.9.2.1 Configuration of CCGT Plants

The combined cycle system may have *single-shaft* and *multi-shaft configurations*. A single-shaft system consists of a gas turbine, a steam turbine, a generator and a heat recovery steam generator (HRSG). The gas turbine and steam turbine are coupled to the single generator on a single shaft. In a multi-shaft system, one or more gas turbine-generators and HRSGs are used to supply steam through a common header to a separate single-steam turbine generator. Overall investment on a multi-shaft system is about 5 per cent higher compared to a single-shaft system. The major disadvantage of multi-shaft system is that the number of steam turbines, condensers and condensate systems, cooling towers and circulating water systems also proportionately increases to match the number of gas turbines.

12.9.2.2 Efficiency of CCGT Plant

Roughly, the steam turbine cycle produces *one-third of the power* and gas turbine cycle produces *two-thirds of the power output* of the CCGT. By combining both gas and steam cycles, high input temperatures and low output temperatures can be achieved. The efficiency of the cycles adds because they are powered by the same fuel source. To increase the power system efficiency, it is necessary that the HRSG be optimized, which serves as the critical link between the gas turbine cycle and the steam turbine cycle with the objective of increasing the steam turbine output.

Overall efficiency of the combined cycle power plant depends on the performance of HRSG. There are instances of electric efficiency of a new combined cycle power station reaching 58 percent at continuous output. In addition, combined cycle units may be utilized to deliver low temperature heat energy for industrial processes, district heating and other uses. This is called cogeneration, and such power plants are of then referred to as a combined heat and power (CHP) plant.

The efficiency of CCGT is increased by supplementary firing and blade cooling. A gas turbine cooling air (TCA) cooler, with a heat exchanger using feed water, and a fuel gas heater (FGH) were also used to further enhance the efficiency of the plant in Kawasaki Thermal Power Station. This minimized the loss of heat energy allowing the plant to achieve higher thermal efficiency, outperforming conventional combined cycle plants. In this plant, supplementary firing was arranged at HRSG and in gas turbine a part of the compressed air flow bypasses to cool the turbine blades.

12.9.2.3 Fuels

The turbines used in combined cycle plants are commonly fuelled with natural gas, and it is more versatile than coal or oil and can be used in 90 per cent of energy applications. Combined cycle plants are usually powered by natural gas, although fuel oil, synthesis gas or other fuels can be used.

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12.9.2.4 Emissions Control

In order to control the emission, different tools and techniques are used, namely, a selective catalytic reduction (SCR) system, aqueous ammonia solution, a low NO_x combustor and a dry NO_x removal apparatus, as discussed below:

1. *Selective catalytic reduction*

To control the emissions in the exhaust gas so that it remains within permitted levels as it enters the atmosphere, the exhaust gas passes though two catalysts located in the HRSG. One catalyst controls carbon monoxide (CO) emissions, whereas the other catalyst controls oxides of nitrogen, (NO_x) emissions.

(i) Aqueous ammonia: Apart from the SCR, a mixture of 22 per cent ammonia and 78 per cent water, known as aqueous ammonia is also injected into system to further reduce NO_x levels.

(ii) To reduce NO_x emissions that increase with the combustor exit gas temperature (1500°C), a low NO_x combustor can also be used.

(iii) A dry NO_x removal apparatus can be built in the heat recovery steam generator to operate the plant confirming environmental regulation limits.

12.9.2.5 Advantages and Disadvantages of CCGT Plants

Advantages

1. Fuel efficiency: As against the conventional power plants turbines fuel-conversion efficiency of 33 per cent, turbines in combined cycle power plant have a fuel-conversion efficiency of 50 per cent or more. This means the CCGT plant use only 50 per cent of fuel compared to a conventional plant to generate same amount of electricity.
2. Low capital costs: The capital cost for building a combined cycle unit is two-thirds the capital cost of a comparable coal-based power plant.
3. Ability to handle variety of fuel sources: The turbines used in combined cycle plants are fuelled with natural gas, which is more versatile than a conventional coal or oil. To meet the energy demand, the plant can also be designed to run on alternative fuels such as agriculture-based bio gas.
4. Lower emission and fuel consumption: The specific fuel consumption of combined cycle plants is less (fuel per kilowatt hour) and produces fewer emissions than conventional thermal power plants. This reduces the environmental damage caused by electricity production. The fuel used in CCPT is much cleaner when compared to coal-fired power plant.
5. Commercial availability: Combined cycle units are commercially available from all parts of the world. They can be easily manufactured, shipped, transported and commissioned.

Disadvantages

1. The gas turbine can only use natural gas or high-grade oils such as diesel fuel.
2. Its operation is location specific because the combined cycle can be operated only in locations where these fuels are available and cost effective.

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12.9.3 Thermodynamic Analysis of a Simple Combined Cycle GT–ST Plant

Consider a simple combined cycle plant, the gas turbine drives one electric generator and the steam turbine runs another, as illustrated in Figure 12.31. For the analysis purpose, let us ignore the losses in the heat exchanger.

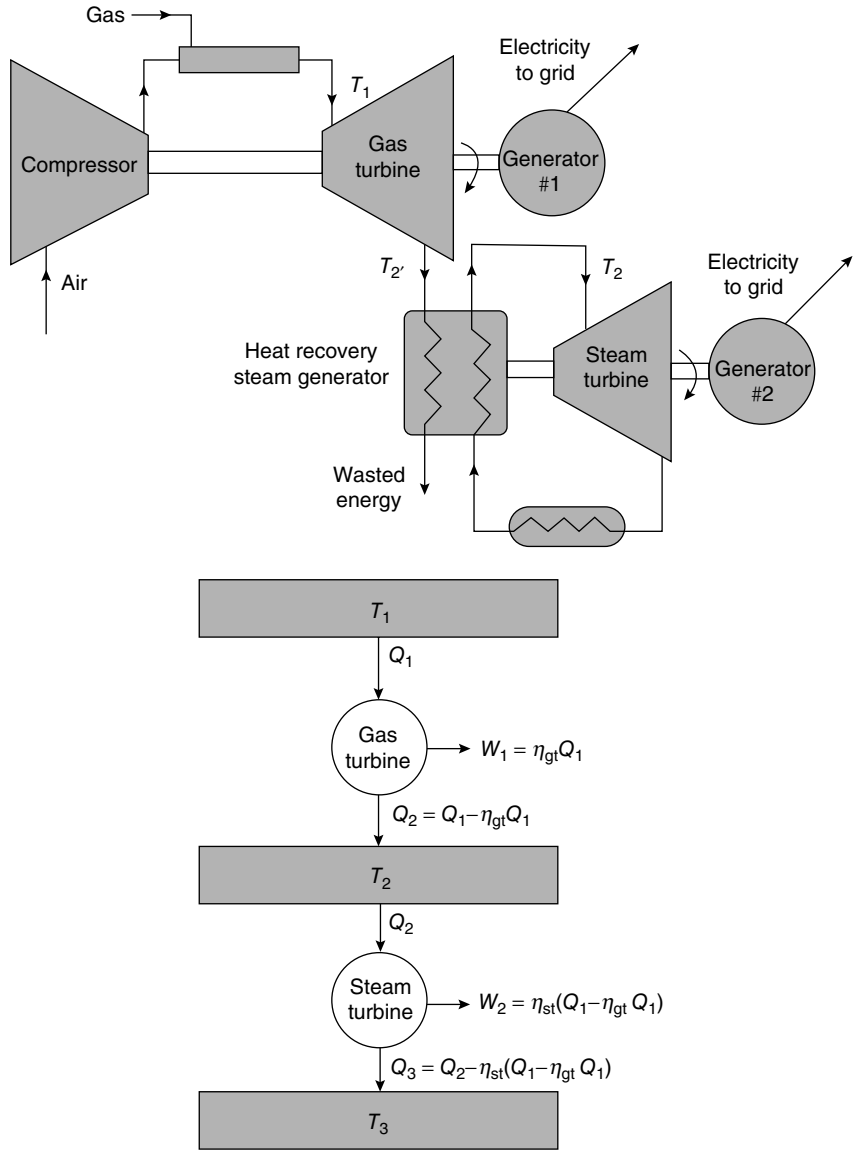


Fig. 12.31 *Thermodynamic Analysis of a Simple Combined Cycle Plant*

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Let η_{gt} = efficiency of the gas turbine cycle (topping cycle)
T_1 = inlet temperature of gas in gas turbine cycle (topping cycle)
Q_1 = heat rejected in gas turbine cycle (topping cycle)
η_{st} = efficiency of the steam turbine cycle (bottoming cycle)
T_2 = inlet temperature of steam in the steam turbine cycle (bottoming cycle)
Q_2 = heat rejected in the steam turbine cycle (bottoming cycle)

Efficiency of the topping plant is given by

$$\eta_{gt} = 1 - \frac{Q_2}{Q_1}$$

$$\therefore \quad Q_2 = (1 - \eta_{gt})Q_1$$

Efficiency of the topping plant is given by

$$\eta_{gt} = 1 - \frac{Q_3}{Q_2}$$

$$\therefore \quad Q_3 = (1 - \eta_{gt})Q_2$$

Combined cycle efficiency is given by

$$\eta_{cc} = 1 - \frac{Q_3}{Q_1} = 1 - \frac{Q_2\left(1-\eta_{gt}\right)}{Q_1} = 1 - \frac{Q_1\left(1-\eta_{gt}\right)\left(1-\eta_{st}\right)}{Q_1}$$

$$\therefore \quad \eta_{cc} = 1 - \left(1-\eta_{gt}\right)\left(1-\eta_{st}\right)$$

$$\eta_{cc} = \eta_{gt} + \eta_{st} - \eta_{gt}\eta_{st}$$

If there are n number of cycles in series, then combined cycle efficiency is given by

$$\therefore \quad 1 - \eta_{cc} = \left(1-\eta_1\right)\left(1-\eta_2\right)\cdots\left(1-\eta_n\right)$$

Or

$$1 - \eta_{cc} = \prod_{i=1}^{n}\left(1-\eta_i\right)$$

Considering the gas turbine cycle (topping cycle)
Work or power delivered by the steam turbine plant is given by

$$W_1 = \eta_{gt}\, Q_1$$

Considering the exhaust of the turbine, we have

$$Q_2 = Q_1 - \eta_{gt} Q_1$$

Work or power delivered by the steam turbine plant is given by

$$W_2 = \eta_{st} Q_2 = \eta_{st}\left(Q_1 - \eta_{gt} Q_1\right)$$

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Total power delivered by the steam and gas turbine is given by

$$W_{tot} = W_1 + W_2 = \eta_{gt} Q_1 + \eta_{st} \left(Q_1 - \eta_{gt} Q_1 \right)$$

$$W_{tot} = \left[\eta_{gt} + \eta_{st} \left(1 - \eta_{gt} \right) \right] Q_1$$

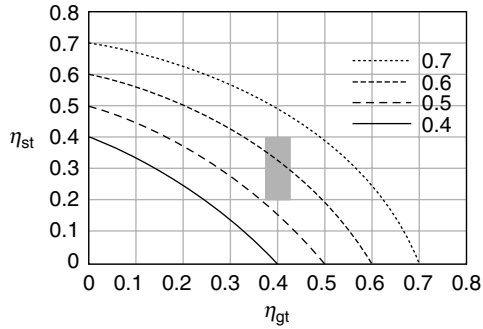


Fig. 12.32 *Plot of η_{gt} v/s η_{st} for Various Values of Combined Cycle Efficiency [11]*

Figure 12.32 shows the plot of η_{gt} v/s η_{st} for various values of combined cycle efficiency, η_{cc}. The shaded rectangle in this graph illustrates that overall efficiencies above 0.50 are possible with η_{st} as low as 0.20 if the corresponding η_{gt} is at least 0.38. With a steam turbine efficiency of 0.32 and a gas turbine efficiency of 0.42, the combined cycle efficiency is η_{cc} = 0.60. If the steam turbine efficiency is 0.40, then η_{cc} = 0.64.

Figure 12.33 shows the variation of efficiency of gas turbine, steam turbine and combined cycle plant with gas turbine exhaust temperature, T_2. It is evident from the plot that combined cycle offers best and highest efficiency compared to GT and ST plants as turbine exhaust temperature also increases.

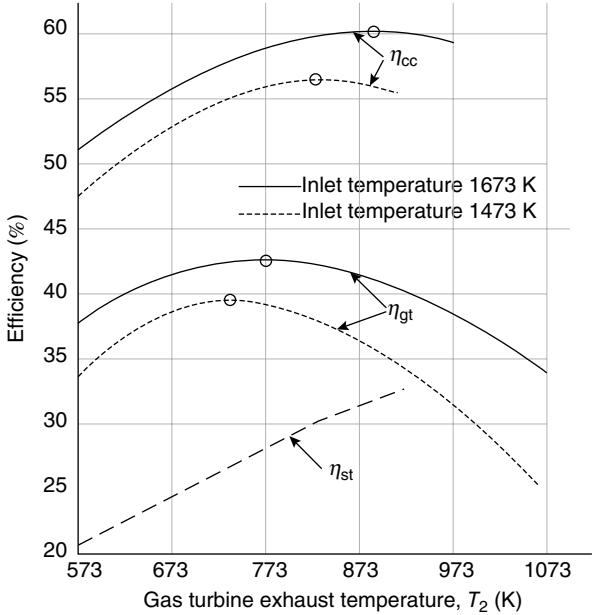


Fig. 12.33 *Variation of Efficiency of Various Cycles with Gas Turbine Exhaust Temperature, T2*
Source: Kehlhofer, R., R. Bachmann, H. Nielsen, and J. Warner, 1999, Combined Cycle Gas and Steam Turbine Power Plants, 2nd edition. Tulsa, OK: PennWell.

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12.10 LAYOUT OF GAS TURBINE PLANT

Figure 12.34 shows typical layout of a gas turbine plant. The air filler is used to clean the air. The air leaving the air filler flows to the LP compressor. The compressed air then flows through the intercooler and then it enters the high-pressure compressor. The air leaving the air compressor flows through the heat exchanger and the hot air flows to the combustion chamber. Products of combustion then enter in to low-pressure turbine, where it expands doing useful work.

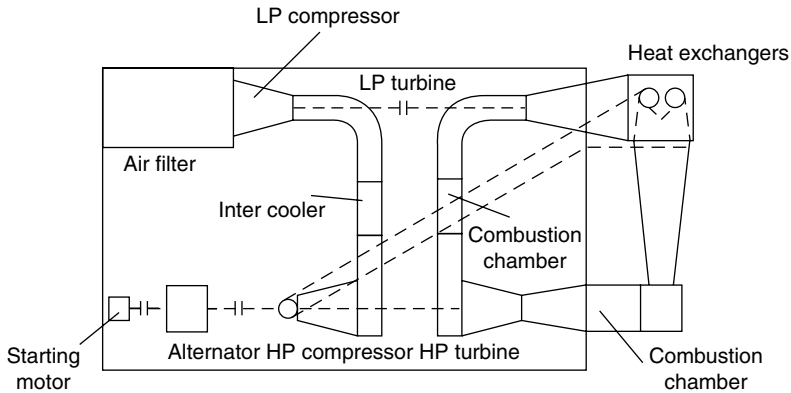


Fig. 12.34 *Layout of Gas Turbine Plant*

12.11 QUESTIONS

12.11.1 Objective Questions

1. In a simple Brayton cycle, the pressure ratio is 8 and temperatures at the entrance of compressor and turbine are 300 K and 1400 K, respectively. Both compressor and gas turbine have isentropic efficiencies equal to 0.8. For the gas, assume a constant value of p c (specific heat at constant pressure) equal to 1 kJ/kgK and ratio of specific heats as 1.4. Neglect changes in kinetic and potential energies.

   (i) The power required by the compressor in kW/kg of gas flow rate is

   (a) 194.7 (b) 243.4

   (c) 304.3 (d) 378.5

   (ii) The thermal efficiency of the cycle in percentage (%) is

   (a) 24.8 (b) 38.6

   (c) 44.8 (d) 53.1

2. An ideal Brayton cycle, operating between the pressure limits of 1 bar and 6 bar, has minimum and maximum temperatures of 300 K and 1500 K. The ratio of specific heats of the working fluid is 1:4. The approximate final temperatures in Kelvin at the end of the compression and expansion processes are, respectively,

   (a) 500 and 900 (b) 900 and 500

   (c) 500 and 500 (d) 900 and 900

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3. A gas turbine develops 120 kJ of work which the compressor absorbs 60 kJ of work and the heat supplied is 200 kJ. If a generator that would recover 40 per cent of the heat in exhaust were used, then the increase in the overall thermal efficiency would be
   (a) 10.2% (b) 8.6%
   (c) 6.9% (d) 5.7%
4. In a gas turbine cycle, the turbine output is 600 kJ/kg the compressor work is 400 kJ/kg and the heat supplied is 1000 kJ/kg. The thermal efficiency of the cycle is
   (a) 20% (b) 40%
   (c) 60% (d) 80%
5. In a single-stage cycle gas turbine, the mass flow through the turbine is higher than the mass flow through the compressor because
   (a) the specific volume of air increase by use of inter cooler
   (b) the temperature of air increases in the reheater
   (c) the combustion of fuel takes place in the combustion chamber
   (d) the specific heats at constant pressure for incoming air and exhaust gases are different
6. Consider the following statement with reference to a gas turbine cycle:
   1. Regeneration increases thermal efficiency.
   2. Reheating decrease thermal efficiency.
   3. Cycle efficiency increases when maximum temperatures of in cycle is increased.
   Identify the correct statements using the code given below:
   (a) 1, 2 and 3 (b) 2 and 3
   (c) 1 and 2 (d) 1 and 3
7. The use of regenerator in a gas turbine cycle increases
   (a) output but has no effect on efficiency
   (b) outpoint but has no effect on efficiency
   (c) both efficiency and out put
   (d) efficiency but decreases output
8. Consider the following statements comparing I.C. engines and gas turbines
   1. Gas turbines are simple, compact and light in weight
   2. Complete expansion of working substances is possible in gas turbine
   3. There is flexibility in the design of different components of gas turbines.
   4. Even low-grade fuels can be burnt in gas turbines.
   Select the correct statements using the codes given below:
   (a) 1, 2 and 3 (b) 1, 3 and 4
   (c) 2, 3 and 4 (d) 1, 2 and 4
9. Mechanical efficiency of a gas turbine as compared to internal combustion reciprocating engine is
   (a) same (b) lower
   (c) higher (d) none of these
10. The pressure ratio for a gas turbine is about
   (a) 15–20 (b) 10–15
   (c) 5–10 (d) 3–5

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11. For a fixed turbine inlet temperature, increasing the pressure ratio thermal efficiency of the cycle
    (a) decreases (b) first increases and then decreases
    (c) increases (d) first decreases and the increases
12. Reheating in a multistage gas turbine results in
    (a) decreased work ratio (b) increased work ratio
    (c) same work ratio (d) none of these
13. Gas turbine used in aircraft is
    (a) open-cycle turbine (b) closed-cycle turbine
    (c) constant volume turbine (d) none of these
14. In a multistage compressor, function of the intercooler is to
    (a) increase the efficiency, work done and fuel consumption
    (b) increase the efficiency, reduce work done, and fuel consumption
    (c) increase work done, reduce efficiency and fuel consumption
    (d) decrease efficiency work done and fuel consumption

Answers

1(i). c	1(ii). a	2. a	3. d	4. a	5. d	6. d	7. c	8. b
9. c	10. b	11. b	12. b	13. a	14. b			

12.11.2 Review Questions

1. Derive an expression for the work output of a gas turbine unit in terms of pressure ratio and maximum and minimum temperatures T_3 and T_1.
2. Discuss the methods used for increasing thermal efficiency and specific output of open cycle gas turbine.
3. (i) Derive the relation for optimum pressure ratio for maximum work of gas turbine and show that it is a function of limiting temperatures.
   (ii) Discuss the merits of gas turbines as compared with I.C. engines and steam turbines.
   (iii) Compare open-cycle and closed-cycle gas turbines.
4. Show that the net work output per kilogram of air in an ideal Brayton cycle is maximum when the pressure ratio is such that the compressor and turbine exit temperatures are same.
5. Sketch the flow diagram and the corresponding temperature-entropy diagram of a gas turbine plant having two-stage compression with intercooling, a regenerator and two-state expansion with reheating in between stages. Mark the state points clearly on both the diagrams.
6. Show that the efficiency at air standard Brayton cycle is a function of isentropic pressure ratio.
7. Explain briefly the effects of (i) regeneration, (ii) intercooling and (iii) reheating on Brayton cycle efficiency with a *T-s* diagram.
8. (i) Derive the expression of optimum pressure ratio for maximum net work output in an ideal Brayton cycle. What is the corresponding cycle efficiency?
   (ii) List down the methods of improving the efficiency of Brayton cycle.

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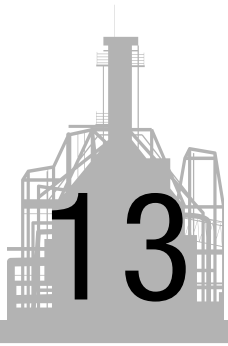
9. Explain with a neat sketch, the difference between open- and closed-cycle gas turbine.
10. Show that the efficiency of air standard Brayton cycle is a function of isentropic pressure ratio.
11. Sketch the ideal regenerative Brayton cycle in two-stage compression and expansion with inter-cooling and reheat. Mark the points on the corresponding *T-s* diagram.

12.11.3 References

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Diesel Engine Power Plant



13

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13.2 Types of diesel plants and components
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13.1 INTRODUCTION TO DIESEL ENGINE PLANT

Since the invention of diesel engines, diesel engine plants are finding increased application as either continuous or peak load source of power generation. Due to the economy of operation, DG plants are used to generate power in the range of 1–50 MW capacities and are extensively used to supplement hydroelectric or thermal power stations, namely, for starting from cold and in emergency conditions.

DG plants are more efficient than any other heat engines of comparable size. It is available at very short delivery times and can be started quickly and brought into service. It can burn fairly in a wide range of fuels. If may be rapidly extended to keep pace with load growth by adding generating units of suitable sizes.

13.1.1 IC Engine Nomenclature

Figure 13.1 shows a typical spark ignition engine. Different components of the engine are discussed below.

1. *Cylinder*

Cylinder is the most important component of the engine where combustion of fuel takes place. The cylinder is supported in position by the cylinder block at the top end and is covered by cylinder head. To reduce the wear and tear, cylinder is fitted with sleeves. Power is developed in the cylinder.

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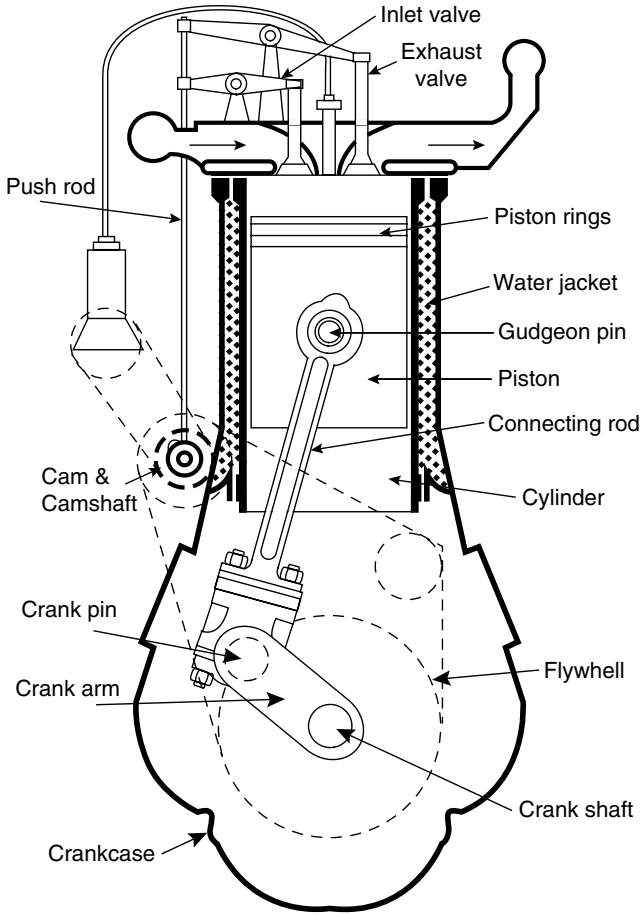


Fig. 13.1 *Cross Section of an IC Engine*

2. *Piston and piston rings*

The piston reciprocates inside the cylinder. It is a hollow cylindrical block that tightly fits inside the cylinder. The energy of the expanding gas is transmitted by the piston through *gudgeon pin* to the connecting rod. Grooves are made circumferentially on the top portion of the piston in which fit metallic *piston rings*. These rings form a gas tight joint so that no *charge* (fuel + air mixture) leaks into the crankcase. They also conduct heat from the piston to the cylinder walls.

3. *Connecting rod*

It connects the piston and the crank by gudgeon pin and crank pin, respectively. The function of the connecting rod is to transfer the reciprocating motion of the piston to the crankshaft.

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4. *Crank arm, crankshaft and crankcase*

Crank arm connects the connecting rod and the crankshaft. Connecting rod and crank arm translate the reciprocating motion of the piston into rotary motion of the crankshaft. Crank pin connects the connecting rod and crank arm, whereas the other end of the crank arm is secured to the crankshaft.

Crankshaft is supported in the bearings attached to the crankcase. Crankcase is the main body of the engine and houses crankshaft and other parts. It also acts as a sump for the lubricating oil.

5. *Inlet and exhaust valves*

Inlet and exhaust valves admit the *charge* inside the engine cylinder and discharge the products of combustion to the atmosphere, respectively. They are operated by valve mechanism. A camshaft is driven by the crankshaft through timing gears. Lobed cams on the camshaft actuate the push rods and rocker arms for opening the valves against the valve spring force. The valves are also called as *poppet valves*.

6. *Fly wheel*

A fly wheel is used to smooth out the power pulses and hence to obtain uniform rotation of the crankshaft, as power stroke exists only for a part of the total time. It is mounted on the crankshaft.

13.1.2 Standard Terminology

Consider a piston cylinder arrangement of an IC engine as shown in Figure 13.2.

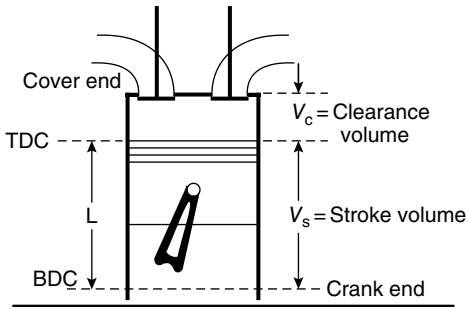


Fig. 13.2 *Piston Cylinder Arrangement of an IC Engine*

1. *Cylinder bore (D)*

It is the inner diameter of the working cylinder.

2. *Piston area (A)*

It is the cross-sectional area of the cylinder.

3. *Stroke (L)*

It is the distance through which the piston travels from crank end to the cover end. Stroke is also equal to twice the crank radius.

4. *Top dead centre (TDC)*

It is the position of the piston farthest from the crankshaft. In horizontal engines, it is also called inner dead centre.

5. *Bottom dead centre (BDC)*

It is the position of the piston nearest to the crankshaft. In horizontal engines, it is also called outer dead centre.

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6. *Displacement volume or swept volume (V_S)*
It is the volume covered by the piston between TDC and BDC. It is also called as stroke volume.

$$V_S = A \times L$$

7. *Clearance volume (V_C)*
It is the volume on the combustion side of the piston at TDC.

8. *Cylinder volume (V)*
It is the sum of piston stroke volume and clearance volume.

$$V = V_S + V_C$$

9. *Compression ratio (r_v)*
It is the ratio of cylinder volume and the clearance volume.

$$r_v = \frac{v}{v_C}$$

13.1.3 Four-Stroke Diesel Engine

A four-stroke diesel engine is similar to a four-stroke petrol engine except for the mode of combustion. Combustion is initiated by injecting fuel from a fuel injector instead of a spark plug. Air alone is compressed and hence the name compression ignition engine or CI engine. All diesel engines work on diesel combustion cycle, as shown in Figure 13.3.

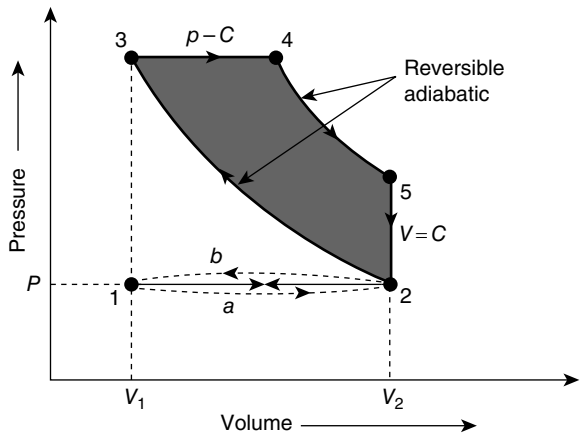


Fig. 13.3 *Diesel Engine Cycle*

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The cycle of operation is completed in four strokes of the piston or two revolutions of the crankshaft. Heat addition takes place during constant pressure combustion process and hence the name constant pressure cycle. The position of the piston during four different strokes is shown in Figure 13.4.

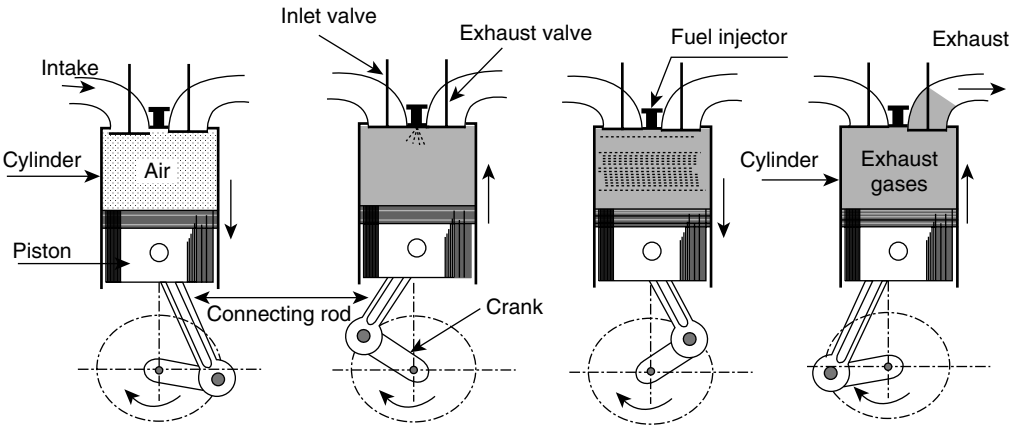


Fig. 13.4 *Four-Stroke Diesel Engine Cycle Operation*

1. *Suction stroke*

During suction, the piston moves from TDC to bottom dead centre (BDC). Due to the downward movement of the piston, suction is created and air alone from air filter enters the cylinder through the inlet valve. Exhaust valve is closed during this stroke. This is shown by the line 1-a-2 on the p-V diagram in Figure 3.10. Suction stroke completes during half revolution of the crankshaft.

Energy for this stroke during starting is provided by cranking and, flywheel supplies energy when the engine is running.

2. *Compression stroke*

During compression, piston moves from BDC to TDC. Air alone gets compressed in the combustion space. Both temperature and pressure of air increase beyond the self-ignition temperature of diesel. This process is represented by the reversible adiabatic curve 2–3 as shown in Figure 13.3. At the end of compression stroke, fuel in injected through the fuel injector. Rapid combustion takes place due to very high temperature of air. The stroke is completed when crankshaft completes one full rotation. The compression ratio ranges between 16 and 22.

Energy for this stroke during starting is provided by cranking and flywheel supplies energy when the engine is running.

3. *Expansion or power stroke*

During expansion stroke or power stroke or working stroke, piston moves from TDC to BDC, as pressure inside the cylinder increases due to the rapid combustion of fuel air mixture. Both inlet and exhaust valves are closed during expansion. Both temperature and pressure inside the cylinder decrease as shown by the reversible adiabatic expansion curve 3–5 in Figure 13.3. At

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the end of the stroke, exhaust valve opens and pressure drops suddenly to atmospheric pressure. But in an actual engine, pressure is slightly above atmospheric pressure.

Power developed during this stroke is used to run the crankshaft and the crankshaft rotates further half revolution.

4. *Exhaust stroke*

During the exhaust stroke, piston starts moving from BDC to TDC. Exhaust valve is open and inlet valve is closed during the stroke. As the pressure inside the cylinder is slightly above atmospheric pressure, piston sweeps out the burnt gases from the cylinder. This process is shown by the curve 2-b-1 in Figure 13.3. At the end of the exhaust stroke, exhaust valve closes leaving some residual gases in the cylinder. When exhaust stroke is completed, crankshaft completes two revolutions.

13.1.4 Two-Stroke Diesel Engine

A two-stroke diesel engine is similar in construction to a two-stroke petrol engine except in the method of ignition and compression of charge. A fuel injector replaces the spark plug and air alone is compressed in the crankcase.

A typical two-stroke diesel engine is shown in Figure 13.5. It consists of a piston-cylinder arrangement and a fuel injector mounted on the cylinder head. The crankcase in hermetically sealed and the cylinder is provided with inlet and exhaust ports. The transfer port is diametrically opposite to the exhaust port but slightly below its level.

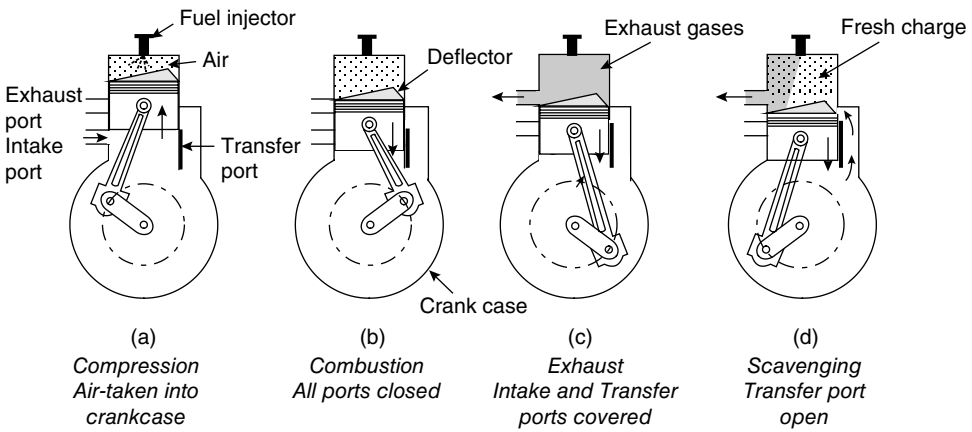


Fig. 13.5 *Two-Stroke Diesel Engine*

When the piston is at TDC both exhaust and transfer ports are covered by it as shown in Figure 13.5(a). The intake port uncovers and fresh air from air filter enters the crankcase due to the partial vacuum created by the piston moving upwards. During the same period, air in the combustion space gets compressed. This increases both temperature and pressure of air. The temperature of air becomes so high that it exceeds the self-ignition temperature of diesel. At this moment, fuel is injected from the fuel injector and combustion begins instantaneously.

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13.2 TYPES OF DIESEL PLANTS AND COMPONENTS

The diesel engines are generally classified as four-stroke engines and two-stroke engines. The four-stroke engine develops power after every two revolutions of crank shaft, whereas two-stroke engine develops power with each revolution of crank shaft. Generally, two-stroke engines are favoured for diesel power plants as they offer advantages over four-stroke engines.

Duel fuel engines

In duel fuel engines, both gas and oil are used as fuels for the engines. The gas is used as main fuel and oil is used as pilot fuel for ignition. The following figure shows a typical duel fuel engine cycle. In the duel fuel engine, the air and gas are taken in during suction stroke and the mixture is compressed during compression stroke. Near the end of the compression stroke, the pilot oil is injected into the cylinder. The compression heat first ignites the pilot oil and then gas mixture. The combustion and expansion take place similar to a diesel engine. The air–gas ratio is comparatively higher in duel fuel engines compared with gas engines.

Duel fuel engines if used in the diesel power plants can offer better economy by utilizing available gaseous fuels in the country. The gas may be a waste product as in the case of sewage treatment installations or oil fields where the economic advantage is self-evident. With the wider availability of natural gas, the duel fuel engines may become an attractive means of utilizing gas as fuel at off-peak tariffs for the generation of electrical energy.

Advantages and disadvantages of two-stroke engines over four-stroke engines

For comparing the merits and demerits of two-stroke engines, over four-stroke engines, the size and speed of the engines are considered same.

Advantages

(i) Theoretically, a two-stroke engine develops twice the power of four-stroke engine at the same speed. The actual power developed is 1.7–1.8 times of the power developed by four-stroke engines. This is because some of the power is used for compressing the air in crankcase and effective compression stroke is less than four-stroke engine for the same stroke.

(ii) The two-stroke engine is much lighter and more compact, and occupies less floor area for the same power developed.

(iii) The turning moment of two-stroke engine is more uniform than four-stroke engine. This ability of the engine reduces the size of the flywheel required. This further requires lighter foundations and reduces the installation cost to a greater extent.

(iv) It provides mechanical simplicity and, therefore, gives higher mechanical efficiency.

(v) The starting of two-stroke engine is much easier than four-stroke engine.

(vi) The capital cost of the plant with two-stroke engines is considerably less.

Disadvantages

(i) The thermodynamic efficiency of two-stroke engine is less than four stroke as the effective compression ratio is less than the four-stroke engine of the same dimensions.

(ii) The cooling of the engine presents difficulty as the quantity of heat removed per minute is large.

Oil cooling of the piston is necessary as there is possibility of overheating the piston due to firing in each revolution.

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(iii) The lubricating oil consumption is more as the operating temperatures are higher.

(iv) The scavenging is not complete particularly in high-speed engines (above 1000 rpm) as very short time is available for scavenging and hence the fresh charge is highly polluted. The pollution reduces the thermal efficiency of the engines.

The two-stroke CI engines are used only in diesel electric stations because its use is justified over four-stroke engines if the engines are built for larger output.

13.3 SELECTION OF ENGINE TYPE AND ENGINE SIZE

1. *Selection considerations*

Several factors need to be considered before selecting the type of engine, which is most suitable for a specific application. Out of these, the two most important factors used for engine selection are engine power and engine speed.

The power requirement is determined by the maximum load on the engine. Typically, the engine power rating should be 10–20 per cent higher than the power demand to ensure that the engine is not overloaded. This cushioning also gives enough room for the engine for absorbing extra load

(i) during starting/switching off the motors and other types of lighting systems

(ii) higher power consumption due to wear and tear of the equipment.

Engine speed is measured at the output shaft, expressed in revolutions per minute (rpm). Diesel engines operate over a range of speeds typically running at lower speeds (1300–3000 rpm). To avoid poor efficiency and to prevent build up of engine deposits due to incomplete combustion – which will lead to higher maintenance and running costs – all engines should be run as closely as possible to their rated speed. At this optimum speed, fuel efficiency will be greatest. Based on the requirement of the load, engine speed is decided.

If engine speed is matching with the generator speed, then direct coupling of engine and generator is possible; if not, a coupling device is essential, namely, a gearbox or a belt system. This will add to the cost and reduce the system efficiency.

Apart from the above two factors, several other factors that have to be considered while choosing an engine for a given application are cooling system, abnormal environmental conditions (dust, dirt, etc.), fuel quality, speed governing (fixed or variable speed), poor maintenance, control system, starting equipment, drive type, ambient temperature, altitude, humidity, etc.

2. *Selection of diesel generator type for captive power plants*

Diesel engine power plants are most frequently used in small power (captive non-utility) systems, as they offer higher efficiency compared with gas turbines and small steam turbines over the same output range considered. In applications requiring low captive power, without much requirement of process steam, the ideal method of power generation would be by installing diesel generator plants. The fuels burnt in diesel engines range from light distillates to residual fuel oils. The most frequently used diesel engine sizes are between the range 4–15 MW. For continuous operation, low-speed diesel engine is more cost-effective than high-speed diesel engine.

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Diesel power plants have the following benefits:

(i) low installation cost
(ii) short delivery periods and installation period
(iii) higher efficiency (as high as 43–45 per cent)
(iv) more efficient plant performance under part loads
(v) suitable for different type of fuels such as low sulphur heavy stock and heavy fuel oil in case of large capacities
(vi) minimum cooling water requirements
(vii) Adopted with air-cooled heat exchanger in areas where water is not available
(viii) Short start up time

Table 13.1 makes a brief comparison of different types of captive power plants (combined gas turbine and steam turbine, conventional steam plant and diesel engine power plant).

It is evident from the table that captive diesel plant has a distinct edge over the other two types of plants in terms of thermal efficiency, capital cost, space requirements, auxiliary power consumption, plant load factor, etc.

Table 13.1 *Comparison of Different Types of Captive Power Plants*

Description	Units	Combined GT & ST	Conventional steam plant	Diesel engine power plants
Thermal efficiency	%	40–46	33–36	43–45
Initial investment of installed capacity	₹/kW	8500–10,000	15,000–18,000	7500–9000
Space requirement		125% (approx.)	250% (approx.)	100% (approx.)
Construction time	Months	24–30	42–48	12–15
Project period	Months	30–36	52–60	12
Auxiliary power consumption	%	2–4	8–10	1.3–2.1
Plant load factor	kWh/kW	6000–7000	5000–6000	7200–7500
Start-up time from cold	Minutes	About 10	120–180	15–20

13.4 PLANT LAYOUT WITH AUXILIARIES

Figure 13.6 show general layout of a Diesel Engine plant. The essential components of (diesel engine plant) are

1. *Engine:* It is the main component that develops required power. The engine is directly coupled to the generator.
2. *Air filter and supercharger:* Air filter removes the dust from the air before it enters the engine. Supercharger increases the pressure of air at engine inlet and hence increases engine power. They are usually driven by the engines.

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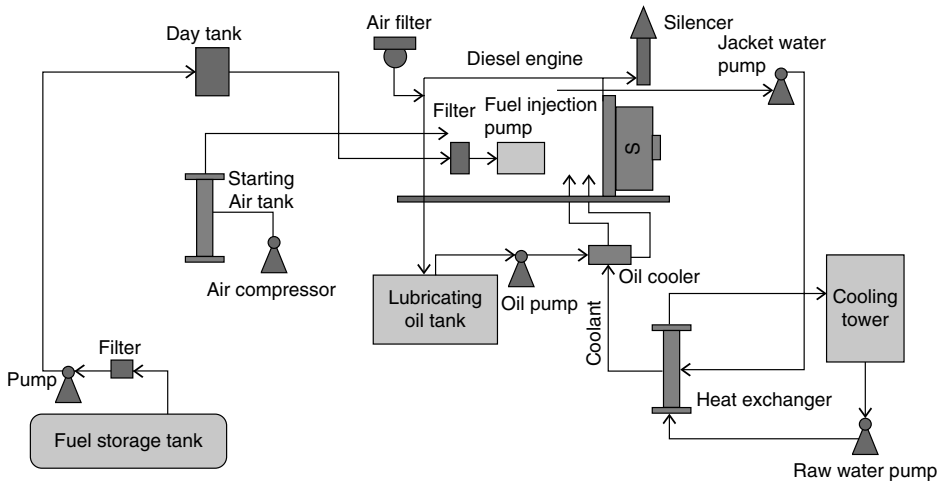


Fig. 13.6 *General Layout of DG Plant*

3. *Exhaust system:* The system includes silencers and connecting ducts. As the exhaust gases have higher temperatures, heat of exhaust gases is utilized for heating the oil or air supplied to the engine.
4. *Fuel system:* It contains the storage tank, fuel pump, fuel transfer pump, oil strainers and heaters. The amount of fuel supplied depends on the load on the plant.
5. *Cooling system:* The system includes water circulating pumps, cooling towers or spray ponds and water filtration or treatment plant. The purpose of cooling system is to ensure the life of the cylinder by extracting the heat developed from the engine cylinder walls and hence keeping the temperature within the safer range.
6. *Lubrication system:* The system includes oil pumps, oil tanks coolers and connecting pipes. The system reduces friction between the moving parts and hence reduces wear and tear.
7. *Starting system:* The system includes starting aides such as compressed air tanks. The tank supplies compressed air to start the engine from cold.
8. *Governing system:* The governing engine maintains constant speed of the engine irrespective of load on the plant. This is done by varying the fuel supplied to the engine.

13.5 FUEL SUPPLY SYSTEM

Diesel engine fuel injection systems can be divided into two basic types:

1. *Air injection*

In this system, the fuel valve is connected to a high-pressure air line fed by a multistage air compressor driven by the engine. The pressure of air may be about 60–70 bar. The blast air sweeps the fuel along with it when the fuel valve is opened. Thus, a well-atomized fuel spray is sent to the combustion chamber.

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2. *Solid injection*

In this system, fuel is directly injected into the combustion chamber without primary atomization. The method is also known as *airless mechanical injection*. The system consists of a pressurizing unit (fuel pump) and an atomizing unit (fuel injector). The high-pressure diesel is sprayed into the engine cylinder by the fuel injector.

Figure 13.7 shows the sectional view of Bosch fuel injection pump. It consists of a *barrel* with reciprocating *plunger* inside it driven by a camshaft. The plunger has a constant stroke and is single acting. A very small clearance of about two to three thousandths of a millimetre is provided between the pump barrel and the plunger. This arrangement provides a perfect sealing without any leakage even at very high pressures and low speeds. The pump barrel has two radially opposing ports known as *inlet port* and *spill port* or bypass port. A vertical channel extending from the top of the plunger to an annual helical groove is provided on the upper part of the plunger. It varies the quantity of fuel delivered per stroke. The top edge of the upper end is milled in the form of a helix. A spring-loaded delivery valve is provided at the top of the barrel. The position of the helical groove with respect to the spill port is changed by rotating the plunger with the *rack* or control rod. By moving the rack, the quantity of fuel injected can be varied from zero to that demanded at full load.

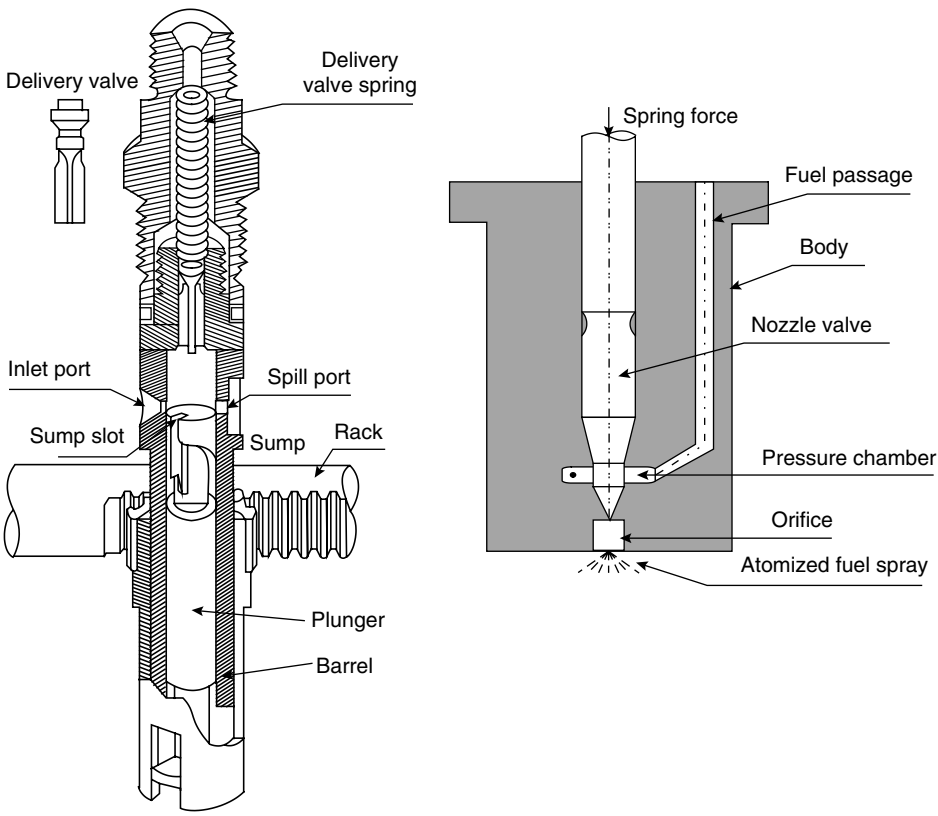


Fig. 13.7 *Bosch Fuel Injection Pump*

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3. *Operation*

During the delivery stroke, a cam raises the plunger upwards and a plunger return spring brings it back to BDC position. When the plunger is at the bottom of its stroke, fuel starts flowing into the barrel through the inlet port. Fuel fills the space above the plunger in the vertical groove and in the space between the helix.

When the plunger starts moving upwards, some amount of fuel enters the barrel as the inlet ports are uncovered. Further movement of the plunger upwards covers both the inlet and spill ports and the trapped fuel gets compressed. Due to high pressure, fuel is forced through the delivery valve and enters the injector which injects fuel into the combustion chamber. The injection process continues till the end of the upward stroke and stops when the lower end of the helix uncovers the spill port. As soon as spill port uncovers, the fuel pressure in the barrel drops suddenly due to the flow of fuel back to the suction chamber through the vertical channel. Injection process ends as both spring-loaded injector valve and spring-loaded delivery valve are suddenly closed. Plunger movement controls the quantity of oil delivered.

Figure 13.8 shows a typical fuel system used in diesel plants. The fuel oil may be delivered at the plant site by trucks, railroad tank cars or barge and tankers. From tank truck, the delivery is done using the unloading facility to main storage tanks. This fuel is then transferred by pumps to small service storage tanks known as *engine day tanks*. The main flow is ensured by arranging the piping equipment with the necessary heaters, by passes, shut offs, drain lines, relief valves, strainers and filters, flow meters and temperature indicators. The actual flow depends on type of fuel, engine equipment, size of the plant, etc. The tanks should contain manholes for internal access and repair, fill lines to receive oil, vent lines to discharge vapours, overflow return lines for controlling oil flow and a suction line to withdraw oil. Coils heated by hot water or steam reduce oil viscosity, which lower pumping power needs.

The minimum storage capacity of at least a month's requirement of oil is generally kept in bulk. However, when the advantages of seasonal fluctuations in cost of oil are to be availed, it may be necessary to provide storage for a few months' requirements. Day tanks supply the daily fuel needs of engines and are placed at a height so that oil may flow to engines under gravity.

The fuel oil supply system operation would be satisfactory if the following points are taken care of:

(i) Provisions for cleanliness and for changing over of lines during emergencies.

(ii) Provisions for tight pipe joints in all suction lines.

(iii) Keep all oil lines under air pressure with the joints tested with soap solution. Small air leaks into the line can be the source of operating difficulties and are hard to rectify once the plant is in operation.

(iv) Flush the piping between filter and the engine thoroughly by oil before being first placed in service.

(v) Due importance should be given for cleanliness in handling bulk fuel oil. Dirt particles will ruin lap of injection pumps or plug the injection nozzle orifices. Thus, high-grade filters are of great importance to the diesel oil supply system.

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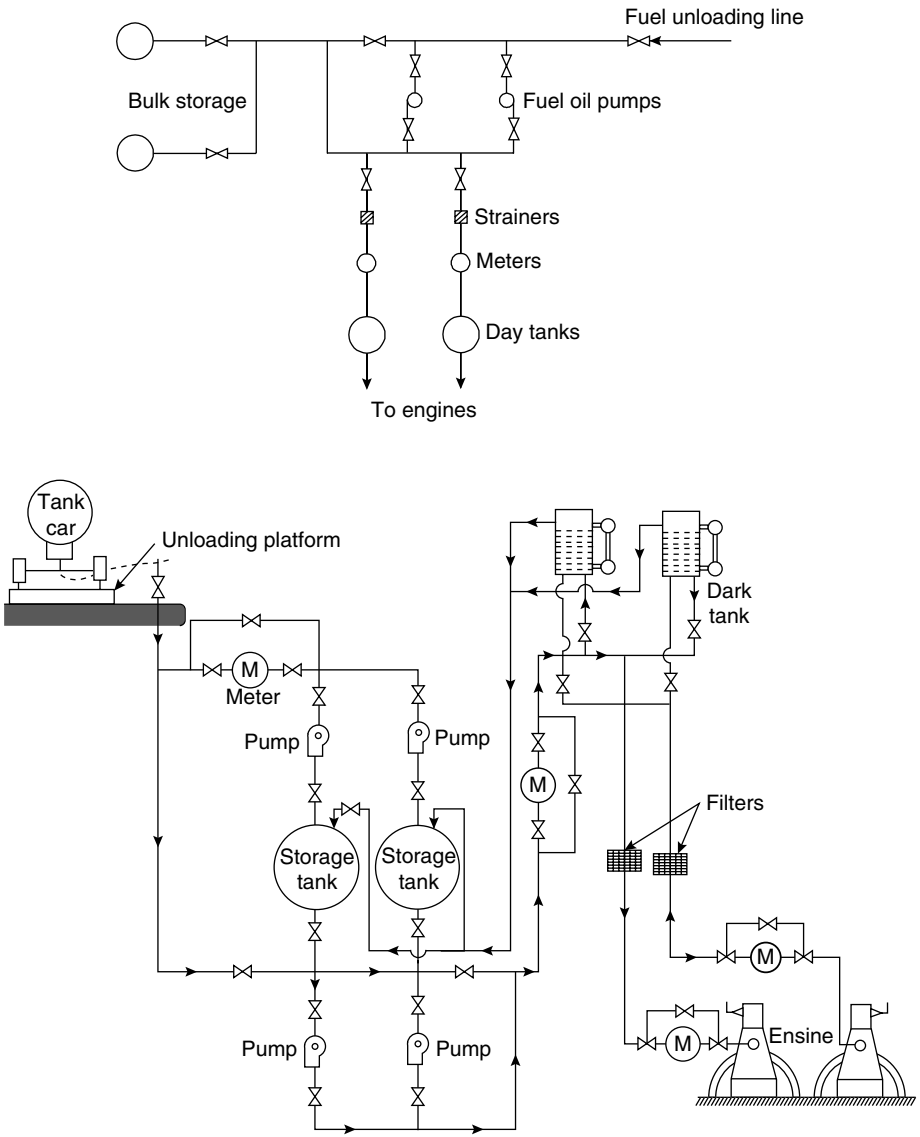


Fig. 13.8 *System of Fuel Storage for a Diesel Power Plant*

13.5.1 Fuel Injection System

Fuel injection system is the heart of the diesel engine. In an injection system, very small quantity of fuel must be measured out, injected, atomized and mixed with combustion air. The mixing problem becomes more difficult, the larger the cylinder and faster the rotational speed.

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However, special combustion arrangements such as pre-combustion chambers, air cells, etc., are necessary to ensure good mixing. Engines driving electrical generators have lower speeds and simple combustion chambers.

Functions of a fuel injection system are as follows:

(i) Filter the fuel
(ii) Meter or measure the correct quantity of fuel to be injected
(iii) Properly time the fuel injection
(iv) Control the rate of fuel injection
(v) Atomize or break-up the fuel into fine particles
(vi) Properly distribute the fuel in the combustion chamber

The injection systems especially the parts that are actually manufactured with great accuracy meter and inject the fuel. Some of the tolerances between the moving parts are so small (of the order of 1 μ) that they require special attention during manufacture. Hence, the injection systems are costly.

13.5.2 Types of Fuel Injection Systems

Most commonly used fuel injection systems in diesel power station are as follows:

(i) Common rail injection system
(ii) Individual pump injection system
(iii) Distributor

Atomization of fuel oil can be done by (i) air blast and (ii) pressure spray. In the olden days, engines used air fuel injection at about 70 bar, which was sufficient not only to inject the oil, but also to atomize it for a rapid and thorough combustion. The expense of providing an air compressor and tank led to the development of 'solid' injection, using liquid pressure of between 100 and 200 bar, which is sufficiently high to atomize the oil it forces through spray nozzles.

1. *Common rail injection system*

Two types of common rail injection systems are shown in Figure 13.9 and 13.10, respectively.

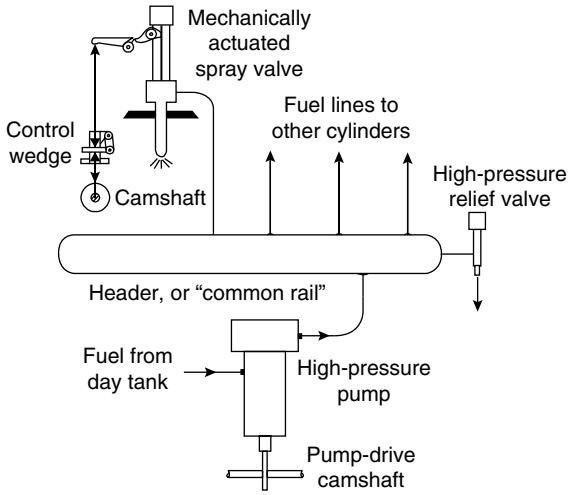


Fig. 13.9 *Common Rail Injection System Using a Single Pump*

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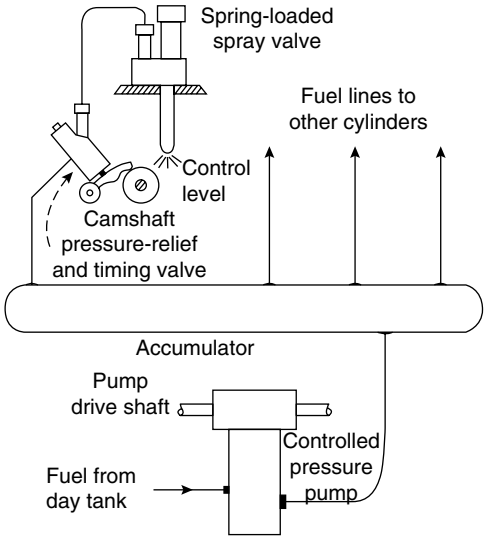


Fig. 13.10 *Controlled Pressure System*

In this case, a single pump supplies high-pressure fuel to header. A relief valve holds pressure constant. The control wedge adjusts the lift of mechanical operated valve to set the amount and time of injection.

Figure 13.10 shows a controlled pressure system. It has a pump that maintains set head pressure. Pressure relief and timing valves regulate injection time and amount. Spring-loaded spray valve acts merely as a check.

2. *Individual pump injection system*

An individual pump injection system is shown in Figure 13.11. In this system, an individual pump cylinder connects directly to each fuel nozzle. Pump meters charge and control injection timing. Nozzles contain a delivery valve that is actuated by the fuel oil pressure.

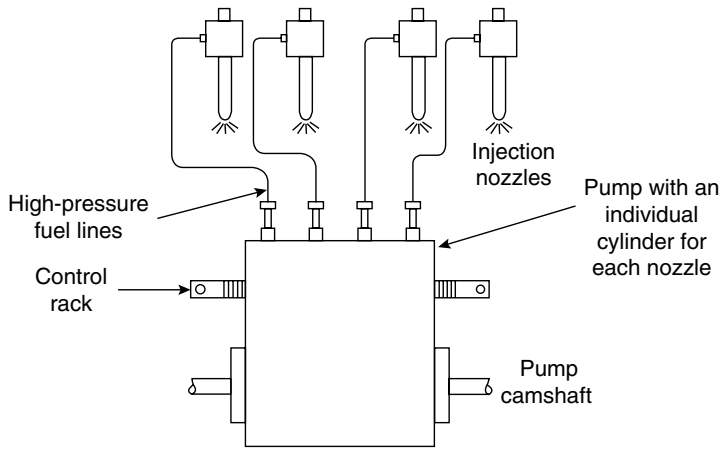


Fig. 13.11 *Individual Pump Injection System*

3. *Distributor system*

Most commonly used distributor system is shown in Figure 13.12. In this system, the fuel is metered at a central point. A pump pressurizes, meters the fuel and times the injection. The fuel is then distributed to cylinders in correct firing order by cam-operated poppet valves that open to admit fuel to the nozzles.

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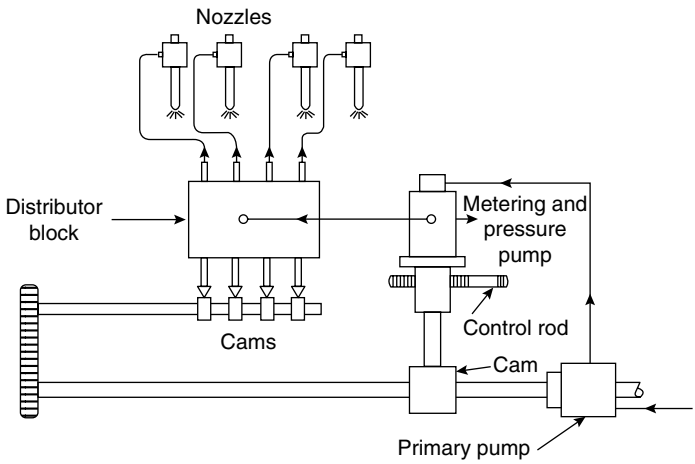


Fig. 13.12 *Distributor System*

13.6 SUPER CHARGING

It is well known that the power output of an engine increases with an increase in amount of air in the cylinder at the beginning of compression stroke. This is because air allows burning more quantity of fuel. Supercharging is a process that helps to increase the suction pressure of the engine above atmospheric pressure and the equipment used for this purpose is known as supercharger.

The advantages of supercharged engines are as follows:

(i) *Power increase:* By supercharging the engine, the engine output can be increased by 30–50 per cent at the same speed of the engine.

(ii) *Fuel economy:* The combustion in supercharged engine is better as it provides better mixing of the air and fuel than un-supercharged engine. Hence, the specific fuel consumption of a supercharged engine is less than a natural aspirated engine. The thermal efficiency of a supercharged engine is also higher.

(iii) *Mechanical efficiency:* The mechanical efficiency of a supercharged engine is better than a natural aspirated engine at the same speed. This is because the power increase due to supercharging increases faster than the rate of increase in friction losses.

(iv) *Scavenging:* With the increase in supercharged pressure, the scavenging action is better in two-stroke supercharged engines as compared to naturally aspirated engines, because the quantity of residual gases is reduced.

(v) *Knocking:* Supercharging reduces the possibility of knocking in diesel engines because delay period is reduced with an increase in supercharged pressure. Actually, supercharging results in smoother running of the engine. It has been found that four-stroke engines are more easily adaptable to supercharging than two-stroke engines.

Due to the number of advantages of supercharging mentioned above, modern diesel engines used in diesel plant are generally supercharged. By supercharging, the size of the engine is

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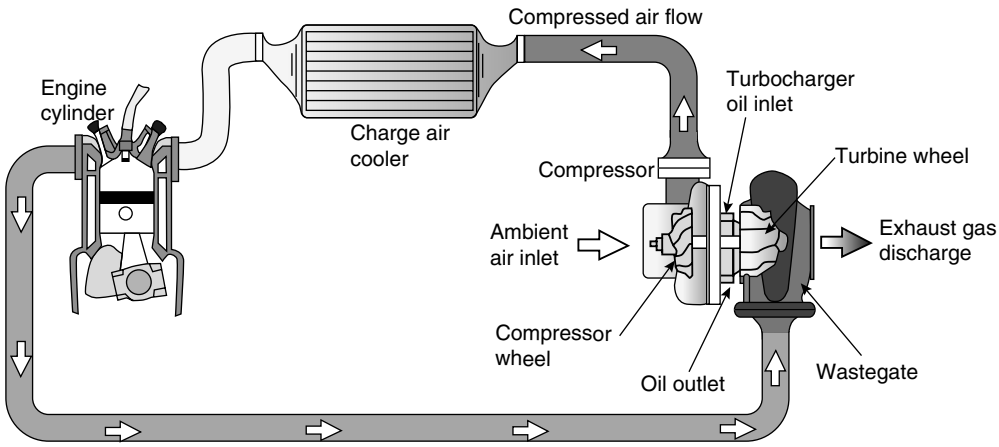


Fig. 13.13 *Working of a Turbo-Charged Engine*

reduced for a given output and consequently the space requirements and civil engineering works also reduced.

The Superchargers that are considered for diesel power plants are positive displacement type, centrifugal type and exhaust turbocharger. The selection depends upon its relative merits for a particular situation. Figure 13.13 shows a typical turbocharged engine.

A turbocharger is used to force air/fuel mixture into an engine at a pressure greater than the natural atmospheric pressure. The exhaust coming out of the engine is pushed through a turbine, mounted on a shaft, which in turn spins an air compressor. The compressor draws air in and blows it into the inlet manifold, and generates the boost. Forcing the air/fuel mixture into an engine allows it to burn more fuel and generate more power without changing engine capacity. When the air is compressed (with a turbo) it gets hotter. As hotter air contains less oxygen than cooler air, less oxygen is available to burn extra fuel going into the engine.

13.7 METHOD OF STARTING DIESEL ENGINES

The SI engines used for power generation in DG plants are usually small in size, which use compression ratio from 7 to 11. Hand and electric motor (6–12V, DC) cranking are generally used to start the engine.

The CI engines use very high compression ratios from 20 to 22 and hence it is difficult to hand crank the engine. Hence, some mechanical cranking systems are used.

1. *Compressed air system*

In this system, shown in Figure 13.14, air at a pressure of 20–30 bar is supplied from an air tank at the engine inlet through intake manifold. In the case of multi-cylinder engine, compressed air enters one or more of the engine cylinders and forces down the piston to turn the engine shaft. During the meantime, suction stroke of some other cylinder takes place and the compressed air pushes this cylinder and causes the engine shaft to rotate. Gradually the engine gains momentum and by supplying the fuel engine starts running.

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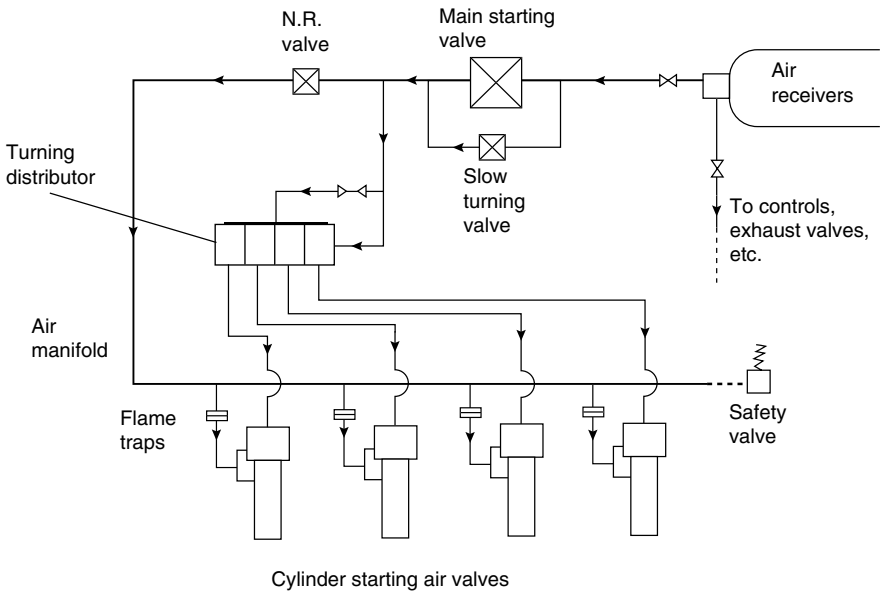


Fig. 13.14 *Compressed Air System*

2. *Electric starting*

An electric starting system is shown in Figure 13.15. It consists of an electric motor driving a pinion, which engages a toothed rim on engine flywheel. Electric supply for the motor is made using a small electric generator driven by the engine. A storage battery of 12–36 V is used to supply power to the electric motor. The electric motor disengages automatically after the engine has started.

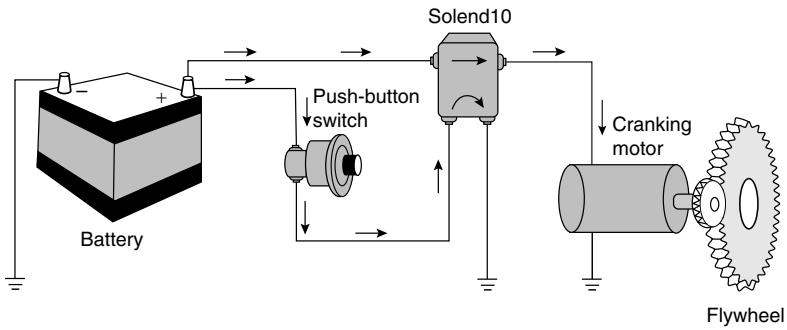


Fig. 13.15 *Electric Starting System*

The starting motor for diesel and gasoline engines operates on the same principle as a direct current electric motor. The motor is designed to carry extremely heavy loads but, because it draws a high current (300–665 A), it tends to overheat quickly. To avoid overheating, the motor

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should not be run for more than the specified amount of time, usually 30 seconds. A time lag of 2 or 3 min is essential before using it again. The starting motor is located near the flywheel. When the starting switch is closed, the drive gear on the starter meshes with the teeth on the flywheel .The drive mechanism (1) transmits the turning power to the engine when the starting motor runs, (2) disconnects the starting motor from the engine immediately after the engine has started and (3) provides a gear reduction ratio between the starting motor and the engine. The drive mechanism disengages the pinion from the flywheel immediately after the engine starts to avoid any damage, since this engaged position may increase the motor shaft speed to 22,500–30,000 rpm as against the engine speed of 1500 rpm.

3. *Hydraulic starting systems*

Several types of hydraulic starting systems are in use for engines. A typical hydraulic starting system is shown in Figure 13.16. In general most systems consist of a hydraulic starting motor, a piston-type accumulator, a manually operated hydraulic pump, an engine-driven hydraulic pump and a reservoir for the hydraulic fluid. Hydraulic pressure is provided in the accumulator by a manually operated hand pump or from the engine-driven pump when the engine is operating. By operating the starting lever, the control valve allows hydraulic oil (under pressure of nitrogen gas) from the accumulator to pass through the hydraulic starting motor, and cranks the engine. When the starting lever is released, spring action disengages the starting pinion and closes the control valve. This stops the flow of hydraulic oil from the accumulator. To protect the starter from the high speeds of the engine, an overrunning clutch is used. The hydraulic starting system is used on some smaller diesel engines.

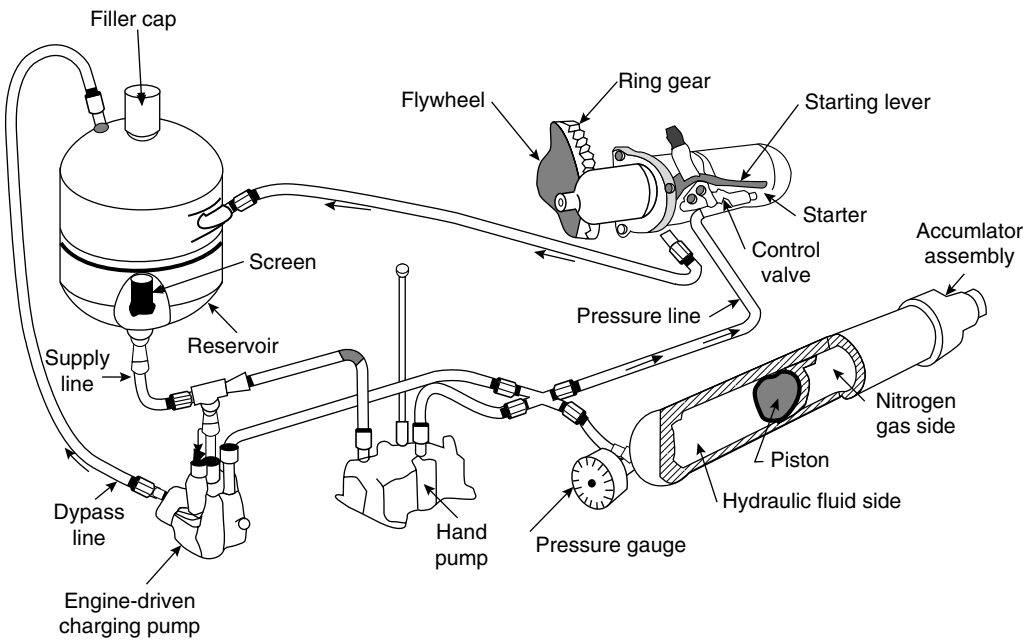


Fig. 13.16 *Hydraulic Starting System*

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4. *Starting by auxiliary engine*

In this method, a small petrol engine is connected to the main engine through clutch and gear arrangement. The clutch is disengaged and the petrol engine is started by hand. Then the clutch is gradually engaged and the main engine is cranked for starting. Clutch is disengaged automatically when the main engine is started.

13.8 COOLING AND LUBRICATION SYSTEM FOR THE DIESEL ENGINE

Cooling and lubrication systems form an essential component of diesel engine plants. An efficient cooling and lubrication system not only improves the plant efficiency, but also ensures longer plant life. Given its importance, both of these systems are discussed hereunder.

13.8.1 Engine Cooling System

During the process of converting the thermal energy to mechanical energy, high temperatures are produced in the cylinders of the engine as a result of combustion. A large portion of heat from the products of combustion is transferred to the cylinder head and walls, piston and valves. This excess heat if not carried away by an efficient cooling system will damage the engine.

Necessity of engine cooling: Engine cooling is necessary for the following reasons:

(i) The maximum operating temperature of lube oil ranges from 160°C to 200°C. If the temperature exceeds this limit, lube oil deteriorates and even might evaporate or burn. This may result in damaging of piston and cylinder surfaces or piston seizure.

(ii) Due to higher engine temperature, the strength of the materials used for various engine parts decreases. This may result in excessive thermal stresses due to uneven expansion of various engine parts and result in *cracking*.

(iii) Hot engine parts result in hot exhaust valve, which causes pre-ignition and knocking.

(iv) Due to higher engine temperature, volumetric efficiency and power output of the engine reduces.

Two basic systems used for cooling the engine are

1. *Air cooling*

In this system, atmospheric air is circulated around the engine cylinder to dissipate the excess heat. Heat transfer area is increased by providing fins on the cylinder and cylinder head and air is passed over them. Air cooling is used for small engines and aircraft engines. Using the fins increases the heat transfer surface by about 5–10 times its initial value. Figure 13.17 shows a sectional view of an engine cylinder using fins. As maximum temperature exists near the exhaust valve and cylinder head, more fins are required here to dissipate heat.

The fins may be cast integral with the cylinder and cylinder head or may be fixed separately to the cylinder block. The number of cast fins may be 2–4 in number per centimetre or 4–6 fins per centimetre in case of machined fins. The spacing between the fins is limited to 2–5 mm and height of the fin varies from 20 to 50 mm.

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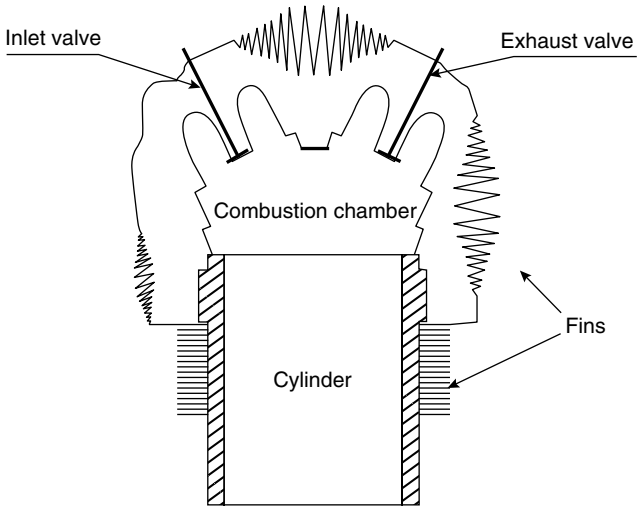


Fig. 13.17 *Air Cooling System*

2. *Water cooling*

In this system, the cylinder and cylinder head are enclosed in a water jacket. The water jacket is connected to a radiator or heat exchanger normally at the front end of the vehicle. Water is made to circulate through the water jacket, which cools the engine. The hot water returns to the radiator where it exchanges heat with atmospheric air. The cold water is again re-circulated.

Water cooling is used for bigger engines such as cars, buses, trucks, etc.

High operating temperature existing in the engine may disintegrate the lube oil film on the cylinder liners resulting in warping of valves and piston seizer, if the engine is not cooled properly. Thus, proper cooling of the engine is necessary to increase engine life. This is done by controlling the exit temperature of cooling water. If the exit temperature is too low, lube oil will not spread over the piston and the cylinders resulting in wear and tear. If it is too high, the lube oil burns and disintegrates. The exit temperature of cooling water is thus limited to 70°C. As constant flow rate of cooling water increases the exit temperature of cooling water with increased load, flow regulation of water is desirable to control the same.

The hot jacket water from the engine is passed through the coolers (hot well) where it is cooled with the help of raw water. The raw water is cooled in the cooling towers using either natural draft or forced draft air circulation. The sensible heat of water is transferred to air. In addition, the latent heat of evaporation of water vaporized is the main source of heat transfer. The degree of cooling action is limited by the vapour that can be absorbed before the air reaches saturation humidity at its leaving temperature. As the circulation of water is concerned, the cooling systems are generally divided into two types.

(i) *Open- or single-circuit system*

In this system shown in Figure 13.18, pump draws water from cooling pond and forces it into the main engine jackets. Water, after circulating through the engine, returns to the cooling pond. The engine jacket is subjected to corrosion because of the dissolved gases in the cooling water.

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Figure 13.19 shows a closed or double circuit system.

(ii) *Closed- or double-circuit system*

In this system, raw water is made to flow through a heat exchanger when it takes up the heat of jacket water and returns back to the cooling pond or tower.

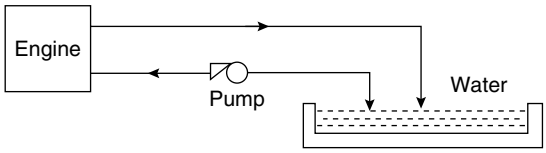


Fig. 13.18 *Open- or Single-Circuit System*

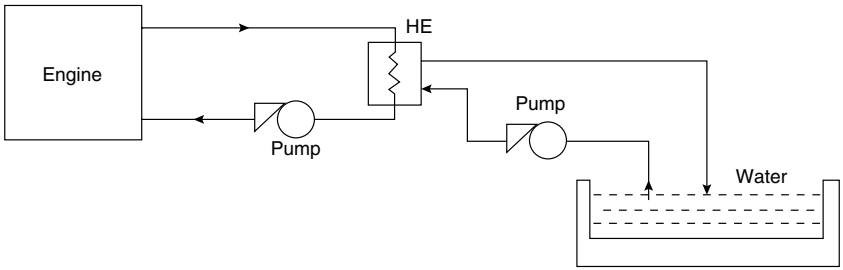


Fig. 13.19 *Closed- or Double-Circuit System*

About 25–35 per cent heat is lost by cooling water, which is known as jacket water loss. The rate of flow of water should be adjusted to maintain outlet cooling water temperature to 60°C and rise in temperature of cooling water is limited to 11°C. Water used for cooling should be free from impurities. Even though cooling by water is uniform, it poses a problem in cold weather. Cooling efficiency is reduced due to scaling in pipes, jackets and radiator. Engine efficiency is affected as some power is utilized to drive the water pump and radiator fan. This type of cooling system eliminates internal jacket corrosion, but the corrosion may exist in the raw water circuit.

13.8.2 Lubrication System

Due to the presence of friction, wear and tear of the engine parts takes place reducing the engine life. The lubricant introduced forms a thin film between the rubbing surfaces and prevents metal to metal contact. The various parts that require lubrication are cylinder walls and pistons, crank pins, gudgeon pins, big end and small end bearings, etc. The maintenance of proper lubrication system for all moving parts is an important problem in the operation of an IC engine.

The purpose of lubrication are (i) to reduce the power required to overcome friction, (ii) to increase the power output and (iii) to increase the engine life. Improper lubrication results in the breakdown of the lubricating films, causing piston seizure and serious damage to the engine.

Two basic systems used for engine lubrication are as follows:

1. *Wet sump*

In the wet sump system, the bottom of the crankcase contains an oil pan or sump which serves the purpose of an oil supplying tank or reservoir tank, or oil cooler. Oil dripping from the cylinders and bearings flows by gravity back into the wet sump where it is picked up by a pump and re-circulated through the engine lubrication system (Figure 13.20).

The types of wet sump systems generally used are (i) splash and circulating pump (force feed) system, (ii) force-feed system and (iii) full force-feed system.

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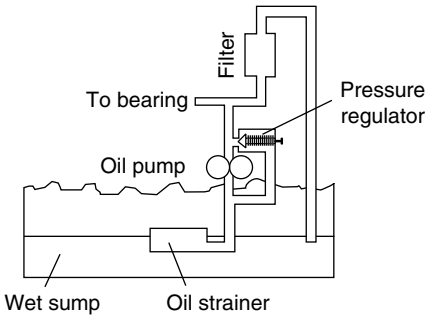


Fig. 13.20 *Wet Sump Lubrication System*

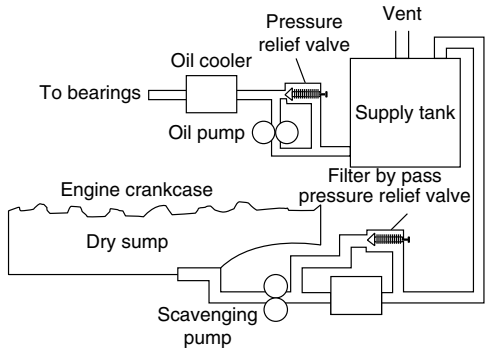


Fig. 13.21 *Dry Pump Lubrication System*

2. *Dry sump*

In this dry sump system, oil is supplied from an external tank. Oil drips from the cylinders and bearings into the sump. Oil is removed from the sump and passed back to the external tank through a filter. As the sump pump capacity is greater than oil pump capacity, oil is prevented from accumulating in the engine base (Figure 13.21).

Lubrication may be achieved in different forms: Full pressure lubrication, mechanical, force feed lubrication or gravity circulation from an overhead tank. Some of the lubrication systems are explained below.

(i) *Splash and force-feed lubrication system*

In a splash and force-feed lubrication system (Figure 13.22), oil is delivered to some parts by means of splashing and other parts through oil passages under pressure from the oil pump. In this system, the oil from the pump entering the oil galleries flows to the main bearings and camshaft bearings. The main bearings have oil-feed holes or grooves that feed oil into drilled passages of the crankshaft. The oil flows through these passages further to the connecting rod bearings and from there through holes drilled in the connecting rods to the piston-pin bearings.

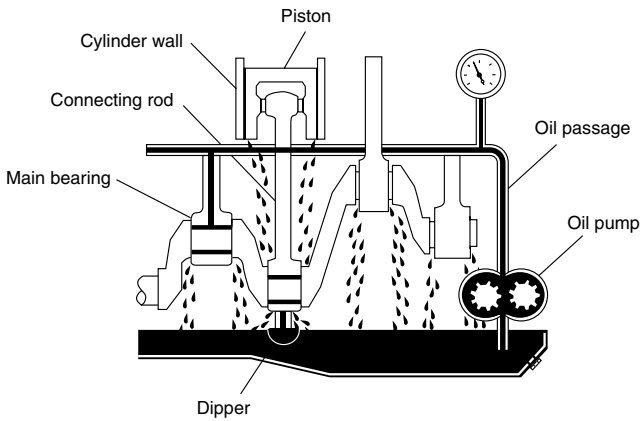


Fig. 13.22 *Splash and Force-Feed Lubrication System*

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Cylinder walls are lubricated by splashing oil thrown off from the connecting-rod bearings. In some engines, small troughs are provided under each connecting rod that are kept full by small nozzles, which deliver oil under pressure from the oil pump. As engine speed increases, these oil nozzles deliver an increasingly heavy stream, powerful enough to strike the dippers directly. This generates a much heavier splash so that adequate lubrication of the pistons and the connecting-rod bearings is provided. In an overhead valve engine, the upper valve train is lubricated by pressure from the pump.

(ii) *Force-feed lubrication system*

In a force-feed lubrication system (Figure 13.23), oil is forced by the oil pump from the crankcase to the main bearings and the camshaft bearings. Unlike the *splash and force-feed lubrication system*, the connecting-rod bearings are also fed oil under pressure from the pump.

Oil passages drilled in the crankshaft lead oil to the connecting-rod bearings. The passages deliver oil from the main-bearing journals to the rod-bearing journals either through holes or through annular grooves. The holes line up once for every crankshaft revolution. In the case of annular grooves in the main bearings, oil can be fed constantly into the hole in the crankshaft.

The pressurized oil that lubricates the connecting-rod bearings also lubricates the pistons and walls by squirting out through strategically drilled holes. This lubrication system is used in all engines that are equipped with semi-floating piston pins.

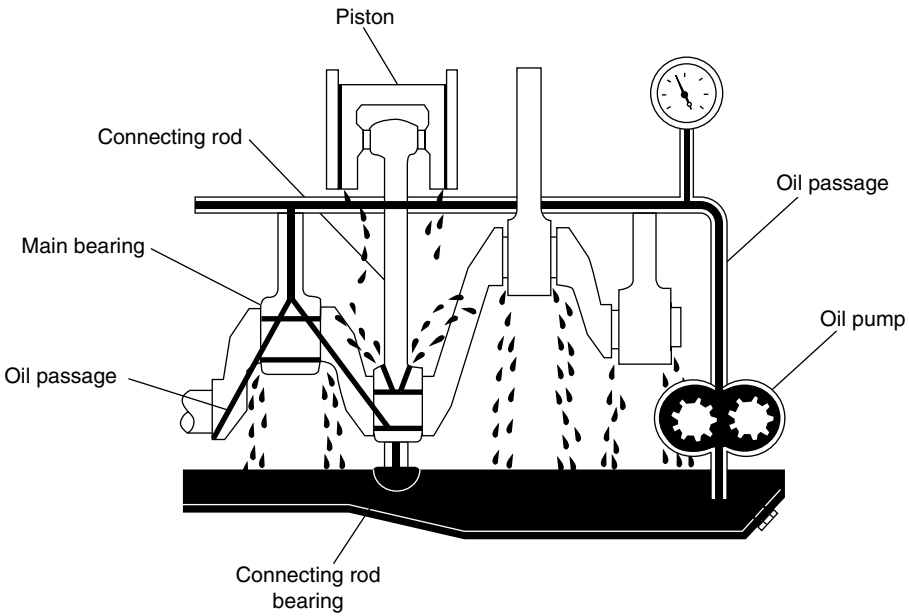


Fig. 13.23 *Force-Feed Lubrication System*

(iii) *Full force-feed (pressure) lubrication system*

In a full force-feed lubrication system, an oil pump supplies lubricating oil to many parts of the engine through duct system and to the crank shaft through drilled holes. The cylinder walls are

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lubricated by oil mist that is slung outward from the connecting rod bearings or by splash of rod ends into oil pools. This system lubricates the main bearings, rod bearings, camshaft bearings and the complete valve mechanism. In addition, lubrication of the pistons and the piston pins is also enabled under pressure by holes drilled throughout the length of the connecting rod, creating an oil passage from the connecting rod bearing to the piston pin bearing. This passage not only feeds the piston pin bearings but also provides lubrication for the pistons and cylinder walls. This system is used in all engines that are equipped with full-floating piston pins (Figure 13.24).

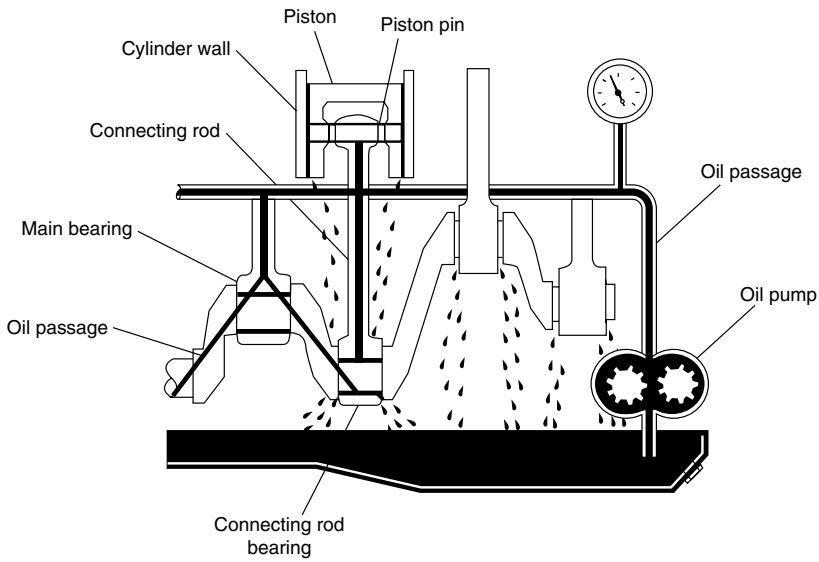


Fig. 13.24 *Full Force-Feed (Pressure) Lubrication System*

13.8.3 Filters, Centrifuges and Oil Heaters

1. *Filters*

The air cleaner or filter filters the incoming air. It also muffles the resonation (i.e. dampens the noise) of the swirling incoming air. The location of the air cleaner depends on the available space and the hood design.

An air filter performs the following functions:

(i) It filters incoming air and removes impurities present in it.

(ii) During back firing of the engine, it acts as a flame arrester.

(iii) It acts as a silencer for the carburetion system by reducing the engine induction noise.

The filters used may be classified depending on the dust type and dust concentrations in the air.

(i) *Oil impingement type*

It consists of a frame filled with crimped (pressed into small folds or corrugated) wire or metal shavings (thin particles). These are coated with a special oil so that air passing through the

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frame breaks up into smaller filaments when it comes into intimate contact with the oil. The oil can seize and hold any dust particles being carried by air. The efficiency of this filter drops progressively when in service. Hence, it should be refreshed periodically by removing, washing and re-oiling.

(ii) *Oil-bath type*

In this type of cleaner, the air is swept over or through a pool of oil. The dust particles become coated to the oil. The air is then passed through the filter, which retains the oil-coated dust particles. Figure 13.25 shows an oil bath-type oil filter containing a filter element saturated with oil. Oil is contained in a separate oil pan. Air from the atmosphere enters through the circumferential gap 1 and takes a turn at corner 2. This movement leaves large particle impurities present in air. Air then passes over the filter element through the oil surface. The clean air then passes through the passage 3.

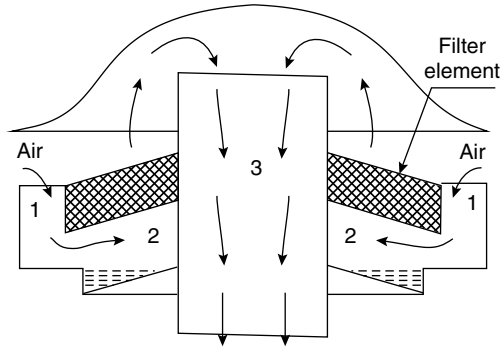


Fig. 13.25 *Oil Bath-Type Air Cleaner*

(iii) *Dry type*

It is made up of cloth, felt, glass, wool, etc. The filters catch dirt by causing it to cling to the surface of the filter material. The capacity of such filters drops progressively when they are in use. The cleaning is done based on the amount of air used by the engines and dust concentrations in it.

Figure 13.26 shows a light duty air cleaner. It consists of a bonded cylindrical cleaning element made of cellulose fibre material placed over a fine mesh screen that provides strength. The sides of the element are sealed against dust. Air passes through the element leaving any impurities present outside it.

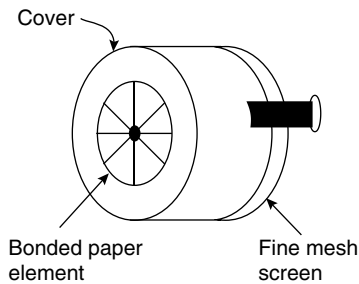


Fig. 13.26 *Light Duty Air Cleaner (Dry)*

2. *Centrifuges and oil heaters*

The complete lubrication system usually includes the following auxiliaries: Pump, oil cleaners, oil coolers, storage and sump tanks, gauges and safety devices. As oil passes through the lubrication cycle, it accumulates impurities in the form of carbon particles, water and metal scrap. For continuous reliable operation, attention should be given to oil cleaning.

For this purpose, filters with centrifuges or chemical action have been employed. Mechanical filters include cloth bags, wool, felt pads, paper discs and cartridge of porous material.

Rough cleaning of oil can be done by passing high-speed centrifuges for final cleaning. Centrifuging can be done by periodic centrifuging of the entire lubricating oil or by continuous cleaning of a small fraction of it by splitting the oil from main flow and returning back to the main stream. Oil should be heated before passing it through the centrifuge.

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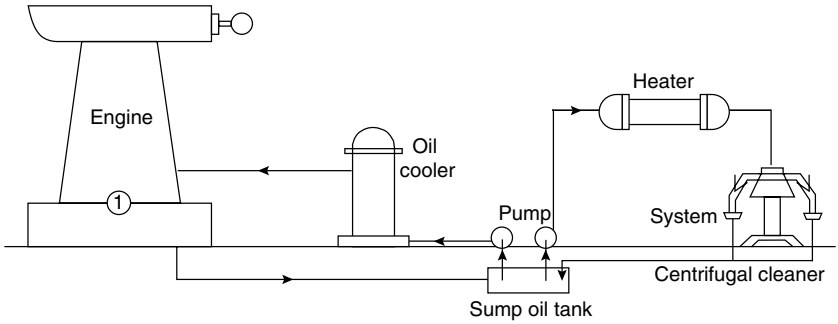


Fig. 13.27 *Continuous Centrifuging System*

Oil should be cooled before supplying it to the engine. As heat is developed due to friction between the moving parts, the cooling is done by using water from the cooling tower (Figure 13.27).

13.9 INTAKE AND EXHAUST SYSTEMS

Intake and exhaust systems used in diesel engines ensure smooth functioning of the engine. As diesel engines draw in air only, speed and power are controlled by the amount of fuel injected at the end of the compression stroke. For two-stroke diesel engines, a blower is used for induction of air and to improve scavenging. An exhaust system must reduce engine noise and discharge exhaust gases safely away from the engine. An efficient exhaust system can improve engine performance. Both these systems and their essential components are discussed in this section.

13.9.1 Intake System

The primary components of the automotive intake system are Intake manifold, throttle body and air induction components such as air cleaner and ducting.

The intake manifold is normally made of an aluminium alloy and is attached to the cylinder head. Its construction and design depends on its application. The intake manifold can accommodate a throttle body Injection unit and the mixing of the air/fuel mixture is done at the manifold base. The butterfly shaft connected to the throttle cable controls the airflow through the unit.

The air induction components consist of an air cleaner and housing, solid and flexible-duct tubing, and connectors. The air induction system draws in ambient air from the atmosphere. The inlet opening may be located in various positions under the hood.

1. *Air cleaner*

The air cleaner filters the incoming air. The air cleaner element may be manufactured from pleated paper, oil impregnated cloth or felt, or in an oil bath configuration. Another function of the air cleaner is to muffle the resonation (i.e. dampen the noise) of the swirling incoming air. The location of the air cleaner depends on the available space and the hood design.

2. *Ducting*

The ducting can be made of hardened plastic with flexible rubber couplings to absorb engine movement. These are usually secured in place by metal worm drive clamps.

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Figure 13.28 shows a typical intake system of a diesel engine plant. A large diesel engine plant requires about 0.076 m^3/min to 0.11 m^3/min of air per kW of power developed. Air contains a lot of dust, hence if it is necessary to remove the dust content in the atmospheric air. The air system contains an intake manifold located outside the building with a filter to catch dirt that would otherwise cause excessive wear in the engine. If the atmospheric temperature is too low, engine misfires at low loads and hence it is necessary to install a heating element using exhaust gas. Occasionally, engine noise may be transmitted back through the air intake system to the outside air. In such cases, a silencer is provided between the engine the intake. It is a light weight steel pipe.

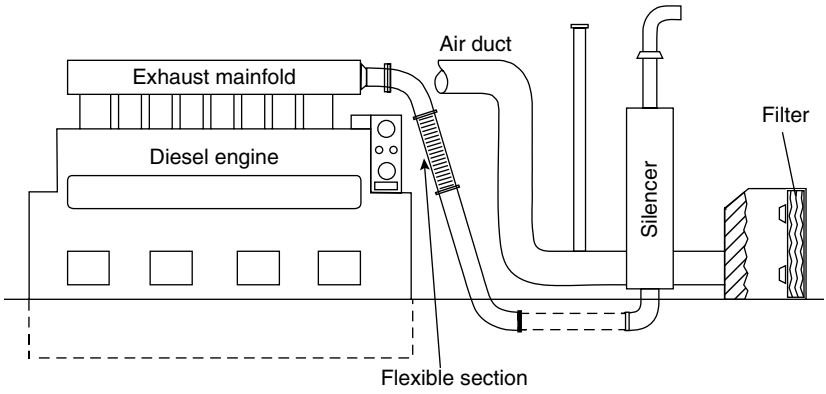


Fig. 13.28 *Engine Intake System*

13.9.2 Exhaust System

The primary components Figure 13.29 shows an engine exhaust system. The primary components of the exhaust system are exhaust manifold, engine pipe, catalytic converter, exhaust brackets, muffler and components such as the resonator and tail pipe.

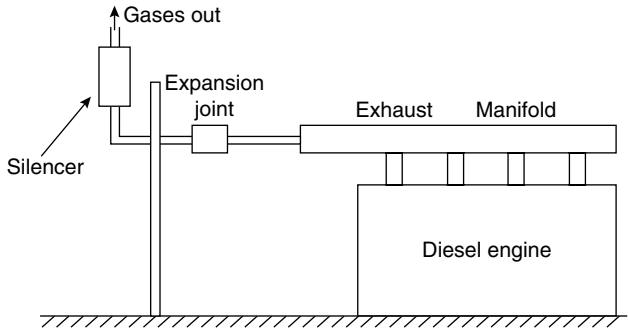


Fig. 13.29 *Engine Exhaust System*

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The exhaust system generally handles approximately 0.23 m^3/min to 0.30 m^3/min of gases per kW developed at the average exhaust temperature. Muffling of the exhaust noise is done by using silencers located outside the building. They may be of CI, steel, etc. A pipe or stack will extend vertically from the silencer outlet to carry the exhaust gases.

The following provisions should be made for the exhaust system:

(i) Silencing of the exhaust noise to the required degree.

(ii) Discharge of the exhaust sufficiently high above the ground level.

(iii) Water-cooled exhaust lines or special high-temperature materials for exhaust pipes.

(iv) Modification to utilize the by-product heat.

(v) Isolation of engine vibration from building and muffler system by using flexible section of the exhaust pipe.

(vi) Arrangement of the exhaust system to minimize the back pressure created by the exhaust system itself.

13.10 APPLICATION OF DIESEL POWER PLANT, ADVANTAGES AND DISADVANTAGES

The diesel electric power plants mainly find application in the following fields:

1. *Peak load plant*

The diesel plants are used in combination with thermal or hydro-electric plants as peak load plants. This plant is used only during peak load plant operation, as it can be started quickly and it has no standby losses.

2. *Mobile plants or outdoor units*

Mobile diesel plants mounted on skids or trailers can be used for temporary or emergency purposes such as for supplying power to large civil engineering works for supplementing electricity supply systems that are temporarily short of power. Mobile plants also find application in various other industries such as film, etc.

3. *Standby units*

Diesel plants are also used as a standby unit to supply part load when required. For example, this can be used with hydro-plant as a standby unit. If the water available is not sufficient due to reduced rainfall, a diesel station supplies power in parallel with hydroelectric station. The use is made temporarily till the water is available to take the full load.

4. *Emergency plant*

The plants can be used for emergency purposes where they act as standby units, normally idle. Whenever there is power interruption, these units are used. For example, they are used in situations when power failure may lead to financial loss or danger in key industrial processes, tunnel lighting and operating rooms of hospitals. They are also used for telecommunication and water supply under emergency conditions.

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5. *Nursery station*

A 'Nursery Station' is used to supply power to a small town in the absence of main grid and which can be moved to another area once power from the main grid is available. This happens especially when the area is remote, not approachable or the extension of grid is not possible due to constructional difficulties.

6. *Starting stations*

The diesel units are used to run the auxiliaries for starting the large steam plants.

7. *Central stations*

Small supply central stations are used for commercial purposes and public utilities, for example cinema hall, hospital and municipalities, etc. The station capacity is generally small (5–10 MW). Before installation of such units, a thorough study is made on the cost of the plant and local conditions regarding the availability of fuel and water, space requirements and non-availability of the grid.

Advantages and disadvantages of diesel plants are as follows:

(i) It has no standby losses and can burn a fairly wide range of fuels.

(ii) Manufacturing periods are short; therefore, plant expansion can be done just by adding the generating units of required size.

(iii) Diesel plants can be fully automated with reasonably low capital costs, namely, for starting, synchronizing and shutting down diesel sets, to overcome scarcity of skilled labour.

(iv) Maintenance can be simplified by the provision of easily replaceable assemblies of parts, thus enabling reconditioning to be undertaken away from generating plants.

(v) Can be installed in remote, cramped or difficult-to-access areas as the unit is compact, light weight and available in smaller capacity.

(vi) The diesel plants can be located very near to the load centres, many times in the heart of the town.

(vii) They can be employed in all climatic zones and they are very adaptable and can be easily extended to the given power requirements.
Machine sets are readily available as standard sets in the range of 500 kW to 40 MW.

(viii) Due to its relatively low cost, it is possible to install a spare unit that can be kept as permanent standby.

(ix) Operational reliability with high availability is ensured.

Disadvantages

(i) The diesel plants are not economical where fuel has to be imported.

(ii) The noise is a serious problem in diesel plant.

(iii) Selected types of fuels are required in diesel engines whereas there is more mobility in case of thermal plant.

(iv) The lubrication cost is high.

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13.10.1 Comparison with Stream Power Plants

(i) The diesel power plants are more efficient than steam power plants in the range of 150 MW capacity. The operating efficiency is high in the load range of 50–100 per cent of full load.

(ii) The first cost diesel plants are lesser than steam plants for plant capacity up to about 7 kW. Above this capacity, the diesel cost rises rapidly while that of steam plants continues to fall.

(iii) It can be quickly started up and brought into service (within 1 min).

(iv) The cooling water required for the same capacity is considerably less in diesel plants than the thermal plant. The site of the thermal plant is dictated by the availability of cooling water in many cases, whereas diesel plant can be located without giving much importance to the availability of cooling water. The water ratio for thermal to diesel is approximately 50:1.

(v) The storage required for the fuel is considerably less than the thermal and it can be handled more easily.

(vi) There is no problem of ash handling as there is practically no refuse.

(vii) The space required for diesel plant is considerably less than thermal plant and, therefore, cost of foundation and buildings is less.

(viii) A diesel engine lubrication system is more economical compared to a steam plant and permits the use of specially compounded cylinder lubricants that provide dramatic improvement in wear rates.

(ix) A diesel engine plant layout is very simple compared with a thermal plant. They are easy to operate and control also.

(x) The unit capacity of diesel engine is considerably smaller than the thermal unit. The cost of unit increases with an increase in unit capacity for diesel plant. However, the cost of the unit goes on decreasing in case of thermal plant with an increase in unit capacity.

(xi) The repair and maintenance costs of diesel plants are generally much higher than for steam plants. These costs are more or less fixed in the case of steam plants and more or less are proportional to output in the case of diesel plants.

(xii) Life of 25–30 years is normal for thermal plant, whereas the life of diesel plant is hardly two to five years or less.

(xiii) The diesel plants are not suitable for continuous operation under overloads, whereas steam plants can work under 25 per cent overload continuously.

13.11 LAYOUT OF DIESEL PLANT

The layouts of diesel power plants for high-capacity (50 MW and above), medium-capacity (25–50 MW) and low-capacity plants are shown in Figure 13.30–13.32. Generally, in all three cases, the generating units are installed parallel with parallel centre lines. Sufficient space must be provided around various units for dismantling and repairing purposes. Fuel storage tanks may be located outside the building to avoid fire hazards. Some space should be left for future expansion. The engine room should be provided with sufficient ventilation. The plant must include many instruments for the guidance of operators to ensure reliability, economy and safety.

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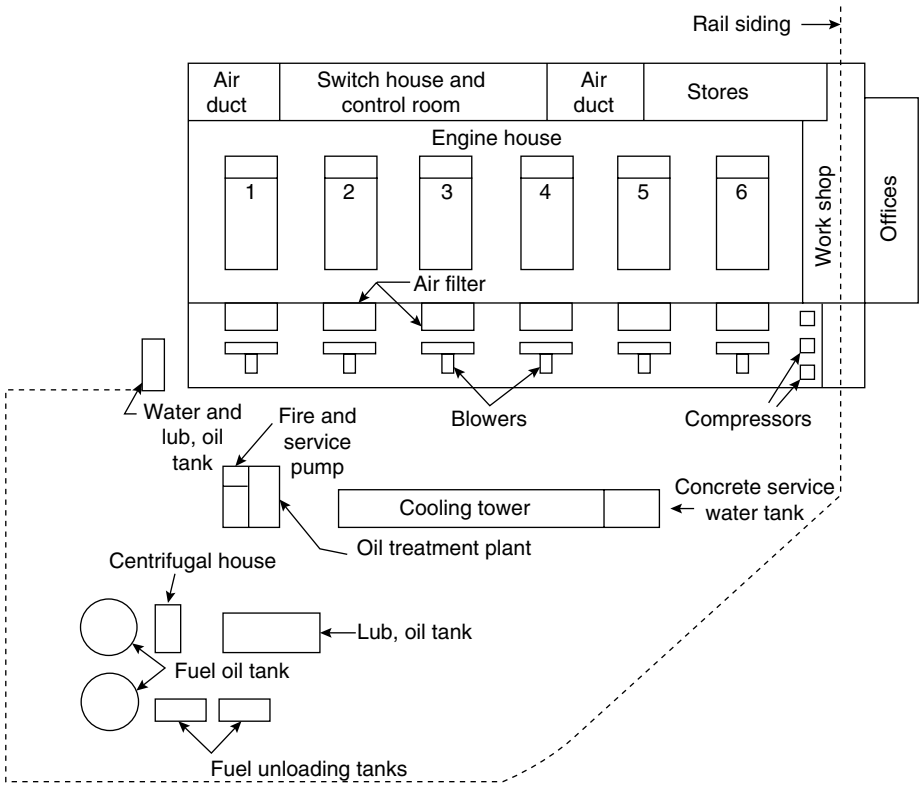


Fig. 13.30 *Layout of Diesel Power Plant for High Capacity (50 MW and above)*

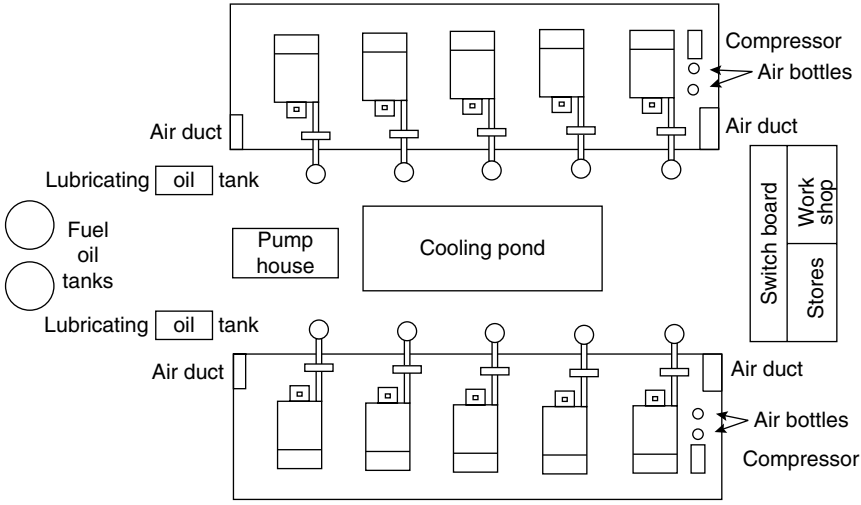


Fig. 13.31 *Layout of Diesel Power Plant for Medium Capacity (25–50 MW and above)*

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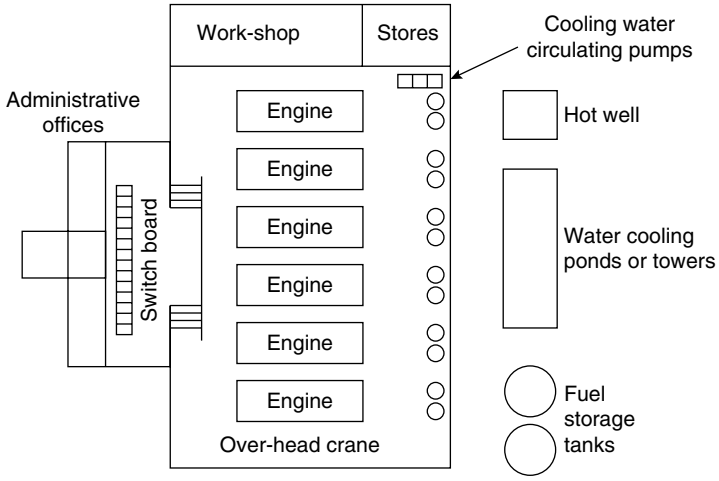


Fig. 13.32 *Layout of Diesel Power Plant for Small Capacity*

13.12 DIESEL ENGINE PERFORMANCE AND OPERATION

1. *Power and mechanical efficiency*

Power is the work done by the engine in unit time, which is the product of force and linear velocity. This can also be expressed in terms of product of engine torque and angular velocity. Torque measurement is done by a dynamometer and speed is measured using a tachometer as discussed earlier.

Power developed by the engine at the shaft output is called brake power, denoted by *BP*.

If T = torque developed, kN m
W = load applied, kN
R = radius of brake drum, m
N = engine speed, rps
then, brake power is given by

$$BP = 2\pi NT \text{ kW}$$

The total power actually developed on the pistons of the engine is called indicated power, denoted by *IP*. This indicated power is always greater than brake power, because a part of the indicated power is spent in overcoming friction of the bearings, pistons and other mechanical parts of the engine. Some of this power is also spent in induction of fuel air charge and exhaust gases delivery. Hence, the difference between the indicated power and brake power is known as friction power, denoted by *FP*.

Thus, $$IP = BP + FP$$

The ratio of power delivered by the engine (*BP*) to the total power developed within the engine (*IP*) is known as mechanical efficiency, denoted by η_m.

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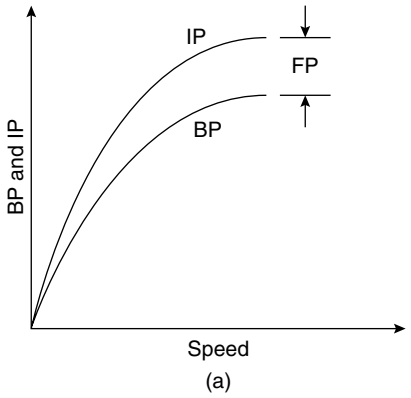


Fig. 13.33(a) *Variation of BP and IP vs. Speed*

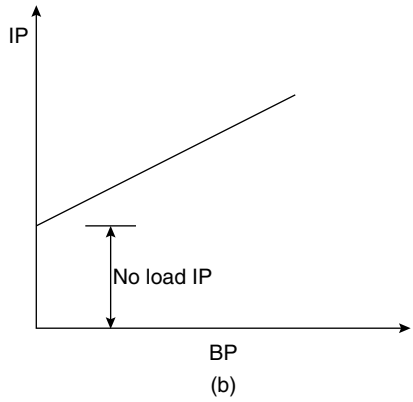


Fig. 13.33(b) *Variation of IP vs. BP*

$$\eta_m = \frac{BP}{IP} = \frac{IP - FP}{IP} = 1 - \frac{FP}{IP}$$

For a variable speed engine, both *IP* and *BP* increase as speed increases. As *BP* is lower than IP, both graphs generally curve slightly towards the top as shown in Figure 13.33.

For a constant speed engine, if the *IP* is plotted against *BP* a nearly straight line of increasing value will occur. The graph passes through the *IP* axis with a positive value, *BP* being 0.

The mechanical efficiency graph appears similar to both variable and constant speed engines as shown in Figure 13.34. Efficiency reaches a maximum at one point and then starts decreasing due to increased friction and engine overloading. For a variable speed engine, mechanical efficiency is plotted against speed and for a constant speed engine mechanical efficiency is plotted against *BP*.

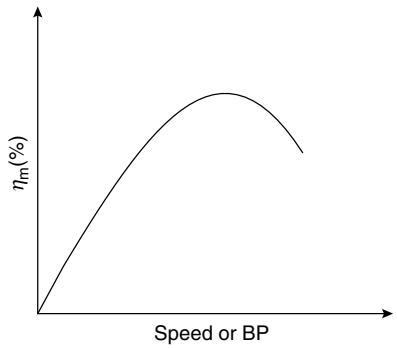


Fig. 13.34 *Variation of BP vs. Mechanical Efficiency*

2. *Mean effective pressure*

Mean effective pressure (MEP) is a hypothetical pressure acting on the piston throughout the power stroke. It is denoted by p_m.

Figure 13.35 shows a p–V diagram of a four-stroke IC engine. Positive area refers to the work output whereas negative area refers to the work input. In general, the negative area is very small compared to positive area and can be neglected. In case of a two-stroke engine, only positive area is present.

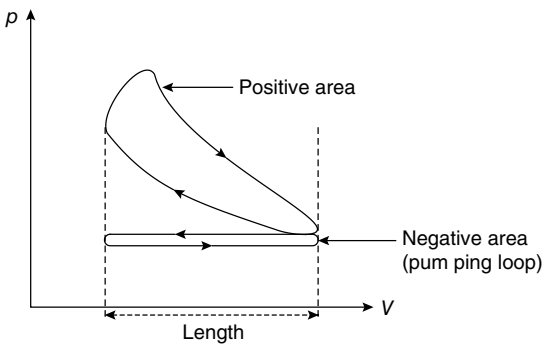


Fig. 13.35 *p–V Diagram of a Four-Stroke IC Engine*

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MEP

$$p_m = \frac{\text{Net area of the indicator diagram}}{\text{Length of the indicator diagram}} \times \text{Spring constant}$$

$$p_m = \frac{A}{l} \times S$$

Indicated MEP can be defined as the theoretical constant pressure that can be imagined to be exerted during each power stroke of the engine to produce power equal to indicated power, *IP*. It is denoted by p_{im}.

If p_{im} = indicated MEP, kN/m^2
L = stroke length, m
A = area of the piston, m^2
N = rotational speed, rps
n = number of revolutions required for each power stroke delivered per cylinder
= 2 for four-stroke engine; = 1 for two-stroke engine
x = number of cylinders in the engine

then $IP = p_{im} \times L \times A \times N \times \frac{x}{n}$ kW

Brake MEP can be defined as the theoretical constant pressure that is exerted during each power stroke of the engine to produce power equal to indicated power. It is denoted by p_{bm}.

The difference between indicated MEP and brake MEP is known as friction MEP. It is denoted by p_{fm}.

Thus,

$$p_{im} = p_{bm} + p_{fm}$$

and

$$\eta_m = \frac{p_{bm}}{p_{im}} = \frac{p_{im} - p_{fm}}{p_{im}} = 1 - \frac{p_{fm}}{p_{im}}$$

3. *Torque and MEP*

We have

$$BP = 2\pi NT = p_{bm} \times L \times A \times \frac{x}{n}$$

Hence, torque is given by

$$T = p_{bm} \times L \times A \times \frac{x}{n} \times \frac{1}{2\pi}$$

The above equation shows that torque and MEP are related by engine size. For larger engine size at constant MEP, torque increases. The parameter MEP shows how well the engine is using its size (displacement) to produce work. Hence, engines can be compared based on MEP because they cannot be compared on their relative brake powers, as brake power depends on both engine size and speed. Thus, while designing the engine, high MEP is always preferred.

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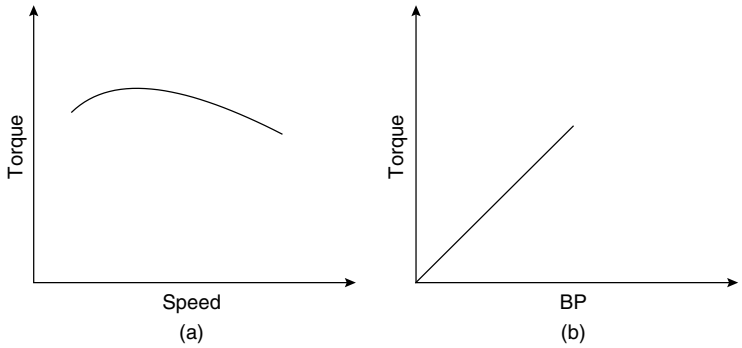


Fig. 13.36 *Torque vs. Speed and BP*

Figure 13.36 shows the variation of torque versus speed and *BP* for a variable speed and a constant speed engine. For a constant speed engine, torque increases proportionately with *BP* and the graph is a straight line.

4. *Specific fuel consumption*

Specific fuel consumption (sfc) is defined as the amount of fuel consumed per hour per unit power developed. It is denoted by sfc. It is a comparative parameter that shows how efficiency of an engine is converting fuel into work.

$$\text{Thus,} \qquad \text{sfc} = \frac{\text{Fuel consumed, kg/h}}{\text{Power developed, kW}} = \frac{m_f}{P}$$

If the specific fuel consumption is based on brake power (*BP*) developed, then it is known as brake-specific fuel consumption, bsfc.

If the specific fuel consumption is based on indicated power (*IP*), then it is known as indicated specific fuel consumption, isfc.

Figure 13.37 shows the variation of specific fuel consumption with engine speed at full throttle of an SI engine.

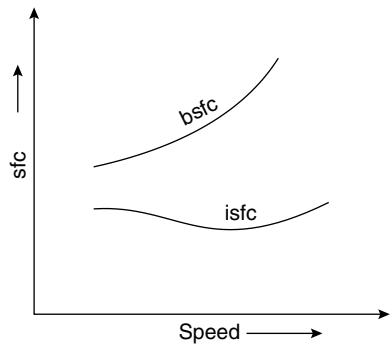


Fig. 13.37 *Variation of sfc vs. Speed*

5. *Air–fuel (AF) and fuel–air ratio (FA)*

AF ratio is defined as the ratio of mass flow rate of air to the mass flow rate of fuel.

$$AF = \frac{\text{Mass flow rate of air}}{\text{Mass flow rate of fuel}}$$

$$AF = \frac{m_a}{m_f}$$

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FA ratio is the reciprocal of *AF* ratio. It affects the phenomenon of combustion and determines the flame propagation velocity, heat released, maximum combustion temperature and completion of combustion.

Chemically correct *FA* ratio is also known as *stoichiometric FA* ratio. The ratio of actual *FA* ratio to stoichiometric *FA* ratio is known as relative *FA* ratio. Thus,

$$\text{Relative } FA \text{ ratio} = \frac{\text{Actual } FA \text{ ratio}}{\text{Stoichiometric } FA \text{ ratio}}$$

6. *Volumetric efficiency*

The volumetric efficiency of an engine is defined as the ratio of actual mass of air inducted by the engine during the intake (suction) stroke to the theoretical mass of air that should have been inducted by filling the piston displacement volume with air at atmospheric pressure and temperature. It is denoted by η_v Thus,

$$\text{Volumetric efficiency} = \eta_v = \frac{m_a}{m_t}$$

In supercharged engines, the volumetric efficiency can be lesser than or greater than unity.

7. *Specific output*

Specific output is defined as the brake power developed per unit volume of piston displacement.

$$\text{Specific output} = \frac{BP}{A \times L}$$

$$= \frac{x}{n} \times \frac{1}{2\pi} \times p_{bm} \times N$$

$$= \text{constant} \times p_{bm} \times N$$

Thus, specific output can be increased by either increasing engine speed or increasing brake MEP. The former method leads to increased mechanical stresses in engine parts whereas the latter leads to more load on the engine cylinder.

8. *Thermal efficiency*

Thermal efficiency of the engine is defined as the ratio of output to the energy supplied by the combustion of fuel. If the output is based on the indicated power, it is known as indicated thermal efficiency, and if the output is based on brake power, it is known as brake thermal efficiency. Thus,

Indicated thermal efficiency,

$$\eta_i = \frac{\text{Indicated power}}{\text{Energy supplied}}$$

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$$\eta_i = \frac{IP}{m_f \times CV_f}$$

Brake thermal efficiency,

$$\eta_b = \frac{\text{Brake power}}{\text{Energy supplied}}$$

$$\eta_b = \frac{BP}{m_f \times CV_f}$$

where m_f = mass or volume of fuel, kg/s or m^3/s and CV_f = calorific value of fuel, kJ/kg or kJ/m^3

9. *Relative efficiency*

The relative efficiency or efficiency ratio is the ratio of the actual efficiency of the engine to the theoretical efficiency of the engine.

Relative efficiency,

$$\eta_r = \frac{\eta_{actual}}{\eta_{airstandard}}$$

The variation of thermal efficiency versus engine speed is shown in Figure 13.38.

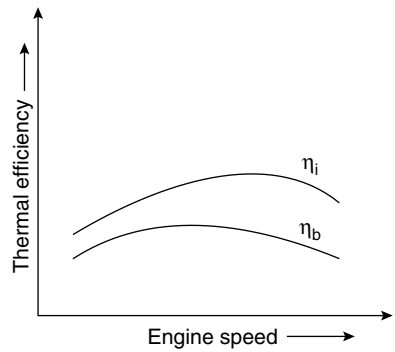


Fig. 13.38 *Variation of Thermal Efficiency versus Speed*

10. *Heat balance*

Heat balance gives an idea about the amount of heat supplied and the amount of heat utilized in the system. It gives useful information about the performance of the engine. The heat balance calculations are done generally either on minute basis or hour basis.

Heat supplied by the engine

Heat supplied by the engine by burning m_f kg of fuel is given by

$$Q_A = m_f \times CV_f$$

where CV_f is the lower calorific value of the fuel.

Heat utilized by the system

(i) Heat equivalent of $BP = BP \times 60$ kJ/min

(ii) Heat lost to cooling water

Let, m_w = Mass flow rate of water, kg/min

C_{pw} = Specific heat of water, kJ/kg K

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T_{wo} = Outlet temperature of water, °C

T_{wi} = Inlet temperature of water, °C

then heat lost to cooling water is

$$Q_w = m_w C_{pw} (T_{wo} - T_{wi})$$

(iii) Heat lost to exhaust

Let

m_g = mass flow rate of exhaust gas, kg/min

C_{pg} = specific heat of gas

T_{ge} = exhaust gas coming out of the engine, °C

T_a = ambient temperature, °C

then heat carried away by exhaust gases.

$$Q_g = m_g C_{pg} (T_{ge} - T_a) \text{ kJ/min}$$

(iv) Heat lost due to radiation, convection and leakage of gases is known as unaccounted heat.

Unaccounted heat loss = $Q_A - [(a) + (b) + (c)]$

The results are tabulated and compared on percentage basis as shown below.

Heat supplied by the combustion of fuel (kJ/min)	Percentage	Heat utilized by the system (kJ/min)	Percentage
Q_s	100	(a) Heat equivalent of *BP*	–
–	–	(b) Heat lost to cooling water	–
–	–	(c) Heat lost to exhaust gases	–
–	–	(d) Unaccounted heat	–
Total	100	–	100

Notes: As the frictional heat will be dissipated in cooling water, exhaust gases, radiation and convection, it is not necessary to include heat lost due to frictional power *FP*. Measurement of heat lost due to cooling water and the heat lost to exhaust gases are carried out as explained earlier.

Figure 13.39 shows the performance of SI engines at constant speed and variable loads. The load can be varied by altering the throttle, and the speed is kept constant by resetting the dynamometer. Closing the throttle does not affect the temperature very much as the *AF* ratio is more or less constant and the gas temperatures in the cycle are too high. But this action reduces the cylinder pressure resulting in high loss to the coolant at low engine load. This is the reason why CI engines are better at part loads than SI engines.

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At low loads, efficiency is about 9–10 per cent and rises to about 25 per cent at full load. The loss to coolant is nearly double (60 per cent) at low loads as against full load (30 per cent). The exhaust temperature varies steadily but slowly with load. As mass flow rate of fuel into the engine is reduced, mass flow rate of exhaust gas is also reduced and percentage of loss of exhaust gas remains nearly constant. However, percentage loss to radiation has almost a threefold increase at low loads as compared with full load.

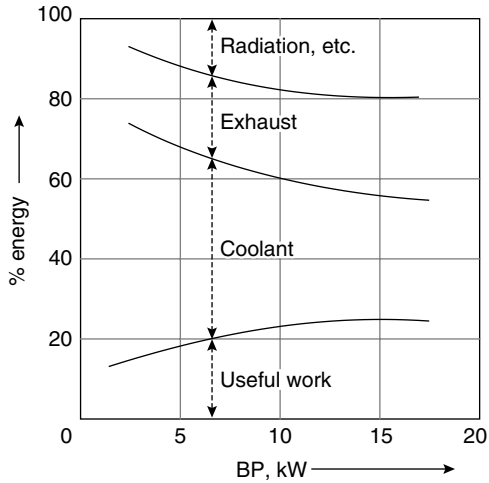


Fig. 13.39 *Performance of SI Engines at Constant Speed and Variable Loads*

Figure 13.40 shows the performance of CI engine at constant speed but at variable load. Due to the higher compression ratio and more thermal efficiency of CI engine, the losses are less. The coolant losses are more at low loads and unaccounted losses due to radiation, etc., are more at high loads. The heat to exhaust is comparatively higher compared to SI engines. However, the useful work is comparatively higher at low loads, reaches maximum and slowly decreases when load is further increased.

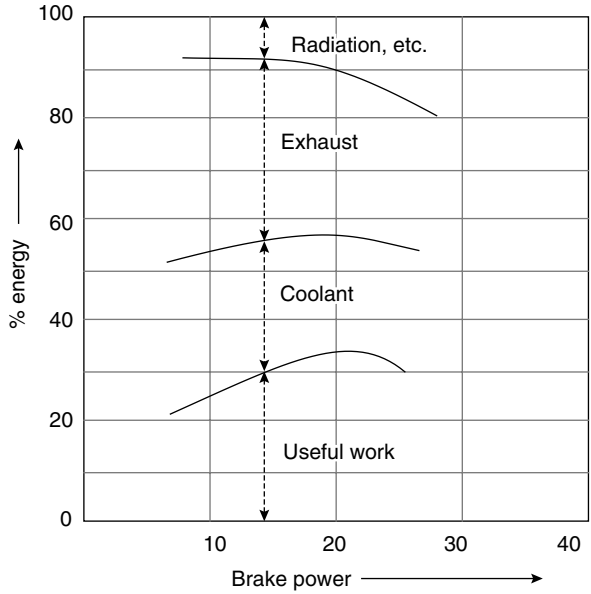


Fig. 13.40 *Performance of CI Engines at Constant Speed and Variable Loads*

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11. *Performance of diesel power plant*

Like other plants, diesel plants are also run at part load conditions. The part load performance of a diesel engine power plant such as specific fuel consumption, brake thermal efficiency and mechanical efficiency is shown in Figure 13.41. As the part load increases, engine characteristics, namely, the specific fuel consumption increases, whereas both the thermal and mechanical efficiencies decrease. However, these effects are not as predominant as in steam power plants.

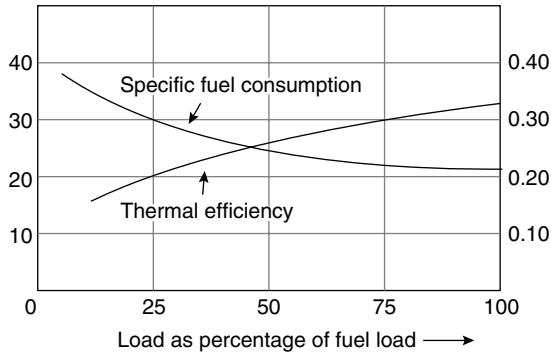


Fig. 13.41 *Performance of Diesel Power Plant*

Example 13.1

From the data given below, draw a heat balance sheet for a single-cylinder, four-stroke oil engine.

Cylinder bore = 200 mm
Stroke = 350 mm
Gross mean effective pressure = 7 bar
Pumping mean effective pressure = 0.4 bar
Engine speed = 5 rev/s
Net brake load = 370 N
Brake drum radius = 1.2 m
Fuel used = 3 kg/h
Gross calorific value of oil = 44,000 kJ/kg
Mass of cooling water = 4 kg/min
Rise in temperature of cooling water = 36°C
Mass of air supplied = 1.4 kg/min
Temperature of exhaust gasses = 320°C
Room temperature = 17°C
Mean specific heat of dry gases = 0.75 kJ/kg K
Mass of exhaust gasses = 2 kg/min

Solution: Net mean effective pressure = Gross MEP – Pumping MEP

$$= 7 - 0.4$$

$$= 6.6 \text{ bar}$$

Area of cylinder, $A = \frac{\pi}{4} D^2$

$$= 0.0314 \text{ m}^2$$

Indicated power, $IP = \frac{\pi}{4} D^2 ; x = 1; n = 2$

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$$= 6.6 \times 10^2 \times 0.35 \times 0.0314 \times \frac{5 \times 1}{2}$$

$$= 18.13 \text{ kW}$$

Brake power $BP = 2\pi NT$

$$= 2\pi \times 5 \times 370 \times 10^{-3} \times 1.2$$

$$= 13.95 \text{ kW}$$

Heat in fuel supplied/minute $Q_A = m_f \times CV_f$

$$= \frac{3}{60} \times 44,000 = 2200 \text{ kJ}$$

Heat equivalent to brake power/minute

$$= BP \times 60 = 13.95 \times 60$$

$$= 837 \text{ kJ}$$

Heat equivalent to indicated power/minute

$$= IP \times 60 = 18.13 \times 60$$

$$= 1087.8 \text{ kJ}$$

Heat in friction/minute

$$= IP - BP = 1087.8 - 837$$

$$= 250.8 \text{ kJ}$$

Heat lost to jacket cooling water/minute

$$Q_w = m_w C_{pw} \Delta T$$

$$= 4 \times 4.186 \times 36$$

$$= 602.78 \text{ kJ}$$

Heat lost to dry exhaust gases/minute

$$Q_g = m_g \times C_{pg} \times (T_{ge} - T_a)$$

$$= 2 \times 0.75 \ (320 - 17)$$

$$= 454.5 \text{ kJ}$$

Unaccounted heat due to radiation, etc.

$$= Q_A - (\text{heat equivalent of BP} + Q_w + Q_g)$$

$$= 2200 - (837 + 602.78 + 454.5)$$

$$= 305.72 \text{ kJ/min}$$

The heat balance sheet is drawn by knowing the amount of heat supplied and the amount of heat utilized by the system.

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Heat supplied	Q_A kJ/min	Percentage	Heat utilized	kJ/min	Percentage
Heat given by fuel	2200	100	(a) Heat equivalent of *BP*	837.0	38.0
–	–	–	(b) Heat lost to cooling water	602.78	27.4
–	–	–	(c) Heat lost to exhaust gases	454.5	20.7
–	–	–	(d) Unaccounted heat loss due to radiation	305.72	13.9
Total	Q_A	100	Qutilized	2200	100

Example 13.2

During a test on a four-stroke cycle oil engine the following data and results were obtained.

Speed of the engine = 360 rpm
Swept volume of cylinder = 12 liters
Effective brake load = 70 kg
Effective brake radius = 0.6 m
Mean height of indicator diagram = 20 mm
Indicator calibration = 28 kN/m^2/mm
Fuel consumption = 0.0016 kg/s
Calorific value of fuel = 44,000 kJ/kg
Cooling water circulation = 0.10 kg/s
Cooling water outlet temperature = 68°C
Cooling water inlet temperature = 35°C
Specific heat capacity of water = 4.18 kJ/kg K
Energy to exhaust gases = 34 kJ/s

Determine the indicated and brake outputs and the mechanical efficiency. Draw up an overall energy balance in kJ/s and as a percentage.

Solution: Indicated mean effective pressure is given by

$$p_{im} = 20 \times 28 = 560 \text{ kN/m}^2$$

Hence, indicated power

$$IP = p_{im}\, LAN \times \frac{x}{n};\ x = 1,\ n = 2$$

Now,

swept volume of the cylinder = $V_S = LA$

$$= 12\ l$$

$$= 12 \times 10^{-3} \text{ m}^3$$

$$\text{Indicated power} = 560 \times 12 \times 10^{-3} \times \frac{360 \times 1}{60 \times 2}$$

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$$= 20.16 \text{ kW}$$

$$\text{Brake power} = 2\pi NT$$

$$= (2\pi \times 6 \times 9.81 \times 70 \times 0.6) \times 10^{-3}$$

$$= 15.53 \text{ kW}$$

$$\text{Mechanical efficiency} = \frac{BP}{IP} \times 100 = \frac{15.53}{20.16} \times 100 = 77.03\%$$

$$\text{Energy from fuel} = \text{Fuel consumption} \times \text{Calorific value of fuel}$$

$$= 0.0016 \times 44{,}000$$

$$= 70.4 \text{ kJ/s}$$

$$\text{Energy to brake power} = 15.53 \text{ kW} = 15.53 \text{ kJ/s}$$

$$\text{Energy to coolant} = m_w \times C_{pw} (T_{wo} - T_{wi})$$

$$= 0.10 \times 4.186 (68 - 35)$$

$$= 13.81 \text{ kJ/s}$$

$$\text{Energy to exhaust (given)} = 34 \text{ kJ/s}$$

$$\text{Energy to surroundings, etc.} = \text{Energy from fuel} - [\text{Energy to } BP + \text{Energy to coolant} + \text{Energy to exhaust}]$$

$$= 70.4 - (15.53 + 13.81 + 34)$$

$$= 7.061 \text{ kJ/s}$$

Energy balance can be tabulated

	kJ/s	Percentage
Energy from fuel	70.4	100.00
Energy to brake power	15.53	22.06
Energy to coolant	13.81	19.62
Energy to exhaust	34.0	48.29
Energy to surroundings, etc.	7.06	10.03

Example 13.3

In a trial of a four-cylinder, four-stroke petrol engine of 90 mm bore and 100 mm stroke, the net dynamometer load was 1400 N at a radius of 460 mm, when the speed was 2200 rpm. At the same speed and throttle opening, the engine required 4 kW to motor it with the ignition switched off.

(i) Calculate the mechanical efficiency and indicated mean effective pressure.

(ii) During a 3 min run at this speed and power, the engine used 2.4 kg of petrol of calorific value 42,980 kJ/kg and 100 kg of cooling water with a temperature rise of 40°C. Draw up a heat balance for the test in kJ/min.

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Solution:

(i) Brake Power

Brake power, $BP = 2\pi NT = 2\pi \times \frac{2200}{60} \times 1400 \times 0.46 \times 10^{-3}$

$BP = 148.37$ kW

Frictional power, $FP = 4$ kW

Indicated power = Brake power + Frictional power

$= 148.37 + 4 = 152.37$ kW

$$\text{Mechanical efficiency} = \frac{BP}{IP}$$

$$= \frac{148.37}{152.37} \times 100 = 97.37\%$$

Indicated mean effective pressure

$$p_{\text{im}} = \frac{\text{Indicated power}}{AL\frac{N}{2}} = \frac{152.37 \times 10^3}{\frac{\pi}{4}(0.09)^2 \times 0.1 \times \frac{2200}{2 \times 60} \times 4 \times 10^5}$$

$= 32.66$ bar

(ii) Heat Input = Fuel consumption × Calorific value of fuel

$$= \frac{2.4}{3} \times 42,980$$

$= 34,384$ kJ/min

Heat to jacket cooling water

$$= m_w C_{pw} \Delta T = \frac{100}{3} \times 4186 \times 40$$

$= 5581.33$ kJ/min

Heat equivalent to $BP = 148.37 \times 60 = 8902.2$ kJ/min

Heat equivalent to $IP = 152.37 \times 60 = 9142.2$ kJ/min

Heat equivalent to $FP = 4 \times 60 = 240$ kJ/min

Heat to exhaust and radiation, etc. = 10,518.27 kJ/min

Heat balance for 1 min

	kJ/min	Percentage
Heat input	34,384	100
Heat to brake power	8902.2	25.9
Heat to jacket water	5581.33	16.2
Heat to indicated power	9142.2	26.6
Heat to friction power	240.0	0.70
Heat to exhaust and radiation	10,518.27	30.60

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Example 13.4

A six-cylinder four-stroke diesel engine of 34 cm bore and 38 cm stroke gave the following results during testing.

BP = 142 kW; N = 350 rpm; p_m = 3.7 bar

m_f = 44 kg/h; $(CV)_f$ = 44,800 kJ/kg; m_a = 38 kg/min

m_w = 60 kg/min; and cooling water temperature rise = 41°C

Piston cooling oil = 35 kg/min, C_p of oil = 2.1 kJ/kg K

Rise in cooling oil temperature = 28°C

Exhaust gas temperature = 190°C, ambient temperature = 20°C

Fuel contains 14 per cent H_2 by mass and C_{pg} = 1.05 kJ/kg K

Partial pressure of water vapour carried in exhaust gases = 0.06 bar

Draw the heat balance sheet on minute basis and percentage basis. Find the specific fuel consumption at full load assuming mechanical efficiency as 0.6.

Solution: Indicated power of the engine,

$$IP = p_{im} \times L \times A \times N \times \frac{x}{n}$$

$$= 3.7 \times 100 \times 0.38 \times \frac{\pi}{4} \times 0.34^2 \times \frac{350}{60} \times \frac{6}{2}$$

$$= 223.4 \text{ kW}$$

Heat supplied = $m_f \times CV_f = \frac{44}{60} \times 44{,}800 = 32{,}853.33$ kJ/min

Heat equivalent of $BP = 142 \times 60$

$$BP = 8520 \text{ kJ/min}$$

Heat lost to cooling water,

$$Q_w = m_w \times C_{pw} \times \Delta T$$

$$= 60 \times 4.186 \times 41$$

$$Q_w = 10{,}297.56 \text{ kJ/min}$$

Heat lost to cooling oil,

$$Q_o = (m \times C_p \times \Delta T) \text{ oil}$$

$$= 35 \times 2.1 \times 28$$

$$Q_o = 2058 \text{ kJ/min}$$

Heat lost to dry exhaust gases

Mass of exhaust gases,

$$m = m_a + m_f$$

$$= 38 + \frac{44}{60} = 38.73 \text{ kg/min}$$

Mass of water vapour formed

$$m_{H_2O} = m_f \times 9 \times H_2$$

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$$= \frac{44}{60} \times 9 \times 0.14 \text{ (1 kg of } H_2 \text{ forms 9 kg of } H_2O$$

$$= 0.924 \text{ kg/min}$$

Mass of dry exhaust gases

$$m_g = m - m_{H_2O}$$
$$= 38.73 - 0.924$$
$$= 37.806 \text{ kg/min}$$

Hence, heat lost to exhaust gases

$$Q_g = m_g \times C_{pg} \times (T_{ge} - T_a)$$
$$= 37.806 \times 1.005 \times (190 - 20)$$
$$Q_g = 6459.16 \text{ kJ/min}$$

Heat carried away by water vapour

From steam tables at 0.06 bar, 190°C

$$h = 2860 \text{ kJ/kg}$$

Heat carried away by water vapour

$$Q_{wv} = 0.924 \times 2860$$
$$Q_{wv} = 2642.64 \text{ kJ/min}$$

The heat balance sheet is prepared now on minute and percentage basis

Heat input	kJ/min	Percentage	Heat output	kJ/min	Percentage
Heat supplied	33,853.33	100	(a) Heat equivalent to BP	8520.00	25.17
–	–	–	(b) Heat to cooling water	10,297.56	30.42
–	–	–	(c) Heat to cooling oil	2058.00	06.08
–	–	–	(d) Heat to dry exhaust gases	6459.16	19.08
–	–	–	(e) Heat to water vapour	2642.64	07.81
–	–	–	(f) Unaccounted heat loss	3876.21	11.44
Total	33,853.33	100	–	33,853.33	100

Now, indicated thermal efficiency

$$= \frac{IP}{m_f \times CV_f}$$
$$= \frac{223.4 \times 60}{33,853.33}$$
$$= 0.396$$
$$= 39.6\%$$

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Mechanical efficiency and brake thermal efficiency are same at full load also.

BP at full load $= 2 \times 142 = 284$ kW

At half-load condition

$$FP = IP - BP$$
$$= 223.4 - 142$$
$$= 81.4 \text{ kW}$$

At full-load condition

$$FP = IP - BP$$
$$IP = BP + FP$$
$$= 284 + 81.4 = 365.4 \text{ kW}$$

Hence, heat supplied

$$= \frac{IP}{\eta_i} = \frac{365.4 \times 60}{0.396}$$
$$= 55{,}363.64 \text{ kJ/min}$$

Hence, specific fuel consumption at full load

$$\text{bsfc} = \frac{\text{Mass of fuel}}{BP}$$
$$= 0.261 \text{ kg/kWh}$$

Example 13.5

The following observations were made during one hour test on a single-cylinder, four-stroke oil engine. Bore = 300 mm; stroke = 450 mm; mass of fuel used = 8.8 kg; calorific value = 41,800 kJ/kg; average speed = 200 rpm; mean effective pressure = 5.8 bar; brake load = 1860 N; mass of cooling water circulated = 650 kg; temperature rise = 22°C; diameter of brake drum = 1.22 m. Calculate (i) mechanical efficiency and (ii) brake thermal efficiency and (iii) draw the heat balance sheet.

Solution:

(i) Mechanical efficiency

Indicated power,

$$IP = p_m \times L \times A \times N \times; x = 1; \frac{x}{n} \; n = 2$$
$$= 5.8 \times 100 \times 0.45 \times (\pi/4) \times 0.32 \times \frac{200}{60} \times \frac{1}{2}$$
$$= 30.748 \text{ kW}$$

Brake power,

$$BP = 2\pi NT$$
$$= 2\pi NW \times R$$

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$$= 2 \times \pi \times \frac{200}{60} \times 1860 \times \frac{1.22}{2} \times 10^{-3}$$

$$= 23.763 \text{ kW}$$

Mechanical efficiency,

$$\eta_m = \frac{BP}{IP} \times 100 = \frac{23.763}{30.748} \times 100$$

$$= 77.283\%$$

(ii) Brake thermal efficiency

$$\eta_{Bth} = \frac{BP}{m_f \times CV_f} = \frac{23.763}{(8.8/3600) \times 41,800}$$

$$= 0.2326 \text{ or } 23.26\%$$

(iii) Heat balance sheet

Heat input from fuel

$$= m_f \times CV_f$$

$$= 8.8 \times 41,800$$

$$= 3,67,840 \text{ kJ/h}$$

Heat equivalent of *BP*

$$= BP \times 3600 = 23.763 \times 3600$$

$$= 85,546.8 \text{ kJ/h}$$

Heat lost to cooling water

$$Q_w = m_w \times C_{pw} \times \Delta T$$

$$= 650 \times 4.186 \times 22$$

$$= 59,859.8 \text{ kJ/h}$$

Heat lost to exhaust, radiation, etc.

$$= 3,67,840 - (85,546.8 + 59,859.8)$$

$$= 2,22,433.4 \text{ kJ/h}$$

Heat balance sheet on hour basis

Heat input	kJ/h	Percentage	Heat output	kJ/h	Percentage
Heat supplied	3,67,840	100	(a) Heat equivalent to *BP*	85,546.8	23.26
–	–	–	(b) Heat lost to cooling water	59,859.8	16.27
–	–	–	(c) Unaccounted heat loss	2,22,433.4	60.47
Total	3,67,840	100	Total	36,784.0	100

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Example 13.6

During a test on a single-cylinder, four-stroke oil engine, the following observations are made: Bore = 30 cm; stroke = 45 cm; duration of trial = 1 h, total fuel consumption = 7.6 kg; calorific value of fuel = 45,000 kJ/kg, total revolutions made = 12,000; mean effective pressure = 6 bar, net brake load = 1.47 kN; brake drum diameter = 1.8 m, rope diameter = 3 cm; mass of jacket cooling water circulated = 550 kg; water enters at 15°C and leaves at 60°C; total air consumption = 360 kg; room temperature = 20°C; exhaust gas temperature = 300°C. Calculate (i) indicated and brake power and (ii) mechanical efficiency and (iii) draw the heat balance sheet on minute basis.

Solution:

(i) Indicated and brake power

Indicated power, $IP = p_m \times L \times A \times N \times \frac{x}{n}$; $x = 1$; $n = 2$

$$= 6 \times 100 \times 0.45 \times (\pi/4) \times 0.3^2 \times \frac{12,000}{3600} \times \frac{1}{2}$$

$$IP = 31.81 \text{ kW}$$

Brake power,

$$BP = 2\pi NT \left(T = W \times R \text{ and } R = \frac{D+d}{2} \right)$$

$$= 2 \times \pi \times \frac{12,000}{3600} \times 1.47 \times \left(\frac{1.8+0.03}{2} \right)$$

$$BP = 28.17 \text{ kW}$$

(ii) Mechanical efficiency

$$\eta_m = \frac{BP}{IP} \times 100 = \frac{28.17}{31.81} \times 100$$

$$\eta_m = 88.56\%$$

(iii) Heat balance sheet on minute basis

Heat supplied $= m_f \times CV_f$

$$= \frac{7.6}{60} \times 45,000$$

$$= 5700 \text{ kJ/min}$$

Heat to BP $= BP \times 60$

$$= 28.17 \times 60$$

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$$= 1690.2 \text{ kJ/min}$$

Heat to cooling water

$$Q_w = m_w \times C_{pw} \times \Delta T = \frac{550}{60} \times 4.186 \times (60 - 15)$$

$$= 1726.73 \text{ kJ/min}$$

Heat carried away by exhaust gases

$$Q_g = m_g \times C_{Pg} \times (T_g - T_a) \; (m_g = m_f + m_a)$$

$$= \left(\frac{7.6 + 360}{60}\right) \times 1.25 \times (300 - 20)$$

$$= 2144.33 \text{ kJ/min}$$

Unaccounted heat loss,

$$= 5700 - (1690.2 + 1726.73 + 2144 - 33)$$

$$= 138.74 \text{ kJ/min}$$

Heat balance sheet on minute basis

Heat input	kJ/h	Percentage	Heat output	kJ/h	Percentage
Heat supplied	5700	100	Heat to *BP*	1690.20	29.65
–	–	–	Heat to cooling water	1726.73	30.29
–	–	–	Heat to exhaust gases	2144.33	37.62
–	–	–	Unaccounted heat loss	138.74	2.44
Total	5700	100	Total	5700.00	100

Example 13.7

A full-load test on a two-stroke engine gave the following data: speed = 450 rpm; brake load = 460 N; indicated MEP = 3 bar; fuel consumption = 5.4 kg/h; jacket water flow rate = 440 kg/h; temperature rise of cooling water = 36°C; A:F = 13; temperature of exhaust gases = 355°C; room temperature = 20°C, *CV* of fuel = 42,000 kJ/kg; cylinder bore = 200 mm; stroke = 270 mm; brake drum diameter = 1500 mm; mean specific heat of exhaust gases = 1.02 kJ/kg K. Determine the indicated thermal efficiency and draw heat balance sheet (VTU, August 2001).

Solution:

(i) Indicated thermal efficiency

Indicated power, $IP = p_m \times L \times A \times N \times \frac{x}{n}$; $x = 1$; $n = 1$

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$$= 3 \times 100 \times 0.270 \times (\pi/4)\ 0.2^2 \times \frac{450}{60} \times \frac{1}{1}$$

$$= 19.085 \text{ kW}$$

Brake power,

$$BP = 2\pi\, NT$$

$$= 2\pi \times \frac{450}{60} \times 460 \times 0.750 \times 10^{-3}$$

$$= 16.26 \text{ kW}$$

Indicated thermal efficiency

$$\eta_{\text{ith}} = \frac{IP}{m_{\text{f}} \times CV_{\text{f}}} = \frac{19.085}{(5.4/3600) \times 42,000}$$

$$\eta_{\text{ith}} = 0.3029 \text{ or } 30.29\%$$

(ii) Heat balance sheet

Heat input $= m_{\text{f}} \times CV_{\text{f}}$

$= 5.4 \times 42,000$

$= 2,26,800$ kJ/h

Heat to BP $= BP \times 3600$

$= 16.26 \times 3600$

$= 58,536$ kJ/h

Heat to cooling water

$$Q_{\text{w}} = m_{\text{w}} \times C_{\text{PW}} \times \Delta T$$

$$= 440 \times 4.186 \times 36$$

$$= 66,306.24 \text{ kJ/h}$$

Heat to exhaust gases

$$Q_{\text{g}} = m_{\text{g}} \times C_{\text{Pg}} \times \Delta T = m_{\text{f}}\left(\frac{m_{\text{a}}}{m_{\text{f}}} + 1\right) \times C_{\text{Pg}} \times \Delta T$$

$$= 5.4\left(\frac{13}{1} + 1\right) \times 1.02 \times (355 - 20)$$

$$= 25,832.52 \text{ kJ/h}$$

Unaccounted heat loss

$$= 2,26,800 - (58,536 + 66,306.24 + 25,832.32)$$

$$= 76,125.44 \text{ kJ/h}$$

Heat balance sheet on hourly basis

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Heat input	kJ/h	Percentage	Heat output	kJ/h	Percentage
Heat supplied	2,26,800	100	Heat to *BP*	58,536.00	25.81
–	–	–	Heat to cooling water	66,306.24	29.24
–	–	–	Heat to exhaust gases	25,832.32	11.39
–	–	–	Unaccounted heat	76,125.44	33.56
Total	2,26,800	100	Total	2,26,800	100

Example 13.8

A two-stroke diesel engine was motored when meter reading was 1.5 kW. Test on the engine was carried out for one hour and data observed were brake torque = 120 Nm; rpm = 600; fuel used 2.5 kg; cooling water = 818 kg; *CV* of fuel = 40.3 MJ/kg; rise in temperature of cooling water = 10°C; room temperature = 27°C; A:F used = 32:1; exhaust gas temperature = 347°C; C_p for exhaust gases = 1.05 kJ/kg K. Determine brake power, indicated power, mechanical efficiency and thermal efficiency. Draw the heat balance sheet on minute and percentage basis.

Solution: Brake power, $BP = 2\pi NT$

$$= 2\pi \times \frac{600}{6} \times 120 \times 10^{-3}$$

$$= 7.54 \text{ kW}$$

Indicated power, $IP = BP + FP$

$$= 7.54 + 1.5$$

$$= 9.04 \text{ kW}$$

Mechanical efficiency,

$$\eta_m = \frac{BP}{IP} \times 100$$

$$= \frac{7.54}{9.04} \times 100$$

$$= 83.41\%$$

Thermal efficiency,

$$\eta_{Bth} = \frac{BP}{m_f \times CV_f} = \frac{7.54}{(2.5/3600) \times 40.3 \times 10^3}$$

$$= 26.94\%$$

Heat balance sheet

Heat input $= m_f \times CV_f$

$$= \frac{2.5}{60} \times 40.3 \times 10^3$$

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$$= 1679.17 \text{ kJ/min}$$

Heat to BP $= BP \times 60$

$$= 7.54 \times 60$$

$$= 452.4 \text{ kJ/min}$$

Heat lost to cooling water

$$Q_w = m_w \times C_{PW} \times \Delta T$$

$$= \frac{818}{60} \times 4.186 \times 10$$

$$= 570.69 \text{ kJ/min}$$

Heat to exhaust gases,

$$Q_g = m_g \times C_{pg} (T_y - T_a)$$

$$= m_f \left(\frac{m_a}{m_f} + 1 \right) \times 1.05 \ (347 - 27)$$

$$= \frac{2.5}{60} \times (32 + 1) \times 1.05 \times 320$$

$$= 462 \text{ kJ/min}$$

Unaccounted heat loss

$$= 1679.17 - (452.4 + 570.69 + 462)$$

$$= 194.08 \text{ kJ/min}$$

Heat input	kJ/min	Percentage	Heat output	kJ/min	Percentage
Heat supplied	1679.17	100	Heat to BP	452.40	26.94
–	–	–	Heat to cooling water	570.69	33.99
–	–	–	Heat to exhaust gases	462.00	27.51
–	–	–	Unaccounted heat	194.08	11.56
Total	1679.17	100	Total	1679.17	100

Example 13.9

After a test on a single-cylinder, four stroke oil engine, the following data were recorded:

Cylinder bore = 150 Nm

Stroke = 250 mm

Area of indicator diagram = 450 sq.mm

Length of indicator diagram = 50 mm

Indicator spring rating = 1.2 mm for a pressure of 9.806 N/cm^2

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Engine speed = 400 rpm
Brake torque = 22.5 Nm
Fuel consumption = 3 kg/h
Calorific value of fuel = 44,200 kJ/kg
Cooling water flow rate = 4 kg/min
Rise of temperature for cooling water = 42°C

Compute (i) Mechanical efficiency
(ii) Brake thermal efficiency
(iii) Specific fuel consumption
(iv) Heat balance in kW

Solution:

(i) Mechanical efficiency

Brake power, $BP = 2\pi NT = 2\pi \times \frac{400}{6} \times 22.5 \times 10^{-3}$

$= 9.425$ kW

Mean effective pressure

$$p_m = \frac{A}{L} \times S = \frac{450}{50} \times \frac{9.806}{1.2} \times 10^4$$

$= 7.354$ bar

Indicated power, $IP = p_m \times L \times A \times N \times \frac{x}{n}; x = 1; n = 2$

$$= 7.354 \times 100 \times 0.25 \times (\pi/4) \times 0.15^2 \times \frac{400}{60} \times \frac{1}{2}$$

$= 10.83$ kW

(i) Mechanical efficiency

$$\eta_m = \frac{BP}{IP} \times 100 = \frac{9.425}{10.83} \times 100$$

$= 87.03\%$

(ii) Brake thermal efficiency

$$\eta_{Bth} = \frac{BP}{m_f \times CV_f} = \frac{9.425}{(3/3600) \times 44,200}$$

$= 0.2559$ or 25.59%

(iii) Specific fuel consumption

$$\text{bsfc} = \left(\frac{m_f}{BP}\right) = \left(\frac{3}{9.425}\right)$$

$= 0.3183$ kg/kW h

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(iv) Heat balance sheet in kW

Heat input $= m_f \times CV_f = \left(\frac{3}{3600}\right) \times 44{,}200 = 36.83$ kW

Heat to BP $= 9.425$ kW

Heat to cooling water

$$Q_w = m_w \times C_{PW} \times \Delta T = \frac{4}{60} \times 4.186 \times 42$$

$= 11.721$ kW

Unaccounted heat loss

$= 36.83 - (9.425 + 11.721)$

$= 15.684$ kW

Heat balance sheet

Heat input	kW	Percentage	Heat output	kW	Percentage
Heat supplied	36.83	100	Heat to BP	9.425	25.59
–	–	–	Heat to cooling water	11.721	31.82
–	–	–	Unaccounted heat	15.684	42.59
Total	36.83	100	Total	36.83	100

13.13 QUESTIONS

13.13.1 Review Questions

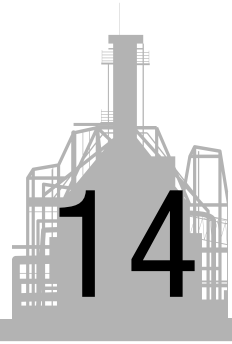
1. What are the merits and demerits of diesel power plants?
2. Draw a neat line diagram of a diesel power plant showing all the systems.
3. Briefly explain with a neat sketch.
   (a) Cooling system in diesel power plant.
4. Describe the auxiliary equipment of a diesel engine power plant with neat sketches.
5. State the applications of a diesel power plant.
6. Draw a neat diagram of a cooling system used for diesel power plant showing all the essential components.
7. Why the cooling and cleaning of lubricating oil is necessary? Draw a neat diagram of lubrication system used for medium capacity diesel power plant.
8. What are the various factors to be considered while selecting the site for diesel engine power plant?
9. Why the starting of diesel plant is more difficult? What different methods are used for starting diesel engine? Which method is common and why?
10. What are the outstanding features of diesel plants over thermal plants?

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11. Why diesel plants are not used for high capacity. What are its drawbacks when used for high capacity compared to steam plants?
12. Why super charging is necessary in diesel plants? What are the methods used for super charging the diesel engines? What are advantages of super charging as fuel consumption and overall efficiency of the plant are concerned?
13. Explain briefly various types of fuel injection systems.
    (a) Wet pump lubrication system
    (b) Dry pump lubrication system

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Power from Non-Conventional Sources



14

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14.1 RECENT DEVELOPMENTS IN METHODS OF POWER GENERATION

There are various energy sources are available to satisfy the demand for it in every country across the world. These sources are broadly classified into commercial or conventional energy sources and non-commercial or non-conventional energy sources. Most of the developed countries are highly dependent on conventional form of energy, whereas in developing countries such as India both the forms of energy sources are required.

14.1.1 Conventional Energy Sources

The following are commercial forms of energy:

(i) Fossil fuel, which may be in solid, liquid or gaseous form

(ii) Water power or energy stored in water

(iii) Nuclear energy

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Table 14.1 *Worldwide Consumption of Total Energy*

Source of energy	Contribution (%)	Overall contribution (%)
Coal	32.5	
Oil	38.3	92
Gas	19.0	
Uranium	0.13	
Hydro	2.00	
Wood	6.6	8
Dung	1.2	
Waste	0.3	

Worldwide consumption of total energy is as shown in Table 14.1.

From the table it is clear that 92 per cent of total energy comes from coal, oil, gas and uranium, and hence these are the most commonly used commercial energy sources.

1. *Coal*

It is the most common source of energy that is being used since industrialization. Modern steam boilers burn coal mainly as primary fuel in any of its available forms. Different ranks of coal available are peat, lignite, bituminous and anthracite.

According to estimates, coal reserves are sufficient enough to last for 200 years. However, it has lower calorific value and its transportation is uneconomical. When burnt it produces pollutants such as CO and CO_2 and hence is responsible for ecological imbalance.

2. *Oil*

Almost 40 per cent of energy needs is met by oil alone. With present consumption and a resource of 2,50,000 million tonnes of oil, it is estimated to last for only 100 years, unless more oil is discovered. A major chunk of oil comes from petroleum.

3. *Gas*

Due to non-availability of ready market for storage and transportation gas is not completely and effectively utilized and is burnt in huge quantities. Its transportation cost is much higher than oil. Large reserves are estimated to be located in inaccessible areas. Gaseous fuels are classified as:

(i) Gases of fixed composition such as acetylene, ethylene, methane, etc.

(ii) Industrial gases such as producer gas, coke oven gas, blast furnace gas, water gas, etc.

4. *Agricultural and organic wastes*

These include saw dust, bagasse, garbage, animal dung, paddy husk, corn stem, etc., accounting for a major energy consumption.

5. *Water*

It is one of the potential sources of energy meant exclusively for hydroelectric power generation. The potential energy of water is utilized to convert it into mechanical energy by using prime movers known as hydraulic turbines. The operating cost of the plant is cheaper compared to

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other types of power plants. It is the only renewable non-depleting source of energy, which does not contribute to pollution.

6. *Nuclear power*

Any matter consists of atoms held together by means of binding energy. Controlled fission of heavier unstable atoms such as U^{235}, Th^{232} and Pu^{239} liberates enormous amount of energy. This is possible only by utilizing small amount of nuclear fuels. It may be noted that the energy released by fission of 1 kg of U^{235} is equivalent to the heat generated by burning 4500 kg of coal. This factor makes the nuclear energy more attractive. The energy generated during nuclear fission reaction is used to produce steam in heat exchangers, which is utilized to run the turbo-generators.

For nuclear power generation three systems are considered. The first system is based on natural uranium yielding power and plutonium. The second system is using plutonium and depleted uranium in fast breeder reactors. The third system is using thorium and converting it into uranium in a fast breeder reactor. India has uranium reserves only enough to produce 6×10^6 kW of energy, which is a meagre 1 per cent of its current energy requirements.

14.1.2 Non-Conventional Energy Sources

1. *Solar energy*

Solar energy has the greatest potential of all the sources of renewable energy, which comes to the earth from sun. This energy keeps the temperature of the earth above that in colder space, causes wind currents in the ocean and the atmosphere, causes water cycle and generates photosynthesis in plants. The solar energy reaching the surface of the earth is about 10^{16} W, whereas the worldwide power demand is about 10^{13} W. That means solar energy gives us 1000 times more energy than our requirement. Even if we use 5 per cent of this energy, it is more than 50 times our requirement. The total solar radiation absorbed by the earth and its atmosphere is 3.8×10^{24} J/yr.

2. *Wind energy*

Wind energy can be economically used for the generation of electrical energy. Winds are caused from the following two main factors:

(i) Heating and cooling of the atmosphere, which generates convection currents. Heating is caused by the absorption of solar energy on the earth's surface and in the atmosphere.

(ii) The rotation of the earth with respect to atmosphere, and its motion around sun.

The energy available in the winds over the earth's surface is estimated to be 1.6×10^7 MW, which is almost the same as the present day energy consumption. Wind energy can be utilized to run wind mill, which in turn is used to drive generators. India has a potential of 20,000 MW of wind power.

3. *Energy from biomass and biogas*

Biomass means organic matter, which is produced in nature through photosynthesis. In the presence of solar radiation, water and carbon dioxide are converted into organic material CH_2O.

CH_2O is stable at low temperature but breaks at higher temperature releasing heat equal to 469 kJ/mol.

$$CH_2O + O_2 \rightarrow CO_2 + H_2O + 469 \text{ kJ/mol}$$

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It is possible to produce large amount of carbohydrate by growing plants such as algae in plastic tubes or ponds. The algae could be harvested, dried and burned for production of heat that could be converted into electricity by conventional methods. The biomass can be either used directly by burning or processed further to produce more convenient liquid or gaseous fuels.

Three different categories of biomass resources are as follows:

(i) Biomass in its traditional solid mass such as wood and agricultural wastes, which are burnt directly to get energy.

(ii) Biomass in its non-traditional form in which biomass is converted into ethanol and methanol, which are used as liquid fuels in engines.

(iii) Biomass in fermented form in which biomass is fermented anaerobically to obtain a gaseous fuel called biogas, which contains 55–65 per cent methane, 30–40 per cent CO_2 and rest as impurities such as H_2, H_2S and some N_2.

Various resources of biomass are

(i) Concentrated waste – municipal solids, sewage wood products, industrial waste, manure of large lots.

(ii) Dispersed waster residue – crop residue, legging residue, disposed manure.

(iii) Harvested biomass, standby biomass, biomass energy plantation.

Total solar radiation absorbed by plants is about 1.3×10^{21} J/yr and world's standing biomass has an energy content of about 1.5×10^{22} J.

4. *Energy from oceans*

A large amount of solar energy is collected and stored in oceans. The surface of water acts as a collector for solar heat, whereas the upper layer of the sea constitutes infinite heat storage reservoir. The heat contained in the oceans could be converted into electricity due to the temperature difference (20–25 K) between the warm surface water of the tropical oceans and the colder waters in the depths. This is the basic idea of OTEC systems. The surface water, which is at higher temperature, could be used to heat some low boiling point organic fluid, the vapours of which would run a heat engine. The amount of energy available from OTEC is enormous, and is replenished continuously.

5. *Tidal energy*

Due to universal gravitational effects of sun and moon on the earth, tides in the sea are generated. Due to fluidity of water mass there results a periodic rise and fall of water level during the rising and setting of sun and moon. This periodic rise and fall of water level results in tidal power. When the water is above the mean sea level, it is called flood tide (high tide), and when the water is below the mean sea level, it is called ebb tide (low tide).

To harness tides, a dam is built across the mouth of the bay with large gates and low head hydraulic reversible turbines. A tidal basin formed thus gets separated from the sea by dam. There always exists a difference between the water levels on either side of the dam during low tide and high tide. Thus, the reversible water turbine runs continuously producing power by using the generator connected to it. The energy dissipated with slowing down the rotation of the earth as a result of tidal action is about 10^{26} J/yr.

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6. *Geothermal energy*

This is the energy that comes from within the earth's crust. In some locations of the earth, the steam and hot water comes naturally to the surface. For large-scale use, bore holes are normally sunk with a depth up to 1000 m releasing steam and water at temperatures 200–300°C and pressure up to 3000 kN/m^2. Two methods are generally used to generate power using geothermal energy.

Method I

In this method, the heat energy is transferred to a working fluid, which operates the power cycle. It is found that molten interior mass of earth vents to the surface through fissures at temperatures ranging between 450 and 550°C.

Method II

In this method, the hot geothermal water and/or steam is used to operate the turbines directly. From the wellhead, steam is transmitted by using pipes of 1 m diameter over distances up to 3000 m to the power plant. In this system, water separators are used to separate moisture and solid particles from steam.

The heat flux from the earth's interior through the surface is 9.5×10^{20} J/yr. The total amount of heat stored in water or steam to a depth of 10 km is estimated to be 4×10^{21} J and that stored in the first 10 km of dry rock is around 10^{27} J.

7. *Hydrogen energy*

Hydrogen as an energy is another alternative for conventional fuels. It can be easily produced from water, which is available abundantly in nature. It has the highest energy content per unit of mass any chemical fuel and is a better substitute for hydrocarbons, with increased combustion efficiency. It is non-polluting and can be used in fuel cells to produce both electricity and useful heat. However, it has technical problems of production, storage and transportation.

8. *Fuel cells*

These are electrochemical devices used for the continuous conversion of the portion of the free energy change in a chemical reaction to electrical energy. It operates with continuous replenishment of the fuel and the oxidant at active electrode area and does not require recharging (Figure 14.1).

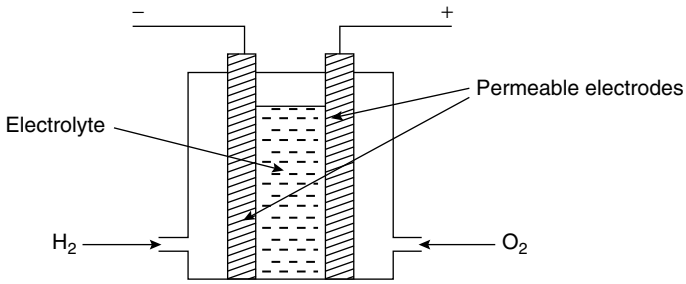


Fig. 14.1 *Fuel Cell*

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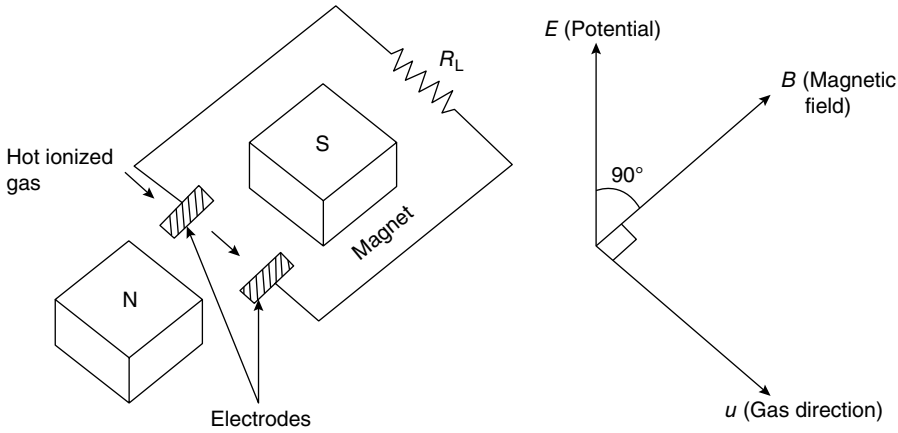


Fig. 14.2 *Principle of MHD Power Generation*

The main components of a cell are (i) a fuel electrode, (ii) an oxidant or air electrode and (iii) an electrolyte. Some of the fuel cells used are hydrogen, oxygen (H_2, O_2), hydrazine (N_2H_4, O_2), carbon/coal (C, O_2), methane (CH_4, O_2), etc.

9. *Magneto hydrodynamic generators*

Magneto hydrodynamic (MHD) generators are used for direct conversion of thermal energy into electrical energy. They work on Faraday principle. When an electric conductor moves across a magnetic field, a voltage is induced in it which produces an electric current. In MHD generators, the solid conductors are replaced by a fluid, which is electrically conducting. The working fluid may be either an ionized gas or liquid metal. The hot, partially ionized and compressed gas is expanded in a duct, and forced through a strong magnetic field; electrical potential is generated in the gas. Electrodes placed on the side of the duct pick up potential generated in the gas. The direct current (DC) thus obtained can be converted into alternating current (AC) using an inverter (Figure 14.2).

The system is simple with large power- and temperature-handling capacity without any moving parts. It is highly reliable and can be brought to full load within 45 seconds. Power output can be changed from no load to full load in a fraction of a second.

10. *Thermionic converter*

It consists of two electrodes held in a container filled with ionized caesium vapour. Heating one electrode 'boils out' electrons that travel to the opposite colder side electrode. The positive ions in the gas neutralize the space–charge effect of the electrons that normally prevent the flow of electrons. Ionized gas offsets space–charge effect that tends to repel migration of electrons.

Electrons, which are emitted by heating cathode, are migrated to cooler anode collector and flow through outer circuit to develop electric power. Low-work function materials such as barium and strontium oxides are used for anodes, whereas high-work function materials such as tungsten impregnated with barium is used for cathodes (Figure 14.3).

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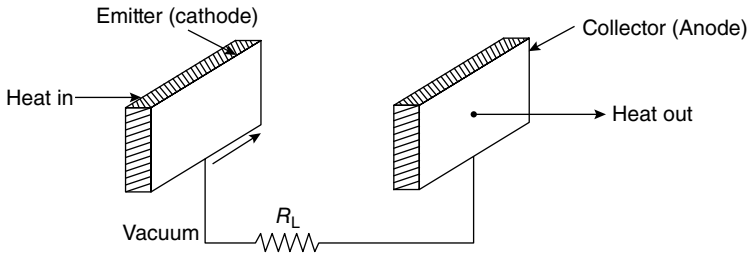


Fig. 14.3 *Thermionic Converter*

11. *Thermo electric power*

It is a device that converts heat directly into electric power. It eliminates the conversion of heat into kinetic energy of gas or steam flow. Its principle is based on Seebeck effect: If two dissimilar materials are joined to form a loop and the two junctions are maintained at different temperatures, an emf will be developed around the loop. The magnitude of the emf (E) developed by the above process is proportional to the temperature difference between the two junctions.

$$E\alpha(T_2 - T_1) = \alpha(T_h - T_c)3$$

where T_h = Temperature of hot junction, T_c = Temperature of cold junction and α = Seebeck coefficient.

The hot junction is maintained at a temperature T_h by the applied heat source, which may be small oil or gas burner, a nuclear reactor, or direct solar radiation by paraboloidal concentrator and the cold junction is maintained at T_c by either water cooling or radiative heat transfer (Figure 14.4).

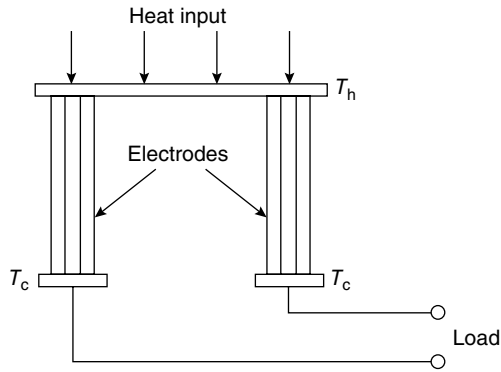


Fig. 14.4 *Thermoelectric Generator*

14.2 UTILIZATION OF SOLAR ENERGY

14.2.1 Solar Radiation

It is the energy produced and radiated by the sun that reaches the surface of radiation can be converted directly or indirectly into other forms of energy such as heat and electricity.

Solar energy is radiated from the sun in the form of electromagnetic waves of shorter wavelengths of 0.2–4.0 μm (1 μm = 10^{-6} m). Out of all the solar energy radiations reaching the earth's atmosphere, 8 per cent is by ultraviolet radiation (wavelength < 0.39 μm), 40 per cent is by visible light (wavelength, 0.39–0.78 μm) and 46 per cent is by infrared radiation (long wave of length > 0.78 μm).

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Major drawbacks

(i) It is intermittent and variable in nature.

(ii) It requires large area to collect the energy at a useful rate.

14.2.2 Solar Radiation at Earth's Surface

The solar radiation that penetrates the earth's atmosphere and reaches the surface differs in both amount and character from radiation at the top of the atmosphere. The radiation entering the atmosphere is partly absorbed by molecules, and a part of the radiation is reflected back into the space by clouds. Part of the solar radiation is scattered by droplets in clouds by atmospheric molecules and dust particles. Oxygen and ozone absorb nearly all the ultraviolet radiation, whereas CO_2 and H_2O vapours absorb some energy from infrared range (Figure 14.5).

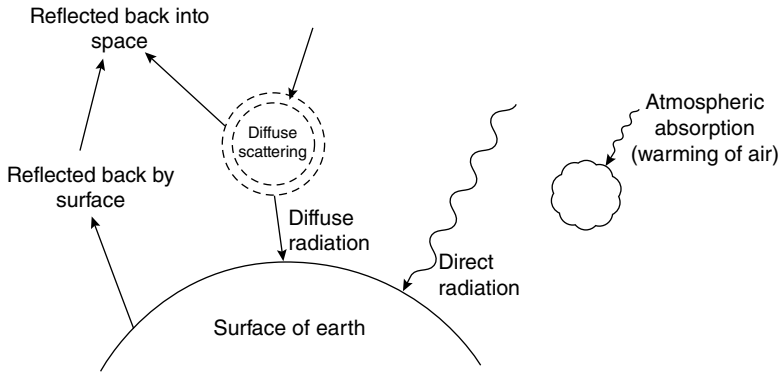


Fig. 14.5 *Direct, Diffuse and Total Radiation*

1. *Direct radiation or beam radiation*

Solar radiation that has not been absorbed or scattered and reaches the ground directly from the sun is called 'direct radiation' or 'beam radiation'. It is the radiation, which produces a shadow when interrupted by an opaque object.

2. *Diffuse radiation*

It is the solar radiation received from the sun after its direction has been changed by reflection and scattering by the atmosphere. Since the solar radiation is scattered in all directions in the atmosphere, diffuse radiation comes to the earth from all parts of the sky.

3. *Insolation*

It is the total solar radiation received at any point on the earth's surface and is the sum of the direct and diffuse radiation. More specifically insolation is defined as the total solar radiation energy received on a horizontal surface of unit area on the ground in unit time.

The insolation at a given point or location on the earth's surface depends, among other factors, on the altitude of the sun in the sky. As a result of absorption and scattering, the insolation is less when the sun is low in the sky than when it is higher. On a clear, cloudless day, about 10–20

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percent of the insolation is from diffuse radiation; the proportion increases up to 100 per cent when the sun is completely obscured by clouds.

14.2.3 Basic Definitions of Solar Angles

Some basic definitions of solar angles and related terms are discussed hereunder.

1. *Sun at zenith*

It is the position of the sun directly over head.

2. *Air mass*

It is the path length of radiation through the atmosphere to the length of path when the sun is at the zenith.

Air mass = cos (altitude angle) except for very low solar altitude angles.

$m = 1$; when the sun is at the zenith (e.g. sea level)

$m = 2$; when zenith angle is 60° ($\theta_z = 60°$)

$m = \sec(\theta_z)$ for $m > 3$

$m = 0$; just above earth's atmosphere

3. *Solar angles*

Let θ = Angle between an incident beam radiation I and the normal to the plane surface.
Then, radiation intensity normal to the surface is

$$I = I\cos\theta$$

Where θ = Incident angle

(i) *Latitude, ϕ_l*

It is the angle made by the radial line joining the location to the centre of earth with the projection of the line on the equatorial plane, denoted by ϕ_l. It is also given by the angular distance north or south of the equator measured from the centre of the earth (Figure 14.6).

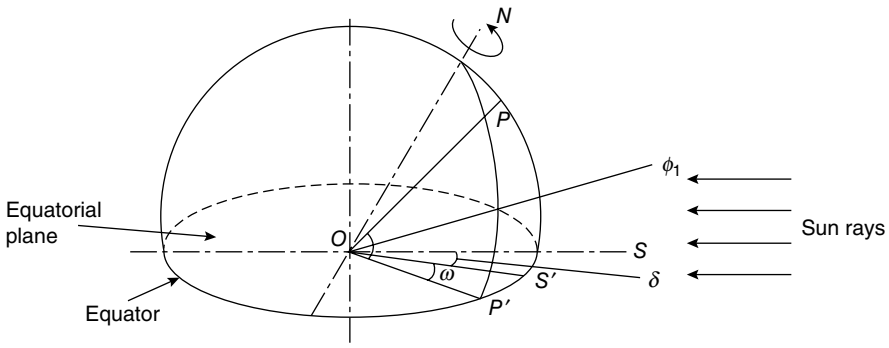


Fig. 14.6 *Latitude (ϕ_l), Hour Angle (ω) and Sun's Declination (δ)*

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If P is the location on the earth's surface and O is the centre of the earth, then ϕ_l is given by the angle between the line OP and projection of OP on the equatorial plane. As a method of convention, the latitude will be measured as +ve for the northern hemisphere.

(ii) *Declination (δ)*

It is the angular distance of sun's rays north or south of the equator. It is the angle between the line extending from the centre of the sun to the centre of the earth and the projection of this line upon the earth's equatorial plane. Declination varies between 23.5° on June 22 to 23.5° on December 22 (Figure 14.7).

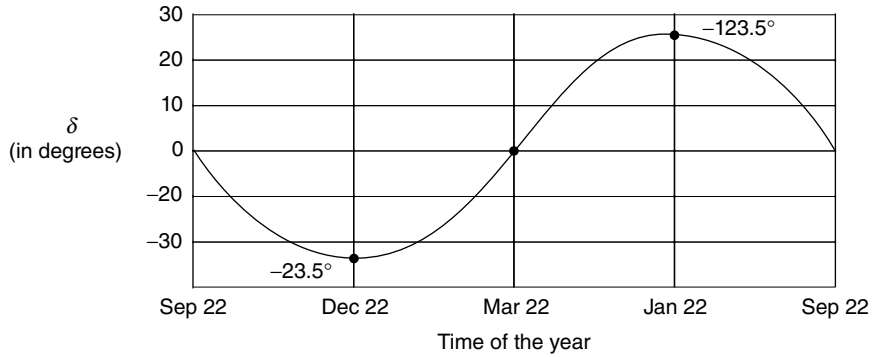


Fig. 14.7 *Variation of Sun's Declination*

The declination in degrees for any given day may be given by Cooper's equation:

$$\delta(\text{in degrees}) = 23.45 \sin\left[\frac{360}{365}(284+n)\right], \text{ where } n \text{ is the day of the year}$$

e.g. March 22 is the 31 + 29 + 22 = 82nd day

$$\therefore \qquad n = 82$$

(iii) *Hour angle (ω)*

It is the angle through which the earth must turn to bring the meridian of a point directly in line with the sun's rays. The hour angle is equivalent to 15° per hour.

It is measured from noon based on the local solar time (LST) or local apparent time, being positive in the morning and negative in the afternoon. It is the angle measured in the earth's equatorial plane, between the projection of OP and the projection of O line from the centre of the sun to the centre of the earth.

(iv) *Altitude angle (α) or solar altitude*

It is the vertical angle between the projection of the sun's rays on the horizontal plane and the direction of sun's rays passing through the point (Figure 14.8).

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(v) *Zenith angle (θ_Z)*

It is the angle between the sun's rays and a line perpendicular to the horizontal plane through the point *P*. That is, the angle between the beam from the sun and the vertical zenith angle is complimentary angle of sun's altitude angle.

$$\theta_z = \frac{\pi}{2} - \alpha$$

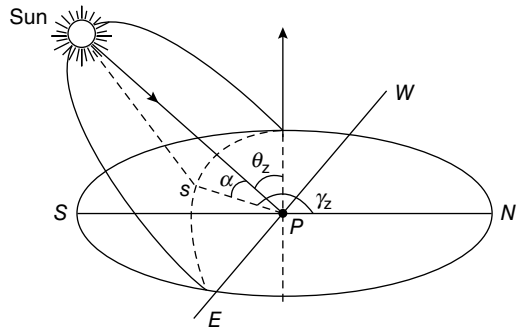


Fig. 14.8 *Sun's Zenith, Altitude and Azimuth Angles*

(vi) *Solar azimuth angle (γ_s)*

It is the solar angle in degrees along the horizon east or west or north or it is the horizontal angle measured from the north to the horizontal projection of sun's rays. It is considered +ve when measured west wise.

In terms of basic angles, ϕ_l, δ and ω, solar altitude, zenith angle and azimuth angle can be expressed as follows:

$$\cos\theta_z = \cos\phi.\cos\omega.\cos\delta + \sin\phi.\sin\delta$$
$$\cos\gamma_s = \sec\alpha\,(\cos\phi.\sin\delta - \cos\delta.\sin\phi.\cos\omega)$$
$$\sin\gamma_s = \sec\alpha.\cos\delta.\sin\omega$$

Note: If north latitudes are considered positive and south latitudes negative, the declination will be positive for summer period between the vernal equinox and autumnal equinox (March 22 to September 22 approximately) and negative at other times.

(vii) *The slope (s)*

It is the angle made by the plane surfaces with the horizontal. It is considered positive for surfaces slopping towards the south and negative for surface slopping towards the north.

(viii) *Surface azimuth angle (γ)*

It is the angle of deviation of the normal to the surface from the local meridian, the zero point being south, east positive and west negative (Figure 14.9).

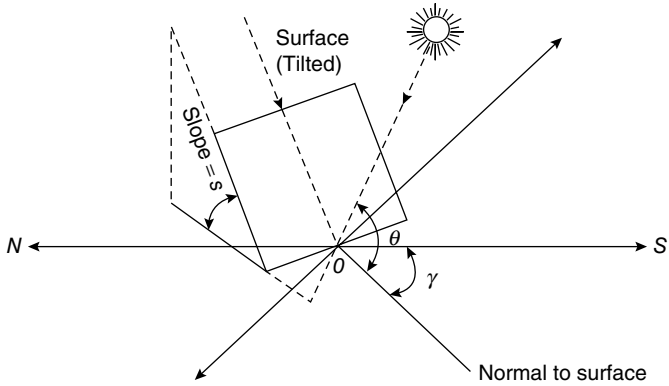


Fig. 14.9 *Surface Azimuth Angle and Slope*

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(ix) *Incident angle (θ)*

It is the angle being measured from a plane and is equal to the angle between the beam of rays and normal to the plane. It is expressed as

$$\begin{aligned}\cos\theta = {} & \sin\phi_l(\sin\delta.\cos s + \cos\delta.\cos\gamma.\cos\omega.\sin s) \\ & + \cos\phi_l(\cos\delta.\cos\omega.\cos s - \sin\delta.\cos\gamma.\sin s) \\ & + \cos\delta.\sin\gamma.\sin\omega.\sin s) \end{aligned} \quad (1)$$

where ϕ_l = Latitude (North positive) and δ = Declination (North positive).

ω = Hour angle (positive between solar midnight and noon, otherwise negative)

Hour angle is mathematically expressed as

$$\omega = 15\ (12 - \text{LST}) \quad (2)$$

Note: At solar noon, $\omega = 0$ and each hour angle is 15° with morning positive and afternoon negative.

e.g. At 11.00 am, $\omega = +15°$

At 2.30 pm, $\omega = -37.5°$

For vertical surfaces $s = 90°$ in Equation 1 above

$$\begin{aligned}\therefore \quad \cos\theta = {} & \sin\phi.\cos\delta.\cos\gamma.\cos\omega - \cos\phi.\sin\delta.\cos\gamma + \cos\delta \\ & \sin\gamma.\sin\omega \end{aligned} \quad (3)$$

For horizontal surfaces $s = 0°$; $\theta = \theta_z$ in Equation 1 above

$$\begin{aligned}\therefore \quad \cos\theta_z &= \sin\delta.\cos\phi + \cos\delta.\cos\phi.\cos\omega \\ &= \sin\alpha \end{aligned}$$

$$\text{i.e.,} \quad \cos\theta = \cos\theta_z = \sin\alpha \quad (4)$$

For surface facing due south, $\gamma = 0$; $\theta = \theta_t$ (tilted)

$$\begin{aligned}\cos\theta_t &= \sin\phi(\sin\delta.\cos s + \cos\delta.\cos\omega.\sin s) \\ &= \cos\phi(\cos\delta.\cos\omega.\cos s - \sin\delta.\sin s) \\ &= \sin\delta.\sin(\phi - s) + \cos\delta.\cos\omega.\cos(\phi - s) \end{aligned} \quad (5)$$

For vertical surfaces facing due south, $s = 90$; $\gamma = 0$

$$\therefore \quad \cos\theta_z = (\sin\phi.\cos\delta.\cos\phi.\sin\delta) \quad (6)$$

4. *Day length*

At the time of sunrise or sunset, $\theta_z = 90°$ substituting in Equation (4), sunrise hour angle ω_s is given by

$$\cos\omega_s = \frac{\sin\phi.\sin\delta}{\cos\phi.\cos\delta} = -\tan\phi.\tan\delta$$

$$\omega_s = \cos^{-1}(-\tan\phi.\tan\delta)$$

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As 15° of the hour angle = 1 hour, day length

$$t_{d1} = \frac{2\omega_s}{15} = \frac{2}{15}\cos^{-1}\left(-\tan\phi.\tan\delta\right) \tag{7}$$

Note: For hour angle at the time of sunrise or sunset on an inclined surface $\theta_z = 90°$, from Equation (5)

$$\omega_{st} = \cos^{-1}[-\tan(\phi - s)\tan\delta]$$

Hence, day length

$$t_d = \frac{2}{15}\cos^{-1}\left[\left(-\tan\left(\phi - s\right).\tan\delta\right)\right] \tag{8}$$

5. *Local solar time*

It is also known as local apparent time, which is the time used for calculating the hour angle. The local solar time is obtained from the standard time observed on a clock by making two corrections.

The first correction takes into account the difference in longitude between a location and a meridian on which the standard time is based. For every degree difference in longitude, this difference is four minutes.

The second correction takes into account time correction arising due to small perturbations in earth's orbit and rate of rotation.

LST = Standard time 4 (Standard time longitude – Longitude of location) + Equation of time correction

Note: The –ve sign is applied for eastern hemisphere.

14.2.4 Solar Constant and Intensity of Solar Radiation

1. *Solar constant*

It is defined as the rate of solar energy arriving at the top of the atmosphere, denoted by I_{SC}. It is the amount of energy received in unit time on a unit area perpendicular to the sun's direction at the mean distance of the earth from the sun. Since sun's activity and distance vary throughout the year, solar energy also varies.

Three common units used for expressing solar constants are

$I_{SC} = 1.353$ kW/m^2
$I_{SC} = 116.5$ Langley (1 Langley = 1 cal/cm^2)
$I_{SC} = 429.2$ Blu/ft^2/hr

2. *Intensity of solar radiation*

Since the distance between sun and earth varies the extraterrestrial flux also varies. Earth is closest to the sun in the summer and farthest away in the winter. This variation in distance produces a nearly sinusoidal variation in the intensity of solar radiation that reaches the earth.

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$$I = I_{SC}\left[1 + 0.033\cos\frac{360n}{365}\right]$$

where n = Day of the year

14.2.5 Average Solar Radiation or Monthly Average Daily Global Radiation

1. *Angstrom relation*

If = Monthly average of the daily global radiation on a horizontal surface at a location (kJ/m^2. day)
= Monthly average of the daily global radiation on a horizontal surface at a location on a clear sky, (kJ/m^2. day)
= Monthly average of the maximum possible sunshine hours/day at the location, that is, day length on a horizontal surface (h)
= Monthly average sunshine hours per day at the location (h)
a = Constant = 0.35
b = Constant = 0.61
then

$$\frac{\bar{H}_g}{\bar{H}_c} = a + b\left(\frac{\bar{s}}{\bar{s}_{max}}\right) \tag{1}$$

Since it is difficult to decide about a clear sky, the above equation is not used extensively.

2. *Page equation*

If $\bar{H}_o$ = Monthly average of the daily extraterrestrial radiation, which would fall on a horizontal surface at the location considered.

then

$$\frac{\bar{H}_g}{\bar{H}_o} = a + b\left(\frac{\bar{s}}{\bar{s}_{max}}\right)$$

where a and b are constants depending on locations and

$$H_o = \left[\frac{24}{\pi} I_{SC}\left(1 + 0.033\cos\left(\frac{360n}{365}\right)\right)\right](\omega_{st}.\sin\phi.\sin\delta + \cos\phi.\cos\delta.\sin\omega_{st})$$

Example 14.1

Calculate the angle made by beam radiation with the normal to a flat plate collector on December 21 at 0900 h (LAT). The collector is located in New Delhi (28°35′*N*, 77°12′*E*) and is tilted at an angle of 36° with the horizontal and is pointing due south.

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Solution: In this case, $\gamma = 0°$. On December 21, $n = 355$.

Angle of declination

$$d = 23.45 \sin\left[\frac{360}{365}(284 + n)\right]$$

$$= 23.45 \sin\left[\frac{360}{365}(284 + 355)\right]$$

$$= -23.45°$$

At 0900 h (LAT), hour angle $\omega = 45°$

For surface facing due south, $\gamma = 0$,

$$\cos\theta_t = \sin\delta.\sin(\phi - s) + \cos\delta.\cos\omega.\cos(\phi - s)$$

$$= \sin(-23.45°).\sin(28.58° - 36°) + \cos(-23.45°)$$

$$\cos 45°.\cos(28.58° - 36°)$$

$$= 0.0514 + 0.6433 = 0.6947$$

$$\therefore \quad \theta_t = 46°$$

Example 14.2

Calculate the angle made by beam radiation with the normal to a flat plate collector on January 26 at 900 hours solar time at a location 28°35′N, 77°17′E. The collector is tilted at an angle plus 10° to the horizontal and is pointing due south.

Solution: Angle of declination

$$\delta = 23.45 \sin\left[\frac{360}{365}(284 + n)\right]$$

$$= 23.45 \sin\left[\frac{360}{365}(284 + 25)\right]$$

$$= -19.26°$$

At 0900 h (LAT), $\omega = 45°$

For $\gamma = 0$;

$$\cos\theta_t = \sin\delta.\sin(\phi - s) + \cos\delta.\cos\omega.\cos(\phi - s)$$

$$= \sin(-19.26°).\sin(28.58° - 10°) + \cos(-19.26°)$$

$$\cos 45°.\cos(28.58° - 10°)$$

$$= -0.1051 + 0.6327$$

$$= 0.5276$$

$$\therefore \quad \theta_t = 58.16°$$

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Example 14.3

Calculate the sun's altitude angle and azimuth angle at 40° North latitude at 3.30 pm IST on November 1 and 21.

Solution: Angle of declination, δ

$$= 23.45 \sin\left[\frac{360}{365}(284+n)\right]$$

$$= -15.06°$$

Hour angle, $\omega = -52.5°$

Note:

At noon $\omega = 0$

In the morning +ve and afternoon −ve, or positive between midnight and noon.

Latitude angle = $\phi_l = 40°$

Now zenith angle $\cos\theta_Z = \cos\phi.\cos\omega.\cos\delta + \sin\phi.\sin\delta$

$= \cos 40°.\cos(-52.5°).\cos(-15.06°) + \sin(40°).\sin(-15.06°)$

$= 0.4503 - 0.1670 = 0.2833$

$\therefore \quad \theta_Z = 73.54°$

But $\theta_Z = \frac{\pi}{2} - \alpha$

$73.54 = 90 - \alpha$

$\alpha = 16.46°$ = altitude angle

Solar azimuth angle is given by

$$\cos\gamma_s = \sec\alpha(\cos\phi.\sin\delta - \cos\delta.\sin\phi.\cos\omega)$$

Example 14.4

Estimate the monthly average daily global radiation on a horizontal surface at Baroda (22°00′N, 73°10′E) during the month of March if average sunshine hour per day is 9.5. Take $a = 0.28$ and $b = 0.48$.

Solution: On March 16, $n = 75$

$\therefore$

$$\delta = 23.45 \sin\left[\frac{360}{365}(284+n)\right]$$

$$= 23.45 \sin\left[\frac{360}{365}(284+75)\right]$$

$$= -2.42°$$

Sunrise hour angle on an inclined surface

$$\omega_{st} = \cos^{-1}\{-\tan(\phi - s).\tan\delta\}$$

$$= \cos^{-1}\{-\tan 22°.\tan(-2.42°)\}$$

$$= 89.02° = 1.554 \text{ rad}$$

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$$\text{Day length, } S_{max} = \frac{2}{15}\omega_{st} = \frac{2}{15} \times 1.554 \text{ rad}$$

$$= \frac{2}{15} \times 89.02^\circ = 11.87 \text{ h}$$

If = Monthly average of the daily extraterrestrial radiation, which would fall on a horizontal surface at the location considered by Klein's approximation.
$= H_o$

$$\text{But } H_o = \left[\frac{24}{\pi} I_{SC}\left(1 + 0.033\cos\left(\frac{360n}{365}\right)\right)\right]\left(\omega_{st}.\sin\phi.\sin\delta + \cos\phi.\cos\delta.\sin\omega_{st}\right)$$

$$= \left[\frac{24}{\pi} I_{SC}\left(1 + 0.033\cos\left(\frac{360 \times 75}{365}\right)\right)\right]$$

$$\left(1.554\sin 22^\circ.\sin(-2.42^\circ) + \cos 22^\circ.\cos(-2.42^\circ).\sin 89.02^\circ\right)$$

$$= 34{,}206 \text{ kJ/m}^2 \text{ day}$$

$$\text{Now, } \bar{H}_g = \frac{\bar{H}_g}{\bar{H}_o} = a + b\left(\frac{\bar{s}}{\bar{s}_{max}}\right)$$

$$\frac{\bar{H}_g}{34{,}206} = 0.28 + 0.48\left(\frac{9.5}{11.87}\right)$$

$$= 22{,}718 \text{ kJ/m}^2 \text{ day}$$

Example 14.5

Determine the local solar time and declination at a location 23°15′N, 76°30′E at 12.30 IST on June 29. Equation of time correction is given from standard table or chart = $-1^1.01^{11}$. Take standard time longitude as 81°30′E.

Solution: The local solar time,

$$\text{LST} = \text{IST} - (\text{standard time longitude} - \text{longitude of location}) + \text{equation of time correction}$$

$$= 12^h.30' - 4\,(81^\circ.30'\text{E} - 76^\circ 30'\text{E}) + (-1^1.0^{11})$$

$$= 12^h.30' - 4 \times 5^1 - 1^1.0^{11}$$

$$= 12^h.8^1\,59^{11}$$

$$\text{Declination, } \delta = 23.45\sin\left[\frac{360}{365}(284 + n)\right]$$

$$= 23.45\sin\left[\frac{360}{365}(284 + 180)\right]$$

$$= 23.24^\circ$$

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14.2.6 Solar Radiation on Tilted Surfaces

The rate of receipt of solar energy on a given surface on the ground depends on the orientation of the surface with reference to the sun. A fully tracking surface of sun's energy is the one that directly faces the sun than the surface of same area with different orientation. Most of the solar collectors and devices are tilted at an angle to the horizontal. Hence, it is necessary to convert data for an hourly radiation on a horizontal surface of radiation on a tilted surface.

1. *Beam radiation*

In most cases, the tilted surface faces due south.

$$\therefore \quad \gamma = 0$$

Hence, $$\cos\theta_t = \sin\delta.\sin(\phi - s) + \cos\delta.\cos\omega.\cos(\phi - s)$$

For horizontal surface, $$\theta = \theta_Z$$

$$\cos\theta_Z = \sin\phi.\sin\delta + \cos\phi.\cos\delta.\cos\omega$$

The ratio of beam radiation falling on the tilted surface to that falling on horizontal surface is given by

$$R_b = \frac{H_t}{H} = \frac{H_n \cos\theta_t}{H_n \cos\theta_Z}$$

$$\text{i.e.} \quad R_b = \left[\frac{\sin(\phi - s).\sin\delta + \cos(\phi - s).\cos\delta.\cos\omega}{\sin\phi.\sin\delta + \cos\phi.\cos\delta.\cos\omega}\right] \tag{1}$$

The above ratio R_b is known as tilt factor for beam radiation (Figure 14.10).

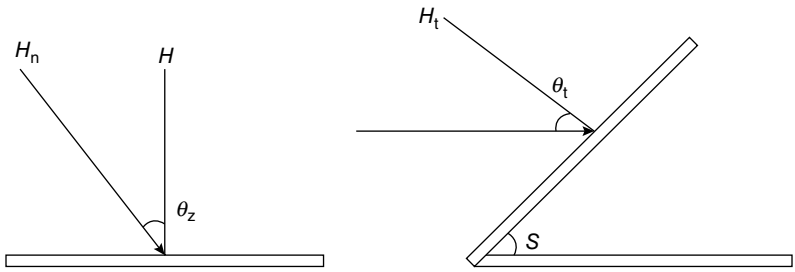


Fig. 14.10 *Radiation on Horizontal and Tilted Surfaces*

2. *Total radiation*

A flat plate collector absorbs both beam and diffuses type radiation. The angular correction factor for beam radiation is given by the above Equation (1).

Since it is assumed that diffuse radiation originates near the sun and scatters in the forward direction, the correction factor for diffuse radiation is assumed as equal to R_b. Hence, correction factor for both beam and diffuse radiation is given by

$$R = \frac{H_t}{H} = \left[\frac{H_t}{H_b + H_d}\right] \rightarrow \text{for clear sky} \tag{2}$$

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For cloudy or hazy days, H_d is assumed to be uniformly distributed over the sky. Then

$$R = \frac{H_t}{H} = \frac{H_b}{H} R_b + \frac{H_d}{H} \tag{3}$$

3. *Diffuse radiation*

The tilt factor for diffused radiation is the ratio of the diffuse radiation flux falling on the tilted surface to that falling on the horizontal surface. It depends on the distribution of diffuse radiation over the sky and on the portion of the sky dome seen by the tilted surface.

Assuming sky is an isotropic source of radiation (diffuse), for a tilted surface of slope s

$$R_d = \frac{1+\cos s}{2} = \text{Radiation shape factor for a tilted surface w.r.t. sky.} \tag{4}$$

4. *Reflected radiation*

Radiation shape factor for a tilted surface w.r.t. the ground $= \dfrac{1-\cos s}{2}$

If the reflection of beam and diffuse radiation falling on the ground is diffuse and isotropic, we have tilt factor for reflected radiation

$$R_r = \rho\left[\frac{1-\cos s}{2}\right] \tag{5}$$

where ρ = Reflexivity

5. *Flux on tilted surface*

The flux falling on a tilted surface is given by

$$H_t = H_b R_b + H_d R_d + (H_b + H_b) R_r \tag{6}$$

and

$$R = R_b + R_d + R_r \tag{7}$$

Dividing Equation (6) by H_g throughout

$$\frac{H_t}{H_g} = \left[\left(\frac{H_b}{H_g} R_b + \frac{H_d}{H_g} R_t + \frac{H_b + H_d}{H_g} R_r\right)\right]$$

$$= \left[\left(1 - \frac{H_d}{H_g}\right) R_b + \frac{H_d}{H_g} R_d + R_r\right] \tag{8}$$

The values of ρ suggested by Liu and Jordon are

$\rho = 0.2 \rightarrow$ for no snow (Indian condition)

$= 0.7 \rightarrow$ for snow cover

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Example 14.6

Calculate the monthly average hourly radiation falling on a flat-plate collector facing south ($\gamma = 0$) with a slope of 15° given the following data.

Location - Madras (13°00′N)
Month - October
Time - 1100–1200 h (LAT)
- 2408 kJ/m^2-h
- 1073 kJ/m^2-h

Assume ground reflectivity to be 0.2.

Solution: The representative day in October (max radiation) falls roughly in the middle of the month, that is, October 15

On October 15, $n = 288$

$$\therefore \quad \text{Declination } \delta = 23.45 \sin\left[\frac{360}{365}(284+n)\right]$$

$$= 23.45 \sin\left[\frac{360}{365}(284+288)\right]$$

$$= -9.6°$$

Tilt factor for beam radiation

$$R_b = \left[\frac{\sin(\phi - s).\sin\delta + \cos(\phi - s).\cos\delta.\cos\omega}{\sin\phi.\sin\delta + \cos\phi.\cos\delta.\cos\omega}\right]$$

$$= \left[\frac{\sin(13-15).\sin(-9.6) + \cos(13-15).\cos(-9.6).\cos 7.5}{\sin 13.\sin(-9.6) + \cos 13.\cos(-9.6).\cos 7.5}\right]$$

$$R_b = 1.12$$

Tilt factor for diffuse radiation

$$R_d = \left[\frac{1+\cos s}{2}\right] = \left[\frac{1+\cos 15°}{2}\right] = 0.983$$

Tilt factor for reflected radiation

$$R_r = \left[\frac{1-\cos s}{2}\right] = 0.2\left[\frac{1-\cos 15°}{2}\right] = 0.0034$$

Assuming $\bar{H}_g = H_g$ and $\bar{H}_d = H_d$

$$\frac{H_t}{H_g} = \left[\left(1 - \frac{H_d}{H_g}\right)R_b + \frac{H_d}{H_g}R_d + R_r\right]$$

$$H_t = 2408\left[\left(1 - \frac{1073}{2408}\right)1.12 + \frac{1073}{2408} \times 0.983 + 0.0034\right]$$

$$H_t = 2558 \text{ kJ/m}^2\text{-h}$$

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14.2.7 Solar Energy Collectors

1. *Flat plate collectors*

Based on the type of heat transfer fluid (HTF), flat plate collectors are divided into the following two types:

(i) *Liquid heating collectors*
These are used for heating water and non-freezing aqueous solutions and occasionally for non-aqueous HTFs.

(ii) *Air or gas heating collectors*
They are employed as solar air heaters.

Main components

Main components of flat plate collectors are

(a) A transparent cover
It includes one or more sheets of glass or radiation transmitting plastic film or sheet.

(b) Tubes, fins, passages or channels
They are integral with the collector absorber plate or connected to it for carrying water, air or other fluid.

(c) The absorber plate
It is usually metallic or black-coated surface used for absorbing radiation energy.

(d) Insulation
It is provided at the back and sides to minimize heat losses. Fibre glass or styrofoam is used for this purpose.

(e) Container or casing
It is used to enclose the other components to protect them from weather.

(f) Liquid plate collector
A typical liquid collector that features all designs of a flat plate collector is shown in Figure 14.11.

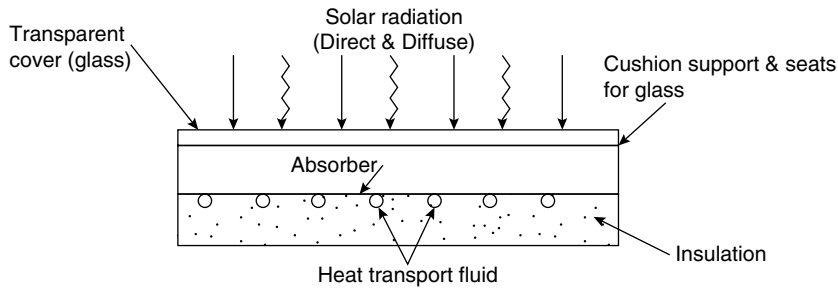


Fig. 14.11 *Plate- and Tube-Type Collector*

It consists of a flat surface known as absorbing surface, which has very high absorptivity for solar radiation. It is made up of copper, steel or aluminium material with copper tubes in thermal contact with the plates. The absorber plate is usually made from a metal sheet of 1–2 mm thickness, whereas the tubes are 10–15 mm in diameter. The tubes are soldered brazed or clamped

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to the bottom or top of the absorber plate with the pitch ranging from 50 to 150 mm. In some designs, the tubes are also in line and integral with the absorber plate. Usually corrugated galvanized sheets shown in Figure 14.12 are used as absorber plates throughout the world.

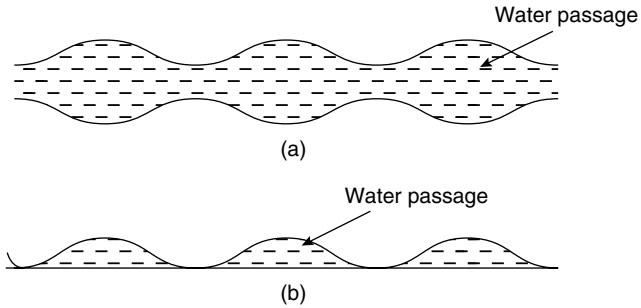


Fig. 14.12 *Corrugated Galvanized Sheets*

Working

Heat is transferred from the absorber plate to a point of use by the circulating fluid, namely water across the solar-heated surface. The heat loss from the rear surface is prevented by placing a thermal insulation of 5–10 cm thickness behind the absorber plate. The front cover is made up of one or more layers of glass, which is transparent to incoming solar radiation and opaque to infrared re-radiation from the absorber. The glass covers act as convection shields to reduce the losses from the absorber plate beneath. It is usual practice to have 1–2 glass covers of 3–4 mm thickness.

Using a second cover of glass has the following advantages:

(i) Losses due to air convection are further reduced.

(ii) Radiation losses in the infrared spectrum are reduced by a further 25 per cent. This is because half of the 50 per cent, which is emitted by the first glass cover is further back radiated.

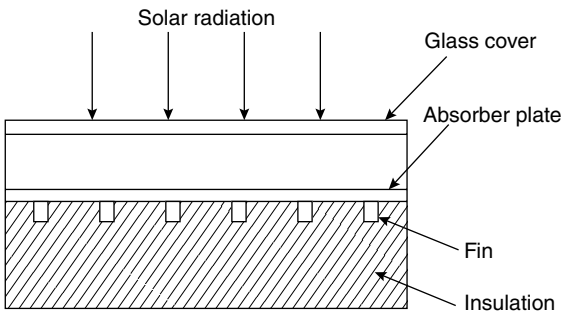


Fig. 14.13 *A Typical Solar Air Collector*

However, more than two glass covers increase the percentage of reflection of incoming light. A typical collector has dimensions 2 m × 1 m × 15 cm.

Air collectors or solar air heater

Figure 14.13 shows a flat plate collector where an air stream is heated by the backside of the collector plate. Fins attached to the surface increase the contact surface. The backside of the collector is heavily insulated with mineral wool or some other material.

The use of air eliminates the freezing and corrosion problems, and small air leaks are of less concern than water leaks. The heated air can be used directly for space heating. However, the system requires large ducts and higher flow rates, which increases pumping power. Basically, two types of air heaters are available.

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1. *Non-porous absorber plate-type collectors*

Three different types of non-porous absorber plate-type collectors are shown in Figure 14.14. In Figure 14.14(a) type, the absorber is cooled by the air stream flowing over both sides of the plate. In Figure 14.14(b) type, air flows over the upper surface and hence increases the

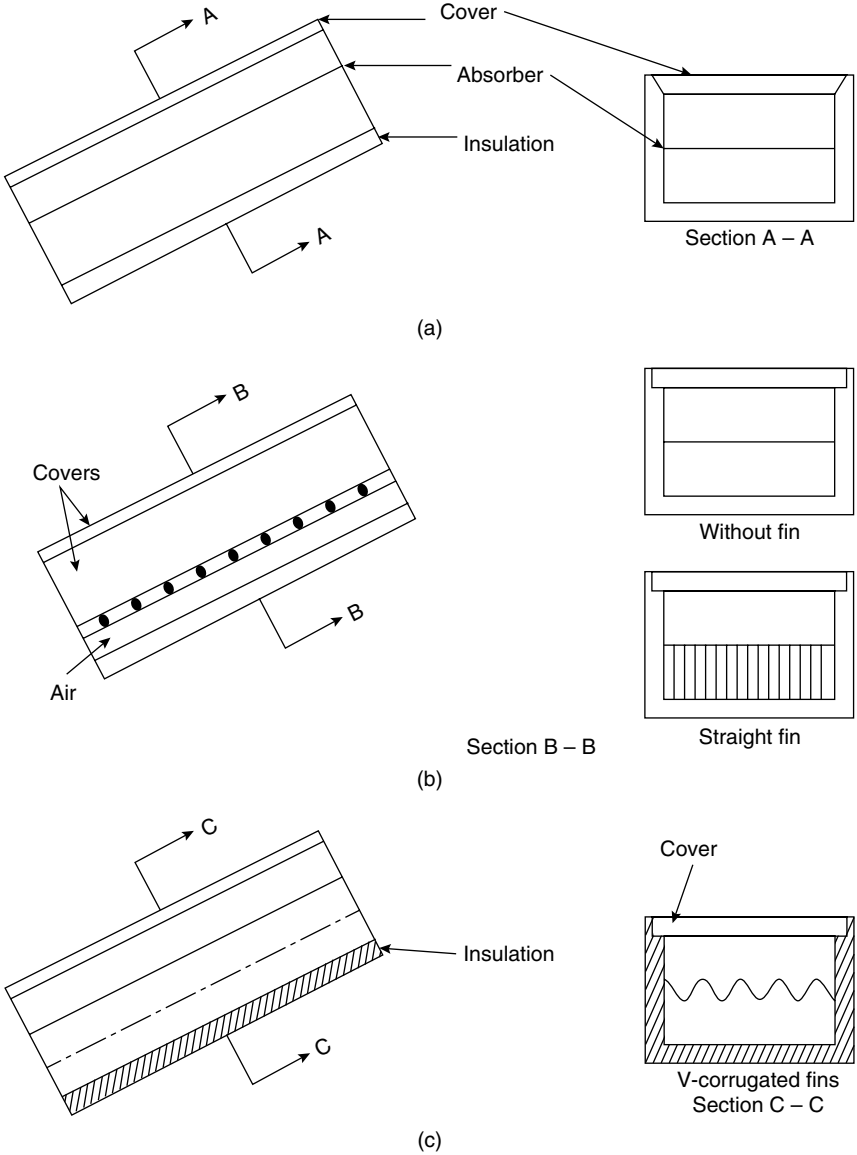


Fig. 14.14 *Non-Porous Absorber Plate-Type Collectors*

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convection losses from the cover plate. This type is not recommended if the air inlet temperature rise at the collector is too large.

The collection efficiency is increased by applying selective coating. Due to low heat transferrates, efficiencies are lower than liquid solar heaters. Performance of air heaters can be improved by

(i) Roughening the rear of the plate to promote turbulence and improving the convective heat transfer coefficient.

(ii) Adding fins to increase the heat transfer surface. This increases the turbulence, increasing the heat transfer coefficient (convective heat transfer).

A solar collector with V-corrugated copper foil shown in Figure 14.14(c) increases the absorption of solar radiation due to surface radioactive characteristics and the geometry of corrugations.

2. *Porous absorber plate-type collectors*

The main drawback of the non-porous absorber plate is the necessity of absorbing all incoming radiation over the projected area from a thin layer over the surface, which is in the order of a few microns. If the coatings are not used collection efficiency decreases. Also, excessive turbulence results in pressure drop. Too many surfaces and too much restrictions to air flow will require a larger fan and large amount of energy to push the air through. These difficulties are overcome by using porous type collectors.

In porous absorbers, the solar radiation penetrates to greater depths and is absorbed gradually depending on the matrix density. The cool air stream introduced from the upper surface of the matrix is first heated by the upper layers, which are cooler than the bottom layers. The air stream warms up while traversing the matrix layers. The lower matrix layers are hotter than the upper layers, and hence the air stream can effectively transfer heat from the matrix. Improper selection of the matrix porosity and the thickness may result in reduced efficiencies since the additional matrix layers beyond optimum may not absorb the solar radiation and heat the air stream further (Figure 14.15).

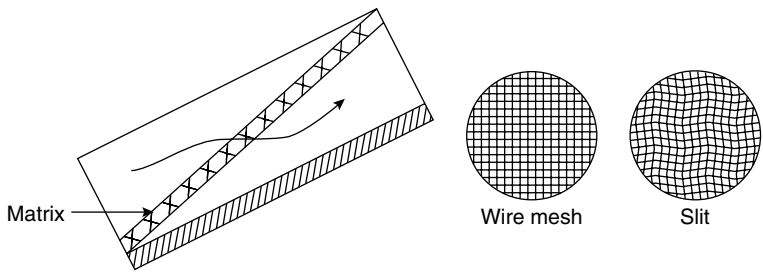


Fig. 14.15 *Slit or Expanded Metal (Porous-Type Absorber)*

Figure 14.16 shows solar air heating by utilizing a transpired honeycomb, which has more flow cross section.

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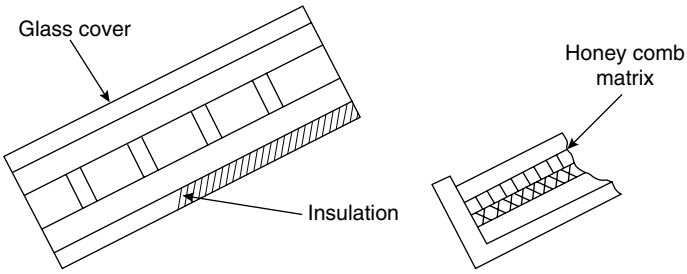


Fig. 14.16 *Solar Air Heating by Utilizing a Transpired Honeycomb*

Figure 14.17 shows another method suggested by Whillier, which uses crushed glass layers to absorb solar radiation and to heat air. A porous bed is formed by using layers of broken bottles (bottom dark top clear glass), which may be used for agricultural drying purposes with minimal expenditure.

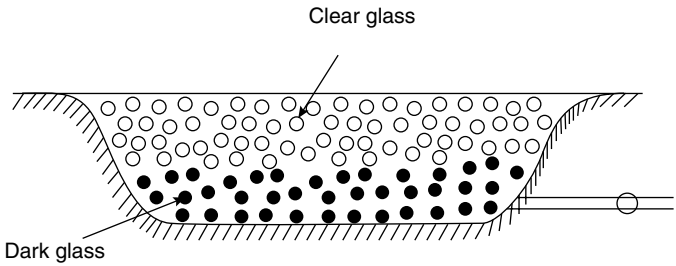


Fig. 14.17 *Porous Absorber-Type Air Heater*

Figure 14.18 shows an overlapped glass plate air heater in which overall flow direction is along the absorber glass plates, instead of being across the matrix. Plate and air stream temperature increase gradually along the collector length and across from top to bottom. This reduces the thermal loss significantly and also pressure drop.

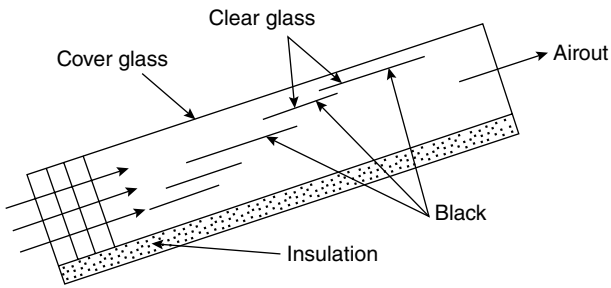


Fig. 14.18 *Overlapped Glass Plate Air-Heating Collector*

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14.2.8 Photovoltaic Power System

Photovoltaic (PV) technology converts one form of energy (sunlight) into another form (electricity) without using moving parts, consuming no conventional fossil fuels, creating no pollution and lasting for decades with very little maintenance. The energy from the sun is widely available and reasonably reliable with no associated storage or transportation difficulties and no emissions.

Solar cells operate on the principle based on semiconductor technology that electricity will flow between two semiconductors when they are put into contact with each other and exposed to light (photons). This phenomenon, known as the PV effect, was first discovered by Edmund Becquerel in 1839.

Although a PV array produces power when exposed to sunlight, a number of other components are required to properly conduct, control, convert, distribute and store the energy produced by the array.

The specific components required for the system may include a DC–AC power inverter, battery bank, system and battery controller, auxiliary energy sources and sometimes the specified electrical load (appliances).

Figure 14.19 show a basic diagram of a PV system and the relationship of individual components. Some PV systems use batteries for the purpose of storing energy produced by the PV array during the day, and to supply it to electrical loads as needed (during the night and periods of cloudy weather). Apart from this batteries are used in PV systems to operate the PV array near its maximum power point, to power electrical loads at stable voltages and to supply surge currents to electrical loads and inverters. In most cases, a battery charge controller is used in these systems to protect the battery from overcharge and over-discharge.

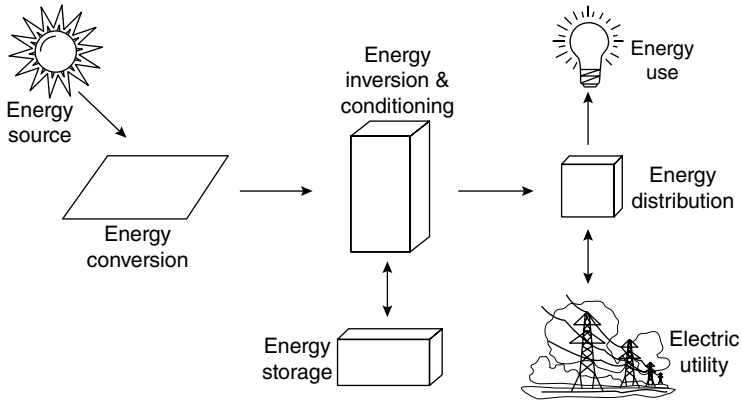


Fig. 14.19 *Major Photovoltaic System Components*

PV power systems are generally classified according to functional and operational requirements, their component configurations and how the equipment is connected to other power sources and electrical loads. The two principal classifications are grid-connected or utility-interactive systems and stand-alone systems. PV systems can be designed to provide DC and/or AC power service, can operate interconnected with or independent of the utility grid and can be connected with other energy sources and energy storage systems.

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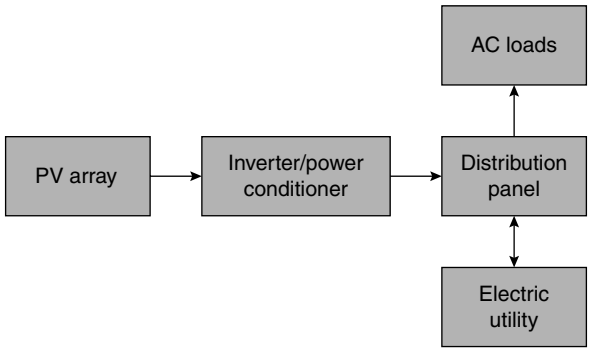


Fig. 14.20 *Diagram of Grid-Connected Photovoltaic System*

Grid-connected or utility-interactive PV systems operate in parallel with and interconnected with the electric utility grid. The primary component in grid-connected PV systems is the inverter, or power conditioning unit (PCU). The PCU converts DC power produced by the PV array into AC power consistent with the voltage and power quality requirements of the utility grid, and automatically stops supplying power to the grid when the utility grid is not energized. Figure 14.20 shows a schematic block diagram of grid-connected PV system. A bidirectional interface is made between the PV system AC output circuits and the electric utility network, typically at an on-site distribution panel or service entrance. This allows the AC power produced by the PV system to either supply on-site electrical loads or to back-feed the grid when the PV system output is greater than the on-site load demand. At night and during other periods when the electrical loads are greater than the PV system output, the balance of power required by the loads is received from the electric utility. This safety feature is required in all grid-connected PV systems, and ensures that the PV system will not continue to operate and feedback into the utility grid when the grid is down for service or repair.

Stand-alone PV systems

Stand-alone PV systems operate independent of the electric utility grid, and are normally designed and sized to supply certain DC and/or AC electrical loads. These types of systems may be powered by a PV array only, or may use wind, an engine-generator or utility power as an auxiliary power source (known as a PV-hybrid system). Figure 14.21, shows the simplest type of a stand-alone PV system (direct-coupled system) in which the DC output of a PV module or array is directly connected to a DC load. Since there is no electrical energy storage (batteries) in direct-coupled systems, the load operates only during sunlight hours, making these designs suitable for common applications such as ventilation fans, water pumps and small circulation pumps for solar thermal water-heating systems. For better performance, matching the impedance of the electrical load to the maximum power output of the PV array is critical. For certain loads such as positive-displacement water pumps, a type of electronic DC–DC converter, called a maximum power point tracker, is used between the array and load. This helps better utilization of the available array to get maximum power output.



Fig. 14.21 *Direct-Coupled PV System*

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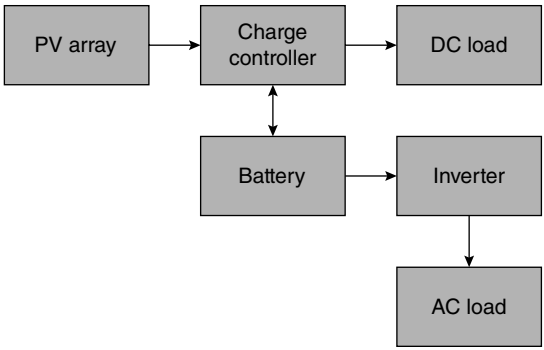


Fig. 14.22 *Diagram of Stand-Alone PV System with Battery Storage Powering DC and AC Loads*

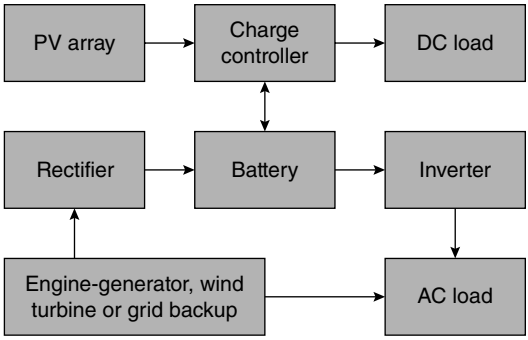


Fig. 14.23 *Diagram of Photovoltaic Hybrid System*

In many stand-alone PV systems, batteries are used for energy storage as shown in Figure 14.22. This stand-alone PV system powers both DC and AC loads. Figure 14.23 shows a typical diagram of a PV hybrid system.

PV or solar cells are semiconductor devices used to convert sunlight into DC electricity. Groups of PV cells are electrically configured into modules and arrays, which can be used for various applications, namely to charge batteries, operate motors and power any number of electrical loads. With the appropriate power conversion equipment, PV systems can produce AC compatible with any conventional appliances, and can operate in parallel with, and interconnected to, the utility grid.

A typical silicon P–V cell is shown in Figure 14.24. The PV cell is composed of a thin wafer made of an ultra-thin layer of phosphorus-doped (N-type) silicon placed on top of a thicker layer of boron-doped (P-type) silicon. An electrical field is created near the top surface of the cell at the point of contact of these two materials (P–N junction). When sunlight strikes the surface of a P–V cell, the generated electrical field provides momentum and direction to light-stimulated electrons. This in turn results in a flow of current when the solar cell is connected to an electrical load.

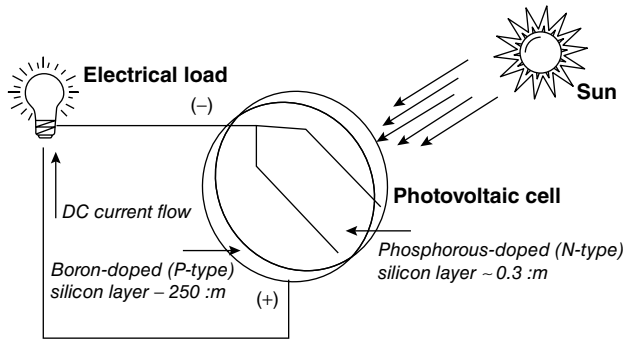


Fig. 14.24 *Diagram of a Photovoltaic Cell*

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Regardless of size, a typical silicon P–V cell produces about 0.5–0.6 volt DC under open-circuit and no-load conditions. The current and power output of a PV cell depends on its efficiency and surface area. It is proportional to the intensity of sunlight striking the surface of the cell. Under peak sunlight conditions, a typical commercial P–V cell with a surface area of 160 cm^2 will produce about 2 W of peak power. If the sunlight intensity were 40 per cent of peak, this cell would produce about 0.8 W.

To produce higher voltages, currents and power levels, PV cells are connected electrically in series and/or parallel circuits. The fundamental building blocks of PV systems are PV modules, which consist of PV cell circuits sealed in an environmentally protective laminate. PV panels include one or more PV modules assembled as a pre-wired, field-installable unit. A PV array is the complete power-generating unit, consisting of any number of PV modules and panels. The sequence in which an array is prepared is shown in Figure 14.25.

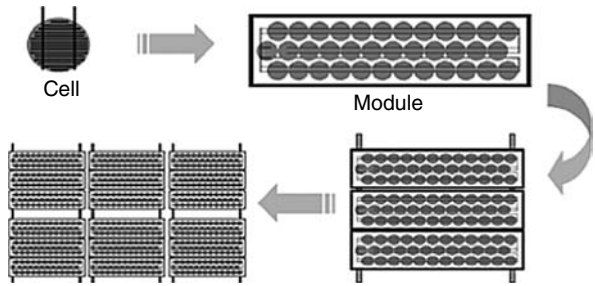


Fig. 14.25 *Photovoltaic Cells, Modules, Panels and Arrays*

The performance of PV modules and arrays is generally rated according to their maximum DC power output (watts) under Standard Test Conditions (STC).

Note: Standard Test Conditions refer to a module (cell) operating temperature of 25°C, and incident solar irradiance level of 1000 W/m^2 and under air mass 1.5 spectral distribution. However, actual performance is usually 85–90 per cent of the STC rating, since these conditions are not always typical of how PV modules and arrays operate in the field.

PV modules produced now-a-days are extremely safe and reliable, with minimal failure rates and have a projected service lifetime of 20–30 years. Most major manufacturers offer warranty of 20 or more years for maintaining a high percentage of initial rated power output.

14.2.9 Solar Central Receiver System

Working principle of a central receiver CSP plant

In a central receiver system, collecting solar energy is based on concentrating the sun's energy onto a common focal point to produce heat to run a steam turbine generator. Hundreds of large mirror assemblies, called heliostats, are used to track the sun to reflect the solar energy onto a tower where a black receiver absorbs the heat. High-temperature HTF is used to transport the heat to a boiler where the steam is used to spin a series of turbines, similar to a traditional steam power plant. This solar thermal storage system improves handling of the central tower plants and considerably increases its capacity factor to 70 per cent or more. Two Power Tower pilot project in Barstow, CA, USA, were commissioned by SQM in 1975, where they were used as HTF and heat storage medium (Figure 14.26).

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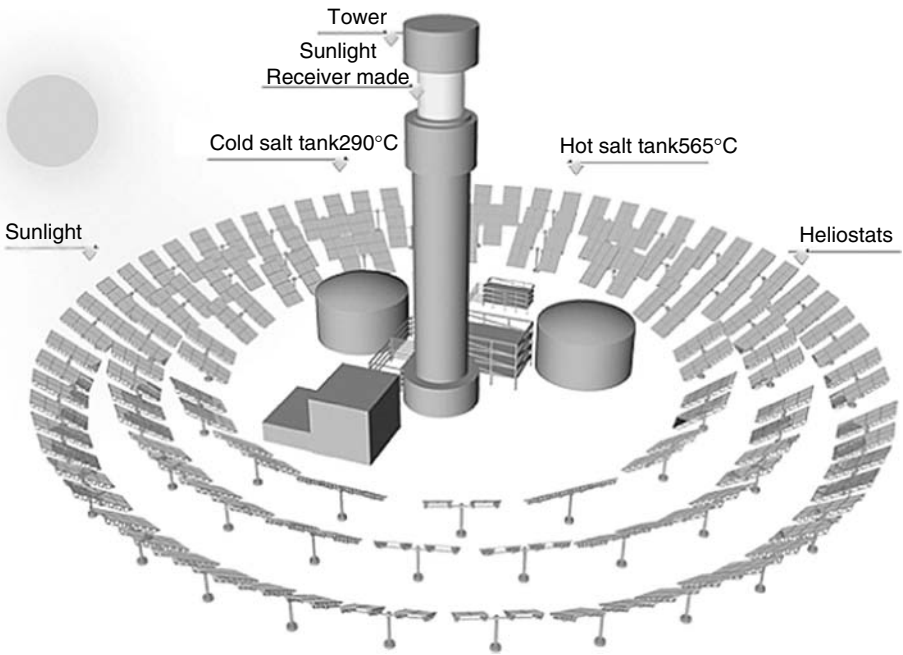


Fig. 14.26 *Schematic Diagram of CSP Central Receiver Plant*

1. *Parabolic trough*

Roughly 60 per cent of the applications use a parabolic trough system (Figure 14.27). Parabolic trough plants achieve at least a 25 per cent capacity factor, meaning about a quarter of the sun's energy that is captured by the system is converted to useful electricity. This system is composed of:

(i) Long parabolic shaped mirrors made of 4 mm thick glass with high transmittance properties having a reflective silver layer on the backside of the glass.

(ii) A receiver tube that runs along the length of the mirrors.

(iii) A tracking support structure and drive components for controlling the movement of the collector throughout the day according to the position of the sun. These components are collectively called as solar collector assembly (SCA). The SCA's can be kept approximately 25–30 feet above ground.

Working principle

The mirrors direct the sun's light to a focal point where the energy is concentrated onto the linear receiver tube, or heat collection element (HCE). The HCE is made of stainless steel, coated with a special absorber surface. This maximizes transfer of heat from the sun to the HTF travelling inside the tube. The HTF used could be either a high-temperature oil or a mixture of water and ethylene glycol. The HTF is supplied to the power plant where it passes through a series of heat exchangers, turning water into high-pressure steam that drives a steam turbine. The used HTF is then returned to the solar collector field and is heated once again.

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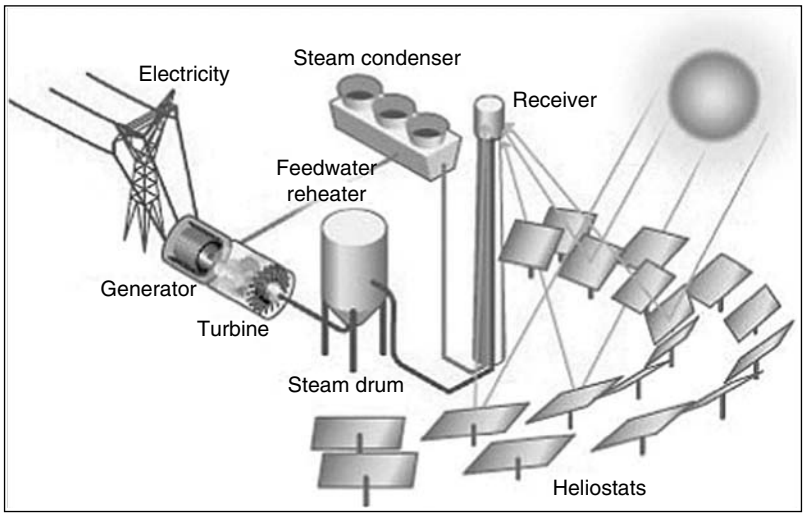


Fig. 14.27 *Power Tower Power Plant*
(*Source: US Department of Energy*)

2. *Power tower*

There are several applications that use power tower systems (Figure 14.27). Power tower technology has a capacity factor of 34 to over 40 per cent.

Working principle

In this system, a large field of mirrors, called heliostats, tracks the sun. A tower collects light concentrated by the heliostats onto a central receiver on top of a tower. Tower heights range from approximately 90–200 m. The HTF, composed of either water or molten nitrate salt, moves through the receiver and is heated to temperatures over 500°C. The heated HTF further enters a heat exchanger where water is turned into steam, which then drives a turbine generator.

Advanced systems now-a-days use molten salt as the HTF, extracting the advantage of the higher heat capacity of the fluid and can store the heat energy, which allows the system to continue to generate electricity during cloudy weather or at night. Thermal storage allows systems to continue to generate electricity for several hours longer compared with those having no such system. This increases the systems' capacity factor. Further, power tower systems use dry cooling as against a wet cooling technology, requiring less water to operate the system.

The amount of solar energy collected is a function of the number of heliostats installed. However, as the number of installed mirrors increases, the height of the tower must also increase. Determining the optimal tower height and field size is driven by system economy. It is relatively inexpensive to increase equipment size once a project has incurred its initial fixed costs of installation. Larger plants, therefore, tend to be more economical. Additionally, the heliostats can be mounted in ground with up to 5 per cent slope because they do not rely on a linear collector to heat the HTF. Like the parabolic trough systems, HTF is an integral part of the power tower system.

3. *Dish/engine*

Some applications make use of dish/engine systems as shown in Figure 14.28. This system consists of a large mirrored dish (also known as a solar collector), a receiver and a small engine.

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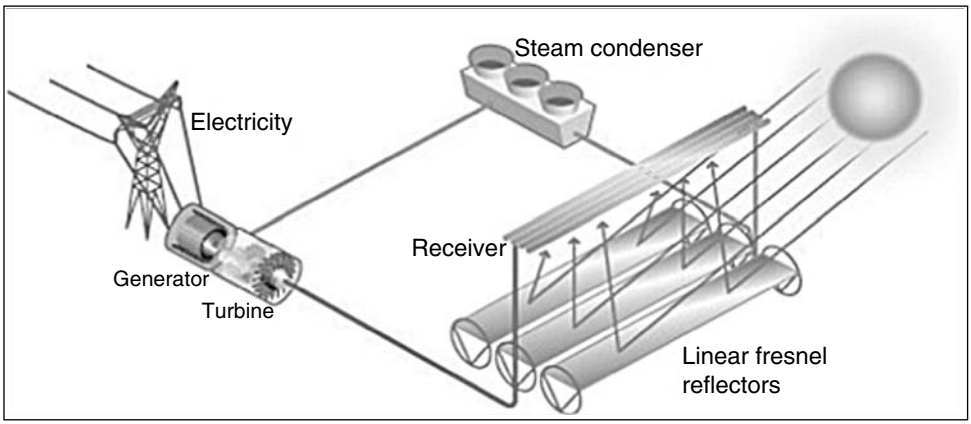


Fig. 14.28 *Linear Fresnel Reflector Power Plant*
(*Source: US Department of Energy*)

The receiver consists of a series of tubes that are filled with a heat transfer medium, namely hydrogen or helium. The engine used is a Sterling engine.

Working principle

The dish is mounted on a tracking system that follows the sun throughout the day. Concentrated sunlight heats the fluid in the receiver and transfers energy to the engine. The hot fluid is used to move engine pistons (turbine), which generate mechanical power. The power generated is used to run a generator to produce electricity. The waste heat from the engine is dissipated by a radiator system. The cooled medium is then recycled to the engine and the process repeats.

The following table 14.2 gives the capacity factors of different concentrated solar power plants.

Table 14.2 *Efficiency Comparison of Concentrated Solar Power Technology*

Technology type	Capacity factor (%)
Parabolic trough	25
Power tower	34
Dish engine	31

14.3 WIND ENERGY

Wind is a free and renewable form of energy, which throughout history has been used to grind grain, power ships and pump water. Wind is created when the sun unevenly heats the earth's surface. Thus, wind energy is a form of solar energy. The cost of using wind to generate electricity has been significantly reduced over the past decades due to technical improvements and mass production.

Since ancient times, people have harnessed wind energy. Over 5000 years ago, the ancient Egyptians used wind to sail ships on the Nile River. Later, people built windmills to grind wheat and other grains. The earliest known windmills were in Persia (Iran). These early windmills

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looked like large paddle wheels. Centuries later, the people of Holland improved the basic design of the windmill. They gave it propeller-type blades, still made with sails. Holland is famous for its windmills.

American colonists used windmills to grind wheat and corn, to pump water and to cut wood at sawmills. As late as the 1920s, Americans used small windmills to generate electricity in rural areas without electric service. When power lines began to transport electricity to rural areas in the 1930s, local windmills were used less and less, though they can still be seen on some Western ranches.

The oil shortages of the 1970s changed the energy picture for the country and the world. It created an interest in alternative energy sources, paving the way for the re-entry of the windmill to generate electricity.

14.3.1 Generation of Wind Energy

Wind is simple air in motion. It is caused by the uneven heating of the earth's surface by the sun. Since the earth's surface is made of very different types of land and water, it absorbs the sun's heat at different rates.

During the day, the air over land heats up more quickly than the air over water. The warm air over the land expands and rises, and the heavier, cooler air rushes in to take its place, creating winds. At night, the winds are reversed because the air cools more rapidly over land than over water.

In the same way, the large atmospheric winds that circle the earth are created because the land near the earth's equator is heated more by the sun than the land near the North and South Poles. Today, wind energy is mainly used to generate electricity. Wind is called a renewable energy source because the wind will blow as long as the sunshines (Figure 14.29).

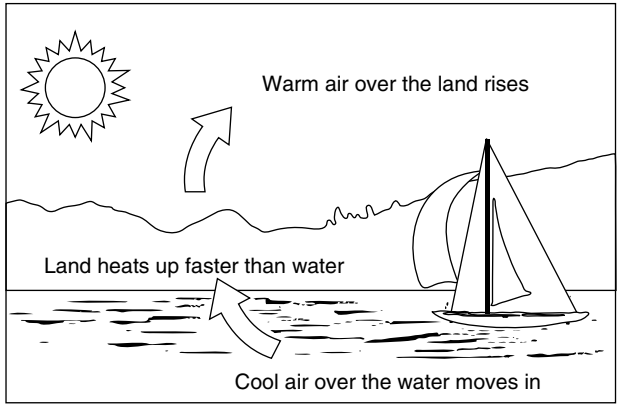


Fig. 14.29 *Generation of Wind Energy*

How does wind energy work?

Wind turbine blades capture wind energy, a form of mechanical energy, and put it to work turning a drive shaft, gearbox and generator to produce electrical energy. Many factors affect wind turbine efficiency including turbine blade aerodynamics (Figure 14.30).

Large utility-scale wind turbines can now generate more than a megawatt (10,00,000 W) of electrical power each and deliver electricity directly into the electric grid. These turbines are over

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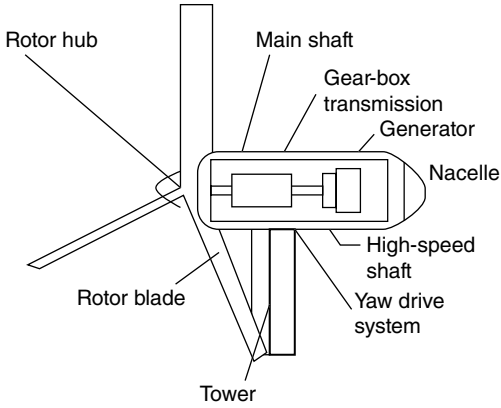


Fig. 14.30 *A Typical Wind Turbine*

200 feet high at the rotor hub and have blades that are 220 feet or more in diameter. Thus, the blades of a single turbine may sweep an area 80 per cent of the size of a football field. Utility scale over water where there is higher and more consistent wind speed. Each wind turbine is controlled by computer and in large projects is connected to a central computer where the turbines can be monitored. Wind turbines are designed with cut-in wind speeds and cut-out speeds (i.e. the wind speeds when the turbines start turning or shut off to prevent drive train damage). Typically, maximum electric generation occurs at speeds of 30–35 mph. Over the course of one year, well-suited wind power plants operate at an average of 30–35 per cent of their rated capacity (Figure 14.31).

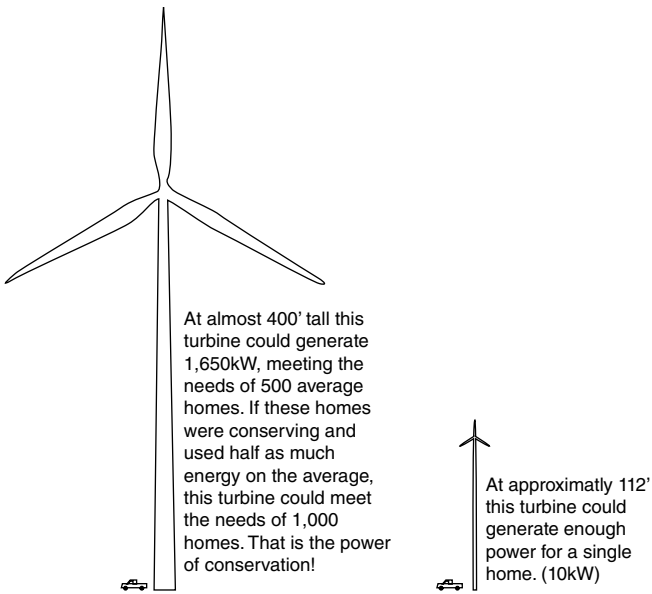


Fig. 14.31 *Wind Mill Power Generation*

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14.3.2 Wind Turbine Operation

Classification of wind conversion systems

Wind energy conversion systems can be classified as follows:

1. *Based on axis*
   (i) Horizontal axis machines: The axis of rotation is horizontal and the aero-turbine plane is vertical facing the wind.
   (ii) Vertical axis machines: The axis of rotation is vertical. The blades may also be vertical.
2. *Based on size*
   (i) Small scale: These might be used on farms, remote applications and other places requiring relatively low power. The generating capacity is upto 2 kW.
   (ii) Medium size machines: These wind turbines may be used to supply less than 100 kW rated capacity, to several residences or local use.
   (iii) Large scale or large machines: Large wind turbines are those of 100 kW rated capacity or greater. They are used to generate power for distribution in central power grids. They can have single generator at a single site or multiple generators sited at several places over an area.
3. *Based on output power*
   Wind aero-generators are classified based on output power as follows:
   (i) DC output
      (a) DC generator
      (b) Alternator rectifier
   (ii) AC output
      (a) Variable frequency, variable or constant voltage AC
      (b) Constant frequency, variable or constant voltage AC
4. *Based on rotational speed*
   Based on rotational speed of turbines
   (i) Constant speed with variable pitch blades using a synchronous generator with constant frequency output
   (ii) Nearly constant speed with fixed pitch blades using an induction generator
   (iii) Variable speed with fixed pitch blades for constant frequency output, using
      (a) Field modulated system
      (b) AC–DC–AC link
      (c) Double output induction generator
      (d) AC commutator generators
      (e) Other variable speed constant frequency-generating systems
5. *Based on the utilization of output*
   Based on rotational speed of turbines
   (i) Battery storage
   (ii) Direct connection to an electromagnetic energy converter

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(iii) Other forms (thermal potential, etc.) of storage

(iv) Interconnection with conventional electric utility grids

Advantages and disadvantages of wind energy conversion system

Advantages

(i) Wind power system is non-polluting and has no adverse influence on the environment.

(ii) It is a renewable source of energy.

(iii) Wind energy systems do not require fuel provision and transport.

(iv) On a large scale, it can be competitive with conventional electricity. Lower costs could be achieved by mass production.

Disadvantages

(i) Wind energy systems are noisy in operation; a large unit can be heard many kilometres away.

(ii) Wind power systems have a relatively high overall weight due to the existence of a gearbox, a hub and pitch changer, a generator, coupling shaft, etc. For large systems a weight of 110 kg/kW (rated) is required.

(iii) Wind energy needs storage capacity because of its irregularity.

(iv) Wind energy available is fluctuating in nature.

(v) They need larger area as propellers with 1–3 m in diameter, deliver power of 30–300 W range.

(vi) Systems are neither maintenance free nor-practically reliable.

14.3.3 Components of a Wind Generator

Figure 14.32 shows the main components of a wind energy conversion system (WECS).

Aero-turbines convert energy in moving air to rotary mechanical energy, which requires pitch control and yaw control. A mechanical interface consisting of a step-up gear and a suitable coupling transmits the rotary mechanical energy to an electrical generator. The output of the generator is connected to the load or power grid depending on the application.

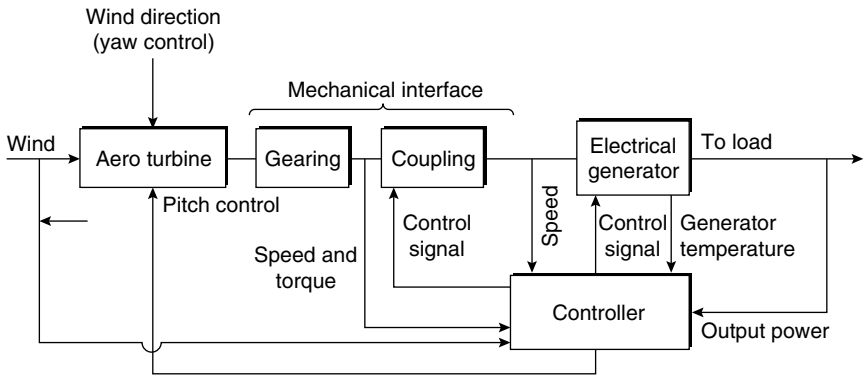


Fig. 14.32 *Basic Components of a Wind Electric System*

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Yaw control: In the locations with the prevailing wind in one direction, the design of a turbine can be greatly simplified. The rotor can be installed in a fixed orientation with the swept area perpendicular to the predominate wind direction. This machine is called yaw fixed. However, in most wind turbines yaw is active. As the wind direction changes, a motor rotates the turbine slowly about the vertical (or yaw) axis so as to face the blades towards the wind. The area of the wind stream swept by the wind rotor is a maximum (Figure 14.33).

In the small turbines, yaw action is controlled by a tail vane, whereas in larger machines, a servomechanism operated by a wind-direction sensor controls the yaw motor keeping the turbine properly oriented. The controller senses wind speed, wind direction, shaft speeds and torques, output power and generator temperature. Control signals are generated with the electrical output corresponding to the wind energy input. This protects the system from extreme conditions such as strong winds, electrical faults, etc.

Components of a wind electric generating system

Figure 14.33 shows the sub-components of a windmill. They are

(i) wind turbine or rotor
(ii) windmill head
(iii) transmission and control
(iv) supporting structure

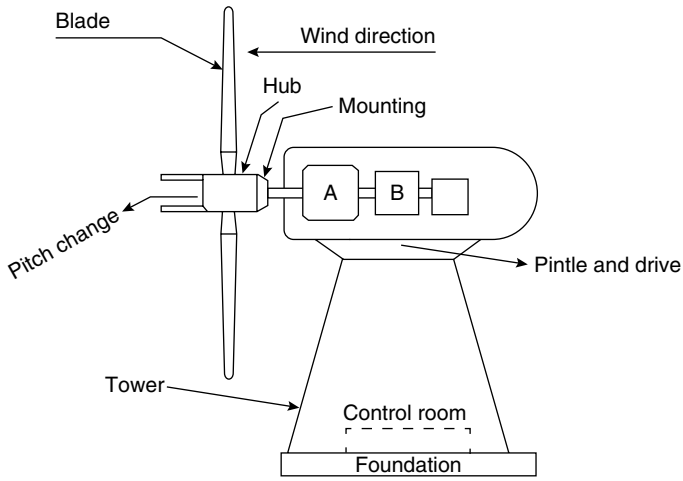


Fig. 14.33 *A Typical Wind Electrical Generating Station*

1. *Wind turbine or rotor*

Rotors of a WEC system can be divided into two types:

(i) Horizontal axis rotor
(ii) Vertical axis rotor

Vertical axis rotors can operate in all wind directions and thus need no yaw adjustment.

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2. *The windmill head*

It supports the rotor, housing and rotor bearings. It also has control mechanism such as changing the pitch of the blades for safety devices and tail vane to orient the rotor to face the wind.

3. *Transmissions and control*

The rate of rotation of large wind turbine generators operating at rated capacity or below are controlled by varying the pitch of the rotor blades. It has low rpm, about 40–50. However, optimum generator output requires much higher rates of rotation, about 1800 rpm. Hence, it is necessary to increase greatly the low rotor rate of turning using transmission mechanism. Transmission systems can be mechanical systems involving fixed ratio gears, belts and chains and/or in combination or hydraulic systems involving fluid pumps and motors. Fixed ratio gears are recommended for top mounted equipment because of their high efficiency and minimum system risk. For bottom mounted equipment requiring a right angle drive, the transmission costs can be reduced on the hub to increase rotor speed to generate speed. This offers a high degree of design flexibility as well as large savings.

Generator: Generators may be either constant or variable speed type. Variable speed units are expensive and/or unproved. Constant speed generators in use are synchronous induction and permanent magnet types. Synchronous unit is used for large aero-generator systems; it is very versatile and has an extensive database.

Controls: The modern large wind turbine generator requires a versatile and reliable control system. A control system is used for

(i) changing the orientation of the rotor into the wind
(ii) start up and cut-in of the equipment
(iii) power control of the rotor by varying the pitch of the blades
(iv) monitoring generator status data computation and storage
(v) shutdown and cut out owing to malfunction or very high winds
(vi) protection for the generator of the utility accepting the power and the prime mover
(vii) auxiliary and/or emergency power
(viii) maintenance mode

A control system may involve any one of the combination the following components:

(i) sensor – mechanical, electrical or pneumatic
(ii) decision elements – relays, logic modules, analog circuits, a microprocessor
(iii) actuators – hydraulic, electric or pneumatic are required

Tower: Four types of supporting towers generally used are

(i) the reinforced concrete tower
(ii) the pole tower
(iii) the built-up shell tube tower
(iv) the truss tower

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Among these, the truss tower is favoured because

(i) it is proved and widely adaptable

(ii) cost is low

(iii) parts are readily available

(iv) it is readily transported and it is potentially stiff

The type of the supporting structure and its height is related to cost and the transmission system incorporated. Horizontal axis wind turbines are mounted on towers so as to be above the level of turbulence and other ground-related effects. The minimum tower height for a small WECS is about 10 m and the maximum practical height is estimated to be roughly 60 m.

The turbine may be located either upwind or downwind of the tower.

Location

1. *Upwind location*

In the upwind location (i.e. the wind encounters the turbine before reaching the tower), the wake of the passing rotor blades causes repeated changes in the wind forces on the tower. Due to this, the tower may vibrate and may eventually be damaged.

2. *Downwind location*

If the turbine is downwind from the tower, vibrations are less but the blades are subjected to severe alternating forces as they pass through the tower wake.

Downwind rotors are generally preferred for the large aero-generators. Although other forces acting on the blades of these large machines are significant, tower effects are still important and tower design is an essential aspect of the overall system design. Both unwind and downwind locations have been used in WEC devices.

14.3.4 Velocity and Power from Wind

1. *Total power*

The total power of a wind stream is equal to the rate of the incoming kinetic energy of that stream KE, or

$$P_t = m \times \text{KE} = m \frac{V_i^2}{2g_c} \tag{1}$$

where P_t = total power, W, = mass-flow rate, kg/s, m_i = incoming velocity m/s and g_c = conversion factor = 1.0 kg/(N.s^2).

The mass-flow rate is given by the continuity equation

$$= \rho A V_i^1 \tag{2}$$

where ρ = incoming wind density, kg/m^3 and A = cross-sectional area of stream, m^2.

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$$\therefore \qquad P_t = \frac{1}{2g_c}\rho A V_i^3 \tag{3}$$

Thus, the total power of a wind stream is directly proportional to its density, area and the cube of its velocity. Figure 14.34 is a plot of P_t as a function of A and V_i for wind at 1 standard atmosphere pressure and 15°C.

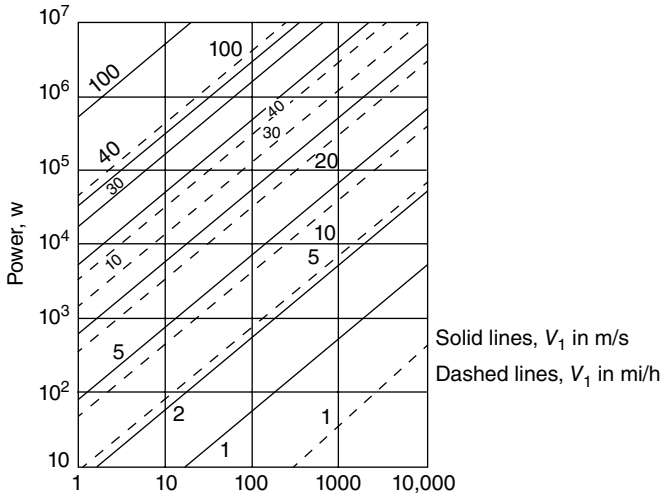


Fig. 14.34 *Total Power versus Area at T = 15° C*

2. *Maximum power*

The total power generated by the mill cannot all be converted to mechanical power. Consider a horizontal-axis and propeller-type windmill (wind turbine). Let the wheel of such a turbine has thickness $a - b$ (Figure 14.35) with wind pressure and velocity at turbine inlet be P_i and V_i. Let the wind pressure and velocity at the turbine exit be P_e and V_e, respectively; V_e is less than V_i because kinetic energy is extracted by the turbine.

Consider the incoming air between i and a, as a thermodynamic system it can be assumed that air density remains constant since the pressure and temperature changes are very small compared to ambient. Also, the change in potential energy is zero, and no heat or work are added or removed between i and a; the general energy equation is given by

$$P_i v + \frac{V_i^2}{2g_c} = P_a v + \frac{V_i^2}{2g_c} \tag{4}$$

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where v and ρ are the specific volume and the density, respectively, both considered constant. Similarly, for the exit region $b - e$ we have

$$P_e + \rho \frac{V_e^2}{2g_c} = P_b + \rho \frac{V_b^2}{2g_c} \tag{5}$$

The wind velocity across the turbine decreases from a to b since kinetic energy is converted to mechanical work there. The incoming velocity V_i does not decrease abruptly but gradually as it approaches the turbine to V_a and as it leaves it to V_e. Thus, $V_i > V_a$ and $V_b > V_c$. Hence, from Equations 4 and 5, $P_a > P_i$ and $P_b < P_e$; that is, the wind pressure rises as it approaches, then as it leaves the wheel.

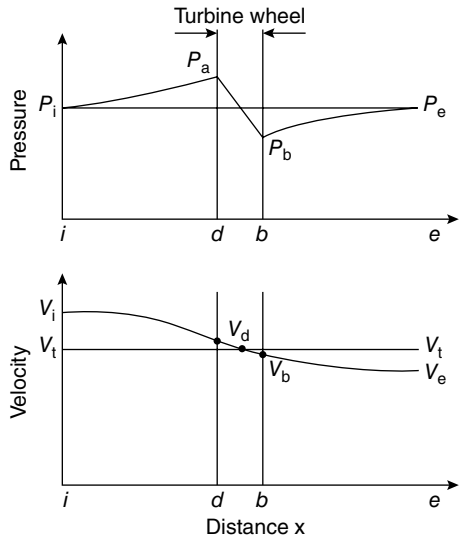


Fig. 14.35 *Pressure and Velocity Profiles of a Wind Moving Through a Horizontal-Axis Propellant-Type Wind Turbine*

Combining Equations 4 and 5 above

$$P_a - P_b = P_i + \rho \left[\frac{V_i^2 - V_a^2}{2g_c} \right] - P_i + \rho \left[\frac{V_i^2 - V_a^2}{2g_c} \right] \tag{6}$$

It is reasonable to assume that, far from the turbine at e, the wind pressure returns to ambient, or

$$P_e = P_i \tag{7}$$

The velocity within the turbine, V_t, does not change because the blade width $a - b$ is thin compared with the total distance considered.

$$\therefore V_t \approx V_a \approx V_b \tag{8}$$

Combining Equations (6)–(8)

$$P_a - P_b = \rho \left[\frac{V_i^2 - V_e^2}{2g_c} \right] \tag{9}$$

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The axial force F_x, in the direction of the wind stream, on a turbine wheel with projected area, perpendicular to the stream A, is given by

$$F_x = (P_a - P_b)\, A = \rho A \left[\frac{V_i^2 - V_e^2}{2g_c} \right] \tag{10}$$

This force is also equal to the change in momentum of the wind, where the mass-flow rate is given by

$$m = \rho A V_t \tag{11}$$

Thus

$$F_x = \frac{1}{g_c} \rho A V_t \left(V_i - V_e \right) \tag{12}$$

Equating Equations (10) and (12)

$$V_t = \frac{1}{2} \left(V_i + V_e \right) \tag{13}$$

Consider the total thermodynamic system bounded 1 by i and e. The changes in potential energy internal energy ($T_i = T_e$) and flow energy ($P_i v = P_e v$), are zero, and no heat is added or rejected. The general energy equation is reduced to

$$W = KE_i - KE_e = \left[\frac{V_i^2 - V_e^2}{2g_c} \right] \tag{14}$$

The power P is the rate of work. From Equation (11)

$$P = \dot{m} \left[\frac{V_i^2 - V_e^2}{2g_c} \right] = \frac{1}{2g_c} \times \rho A V \left(V_i^2 - V_e^2 \right) \tag{15}$$

Combining Equations (15) and (13)

$$P = \frac{1}{4g_c} \times \rho A \left(V_i + V_e \right) \left(V_i^2 - V_e^2 \right) \tag{16}$$

It can be observed that when $V_t = V_i, V_e = 0$ (the wind comes to a complete rest after leaving the turbine) (Figure 14.36). This is impossible because the wind cannot accumulate at turbine exit. It can be seen from Equation (16) that V_e is positive in one term and negative in the other that low or too high a value for V_e results in reduced power. There is an optimum exit

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velocity $V_{e\ opt}$ that results in maximum power P_{max}, which is obtained by differentiating P in Equation (16) with respect to V_e for a given V_i and equating the derivative to zero, that is, $dP/dV_e = 0$.

$$3V_e^2 + 2V_iV_e - V_i^2 = 0$$

Solving for V_e,

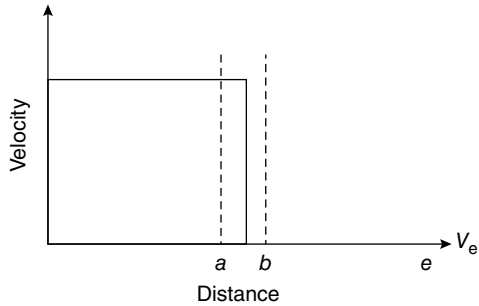


Fig. 14.36 *Total Conversion of Incoming Wind Kinetic Energy into Work*

$$V_{e\ opt} = \frac{1}{3}V_i \tag{17}$$

Combining Equations (17) and (16)

$$P_{max} = \frac{8}{27g_c}\rho A V_i^3 \tag{18}$$

The ideal, or maximum theoretical, efficiency η_{max} called power coefficient of a wind turbine is the ratio of the maximum power obtained from the wind to the total power of the wind.

$$\therefore \qquad \eta_{max} = \frac{P_{max}}{P_{tot}} = \frac{8}{27g_c} \times 2g_c = \frac{16}{27} = 0.5926 \tag{19}$$

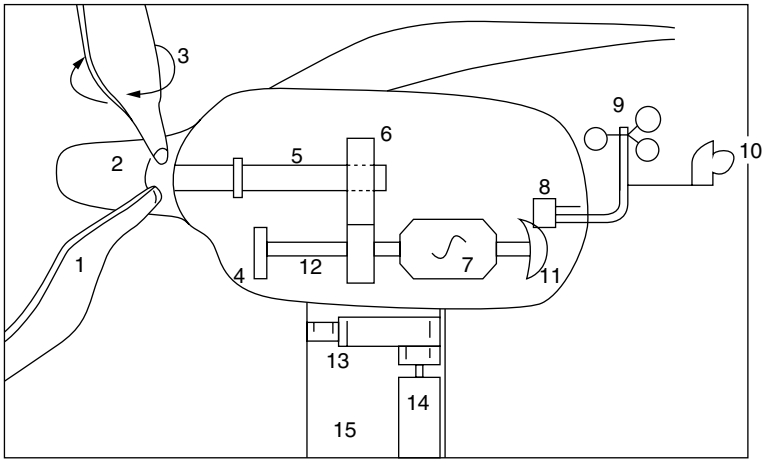
In other words, a wind turbine is capable of converting a less than 60 per cent of the total power of a wind to useful power (Figure 14.37).

3. *Actual power*

Wind-turbine experiences change in velocity dependent on the blade inlet angle and the blade velocity. Since the blades are long, the blade velocity varies with the radius to a greater degree than steam or gas-turbine blades and the blades are therefore twisted. The maximum efficiency (power coefficient) given by Equation (19) is under ideal conditions of the entire blade. Figure 14.38 shows the power coefficient for an ideal propeller-type wind turbine and various other wind turbines.

As a wind-turbine wheel cannot be completely closed, and because of spillage and other effects, practical turbines have 50–70 per cent of the ideal efficiency. The real efficiency η is the product of this and η_{max} and is the ratio of actual to total power

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1. Blade
2. Rotor
3. Pitch
4. Brake
5. Low-speed shaft
6. Gear box
7. Generator
8. Controller
9. Anemometer
10. Wind vane
11. Nacelle
12. High-speed shaft
13. Yaw drive
14. Yaw motor
15. Tower

Fig. 14.37 *Cut Section of a Wind Turbine Showing All Vital Components*

$$P = \eta P_{\text{tot}} = \eta \frac{1}{2g_c} A V_i^3 \tag{20}$$

where η varies between 30 and 40 per cent for real turbines.

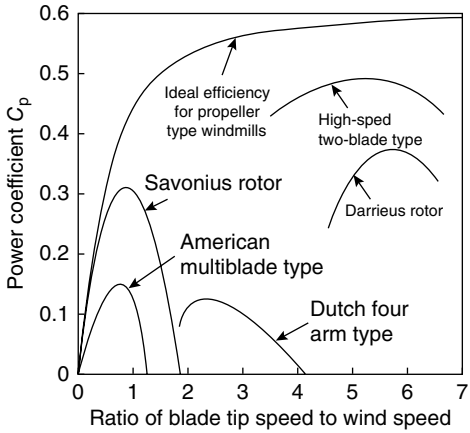


Fig. 14.38 *Power Coefficient of Various Windmills versus Ratio of Blade Tip to Wind Speeds*

4. *Forces on blades*

There are two types of forces operating on the blades of a propeller-type wind turbine. They are the circumferential forces in the direction of wheel rotation that provides the torque and the axial forces in the direction of the wind stream that provides an axial thrust that must be counteracted by proper mechanical design.

The circumferential force, or torque, T is given by

$$T = \frac{P}{\omega} = \frac{P}{\pi DN} \tag{21}$$

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where T = torque, N, ω = angular velocity of turbine wheel, m/s, D = diameter of turbine wheel = $\sqrt{4A/\pi}$, m and N = wheel revolutions per unit time, s^{-1}.

For a turbine operating at power, P, the torque is given by

$$T = \eta \times \frac{1\rho}{8g_c} \times \frac{DV_i^3}{N} \tag{22}$$

For a turbine operating at maximum efficiency $\eta_{max} = 16/27$, the torque is given by T_{max}

$$T_{max} = \frac{2}{27g_c} \times \frac{PDV_i^3}{N} \tag{23}$$

But the axial force, or axial thrust, is

$$F_x = \frac{1}{2g_c} \times \rho A\left(V_i^2 - V_e^2\right) = \frac{\pi}{8g_c} \times \rho D^2\left(V_i^2 - V_e^2\right)$$

The axial force on a turbine wheel operating at maximum efficiency, where $V_e = 1/3\ V_i$, is given by

$$F_{x\,max} = \frac{1}{2g_c} \times \rho A V_i^2 = \frac{\pi}{9g_c} \times \rho D^2 V_i^2 \tag{24}$$

From the above equation it is evident that the axial forces are proportional to the square of the diameter of the turbine wheel, making them difficult to cope with in extremely large-diameter machines. Hence, there is an upper limit of diameter that must be determined by design and economical considerations.

Example 14.7

A 10 m/s wind is at 1 standard atm pressure and 15°C temperature. Calculate (a) the total power density in the wind stream, (b) the maximum obtainable power density, (c) a reasonably obtainable power density, (d) the total power (in kW) produced if the turbine diameter is 120 m and (e) the torque and axial thrust N if the turbine were operating at 40 rpm and maximum efficiency.

Solution: For air, the gas constant $R = 287$ J/(kg.K), 1 atm = 1.01325×10^5 Pa

$$\text{Air density } \rho = \frac{P}{RT} = \frac{1.01325 \times 10^5}{287(15 + 273.15)} = 1.226 \text{ kg/m}^3$$

(a) The total power density

$$\frac{P_{tot}}{A} = \frac{1}{2g_c}\rho V_i^3 = \frac{1}{2 \times 1} \times 1.226 \times 10^3 = 613 \text{ W/m}^2$$

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(b) Maximum power density

$$\frac{P_{\max}}{A} = \frac{8}{27 g_c} \rho V_i^3 = \frac{8}{27 \times 1} \times 1.226 \times 10^3 = 363.26 \text{ W/m}^2$$

(c) Assuming $\eta = 40\%$

$$\frac{P}{A} = 0.4 \left(\frac{P_{\text{tot}}}{A} \right) = 0.4 \times 613 = 245.2 \text{ W/m}^2$$

(d) The total power

$$P = 0.245 \times \frac{\pi D^2}{4} = 0.245 \times \frac{\pi \times 120^2}{4} = 2770.88 \text{ kW}$$

(e) The torque and the axial thrust

$$T_{\max} = \frac{2}{27 g_c} \times \frac{\rho D V_i^3}{N} = \frac{2}{27 \times 1} \times \frac{120 \times 1.226 \times 10^3}{40/60}$$

$$= 16346.67 \text{ N}$$

Axial thrust is given by

$$F_{x\,\max} = \frac{\pi}{9 g_c} \times \rho D^2 V_i^2 = \frac{\pi}{9 \times 1} \times 1.226 \times 120^2 \times 10^2$$

$$= 616254.81 \text{ N}$$

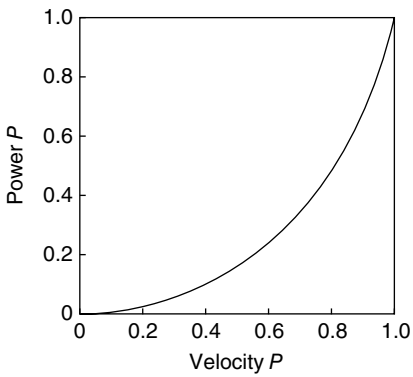


Fig. 14.39 *Power–Velocity Characteristics of a Wind Turbine*

14.3.5 Wind Turbine Operation

An inherent weakness of all wind machines is the strong dependence of the power produced on wheel diameter and wind speed, being proportional to turbine wheel area, that is, to the square of its diameter and to the cube of the wind velocity. The latter dependence means that even small fluctuations in the wind velocity would mean large fluctuations in power. For example, a drop in the wind velocity by only 20 per cent would result in the loss of a little less than half the power, whereas a drop in the wind velocity to approximately one-half (0.464) would result in a drop in power to one-tenth, assuming constant efficiency (Figure 14.39).

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1. *Flat rating*

Severe fluctuations in power are always undesirable, because they pose power-oscillation problems on the grid and severe strains on the windmill hardware. From an economic point of view, a windmill designed to produce a rated power output corresponding to the maximum, or near maximum, prevailing wind velocity at a given site would generate low powers, with the full capacity of the turbine and electric generator unused much of the time. It is, therefore more cost-effective design a windmill to produce rated power at less than the maximum prevailing wind velocity, using a smaller turbine and generator and to maintain a constant output at all wind speeds above rating, which is called flat rating.

2. *Cut-in and cut-out velocity*

Due to severe loss in efficiency and power at low wind velocities, a wind turbine is designed to come into operation at a minimum wind speed called the cut-in velocity. To protect the turbine wheel against damage at very high wind velocities, it is designed to stop operation (such as by feathering the blades) at a cut-out velocity. Thus, the wind turbine operates with variable load over a narrow range between the cut-in and the rated velocities and at constant power between the rated and the cut-out velocities and ceases operation above the cut-out velocity.

3. *Availability factor*

Wind turbine ratings are usually given for a wind velocity occurring at a reference height, usually 30 ft (9.1 m) above ground and with an availability factor of 90 per cent. The availability factor is defined as the fraction of time during a given period that the turbine is actually on line. The actual wind velocity at the propeller hub that determines the turbine power is usually higher.

4. *Overall load factor*

The overall load factor, also called the plant operating factor and the plant capacity factor, is the ratio of the total energy generated during a given period of time to the total rated generation capacity during the same period. This factor thus takes into account operations at less than rated wind velocity non-operation below cut-in and above cut-out velocities, and power outages caused by various situations such as repairs maintenance, etc. (Figure 14.40).

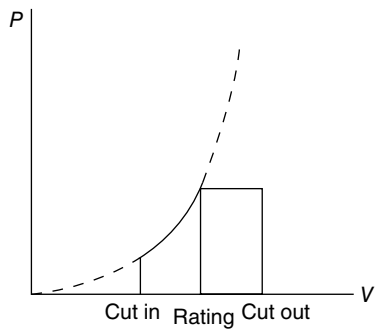


Fig. 14.40 *Power Velocity Characteristics of a Flat-Rated Wind Turbine*

14.3.6 Horizontal and Vertical Axis (HAWT, VAWT) Wind Mills

1. *Horizontal axis wind mill*

Horizontal axis-type wind machines are as follows:

(i) Horizontal axis with two aerodynamic blades: In this machine, rotor drives a generator through a step-up gear box. The blade rotor is designed to orient downwind of the tower. The components are mounted on a bed plate that is attached on a pintle at the top of the tower as shown in Figure 14.41. The rotor blades are continuously flexed by unsteady aerodynamic, gravitational and inertia loads, when the machine is in operation. If the blades are

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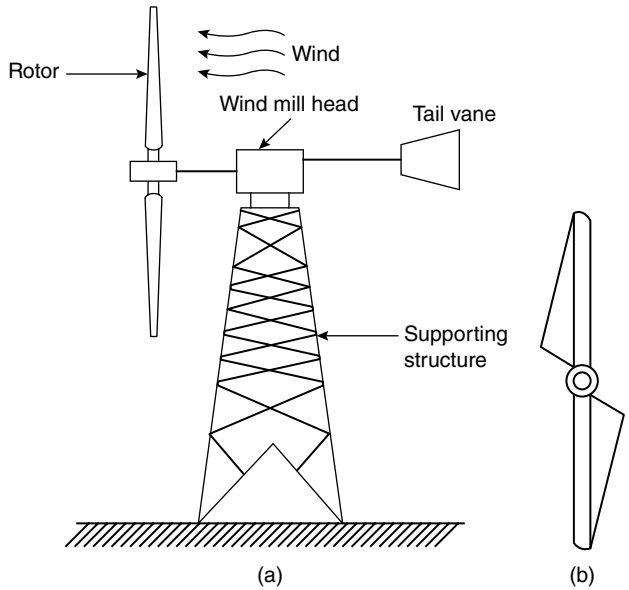


Fig. 14.41 *Horizontal Axis Wind Mill: (a) Horizontal Axis Type Wind Mill; (b) Two Blade Profile*

made of metal, flexing reduces their fatigue life. The tower is also subjected to above loads, which may result in serious damage. Because of the high cost of the blade rotors with more than two blades are not recommended. Rotors with more than two, say three or four, blades would have slightly higher power coefficient.

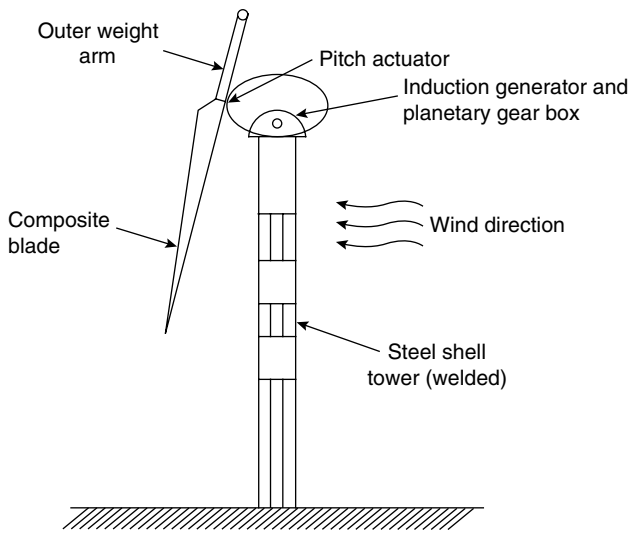


Fig. 14.42 *Horizontal Axis Single Blade Wind Mill*

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(ii) Horizontal axis propeller-type wind mill: It consists of a long blade mounted on a rigid hub, induction generator and gear box as shown. If extremely long blades (above say 60 m) are mounted on rigid hub, large blade root-bending moments occur due to tower shadow, gravity and sudden shifts in wind directions. To reduce rotor cost, use of low-cost counter weight is recommended that balance long blade centrifugally (Figure 14.42).

Advantages

(i) It has simple blade controls with lower blade weight and cost and lower gear box cost.

(ii) Counter weight costs less than a second blade. An inclined counter weight reduces blade coning.

(iii) Pitch bearings do not carry centrifugal force.

(iv) Blade root spar can be more rugged due to large diameter.

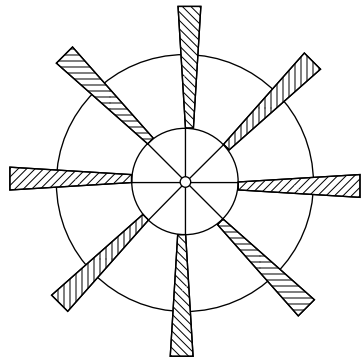


Fig. 14.43 *Horizontal Axis with Multi-Blade*

Disadvantages

(i) Large blade root-bending moment exists.

(ii) Due to flapping one-per-rev, Coriolis torque is produced.

(iii) Vibration problem due to aerodynamic torque.

(iii) Horizontal axis multi-bladed type wind mill: Figure 14.43 shows a horizontal axis multi-blade wind mill made from sheet metal or aluminium. The rotors have high strength to weight ratios. They have good power coefficient, high starting torque and added advantages of simplicity and low cost.

(iv) Horizontal axis wind mill Dutch type: Figure 14.44 shows a Dutch-type mill, which is one of the oldest in designs. The blade surfaces are made from an array of wooden slats that rotate at high wind speeds.

(v) Sail-type wind mill: The blade surface is made of cloth, nylon or plastics arranged as mast and pole or sail wings. There is also variation in the number of sails used (Figure 14.45).

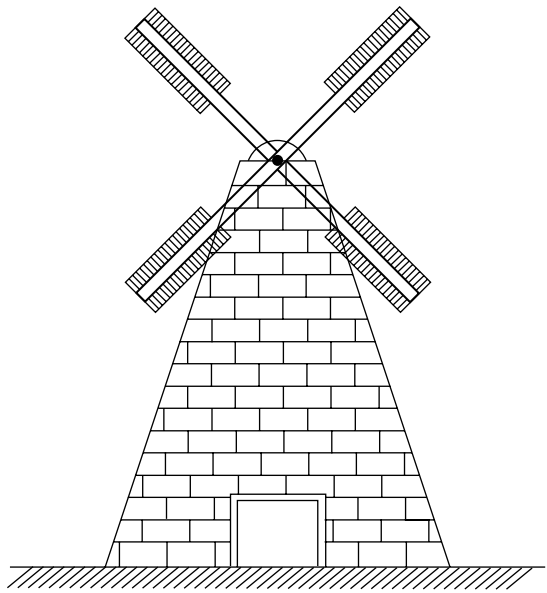


Fig. 14.44 *Horizontal Axis, Dutch-Type Wind Mill*

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2. *Vertical axis wind mill*

Some commonly used vertical axis wind mills are as follows:

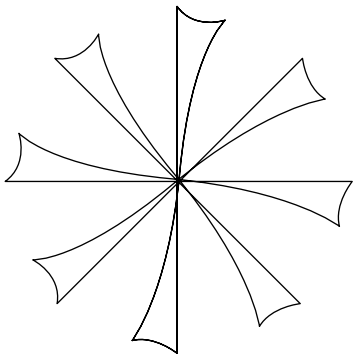


Fig. 14.45 *Blades of Sail-Type Wind Mill*

(i) *Cup anemometer*

Vertical axis rotors can be either drag- or lift-based. The cup anemometer is a drag-based, vertical axis wind device. The drag on a cup is greater when its concave side faces the wind, which causes the device to rotate. Lift also plays a small part; the cups crossing the wind experience a small lift because their convex surfaces deflect the wind and causes a pressure reduction. The main virtue of the cup anemometer is that it tends to rotate within a narrow range of TSRS under all conditions so its rotational speed is closely proportional to wind speed. However, it cannot carry a load with any efficiency and hence it was never constructed on a large scale for use as a wind turbine.

(ii) *The savonius rotor*

It is a simple WEC system that works like a cup anemometer. This type was invented by S.J. Savonius in the year 1920. It requires relatively low-velocity winds for operation. It consists of two half cylinders facing in opposite direction forming an S-shaped cross section as shown in Figure 14.46.

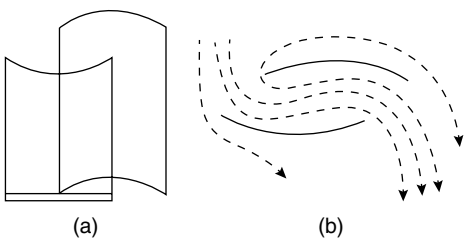


Fig. 14.46 *The Savonious Rotor*

These two semi-circular drums are mounted on a vertical axis perpendicular to the wind direction with a gap at the axis between the two drums. Irrespective of the wind direction, the rotor rotates such as to make convex sides of the buckets head into the wind. From the rotor shaft, generated power can be used for water pumping, battery charging, grain winnowing, etc.

Wind action: The force of the wind is greater on the cupped face than on the rounded face. The wind curving around the back side of the cupped face exerts a reduced pressure over the top of an air-foil and generates the rotation. The wide slot between the two inner edges of the half cylinders allows the air flow around inside the forward moving cupped face, thus pushing both in the direction of the rotation.

Characteristics of Savonius rotor are

(i) self-starting
(ii) low speed
(iii) low efficiency

Advantages

(i) A Savonius wind energy conversion system has a vertical axis and hence eliminates the expensive power transmission system from the rotor to the axis. Since it is a vertical axis machine, it does not matter much about the wind direction.

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(ii) The machine performs even at low wind-velocity ranges.

(iii) It has its low cut in speed, producing power effectively in winds as slow as 8 km/h, as against normal speed 16 km/h.

(iv) The Savonius rotor has the advantage that the weight of the electric generator may be carried at ground level without the use of level gears.

(v) The cost of vertical axis wind turbine is lower.

(vi) It has simple structure, hence is easy to manufacture.

(vii) A constant speed vertical axis wind turbine automatically stalls at high wind speeds without need for yaw and pitch controls.

(viii) Ground level mounting for the generator and gearing means easy access and maintenance and reduction in tower costs.

(ix) Overall weight of the turbine is less than that of conventional systems because of the small amount of material involved in relation to the swept area.

Disadvantages

(i) The machine is too solid having more metal or other material surface compared with the amount of wind intercepted. This leads to excessive weight for a large installation.

(ii) It is not useful for a very tall installation because of long drive shaft problems. Bracing of the topmost bearing above the rotor of a very tall vertical axis, machine is difficult requiring very long guy wires.

The Darrieus-type machines: This machine was invented originally and patented in 1925 by G.J.M. Darrieus, a French engineer. The Darrieus wind mill is a vertical axis machine that has the same advantage of a modern rapidly rotating propeller-type windmill, by use of an efficient airfoil, effectively intercepts large area of wind with a small blade area. It supports its blades in a way that minimizes bending stresses in normal operation.

Figure 14.47 shows a Darrieus wind mill. It has two or three thin curved blades with airfoil cross section and constant chord length. Both ends of blades are attached to a vertical shaft. Thus, the force in the blade due to rotation is pure tension. This gives stiffness to the wind forces it experiences. The blades can be made lighter than in the propeller type. When rotating, these airfoil blades provide a torque about the central shaft with respect to a wind stream. This shaft torque is transmitted to a generator at the base of the central shaft for power generation.

Darrieus-type rotors are lift devices characterized by curved blades with air foil cross sections. They have relatively low solidity and low starting torques, but high tip to wind speeds and therefore relatively high power outputs per given rotor weight and cost.

Characteristics of Darrieus Rotor are

(i) self-starting

(ii) high speed and efficiency

(iii) low capital cost

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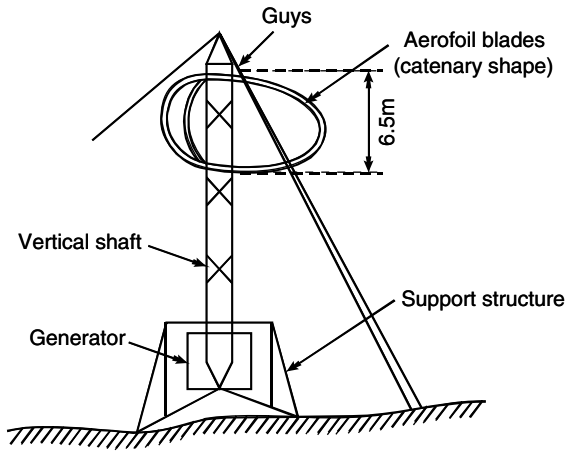


Fig. 14.47 *Vertical Axis Darrieus Wind Mill*

The force that propels the blades of a conventional wind mill comes from the chord of the airfoil, being tilted away from the direction of motion. The thrust acting at right angles to the airfoil is tilted towards the forward direction and has a component in that direction, 'forward thrust'. Even though the Darrieus rotor cannot have this tilt of the airfoil, yet it works without it. The chord is tangent to the circular path in the equatorial cross section shown in Figure 14.48. If the chord were tilted away from the tangent so as to tilt the thrust forward where the wind meets the airfoil on the wind ward side of the circle, as indicated by the broken lines in the figure in the other side of the circle, the wind would meet the other side of the airfoil and the thrust would be acting backward to retard the motion.

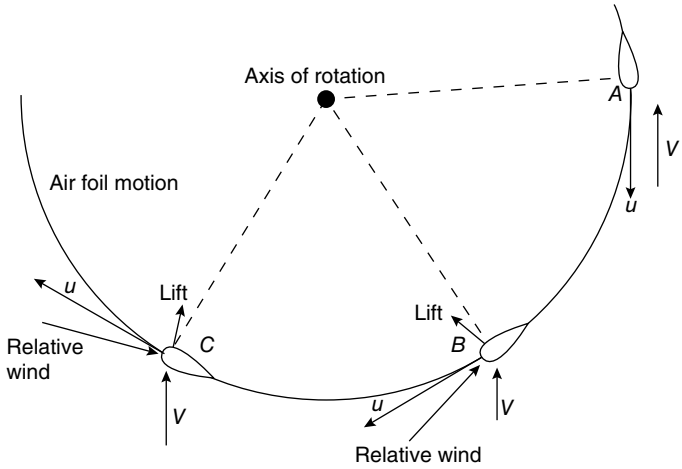


Fig. 14.48 *Darrieus Rotor Principle*

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Advantages

(i) The machine can be mounted on the ground. This facilitates the operation, close to the ground level eliminating tower structures and lifting of huge weight of machine assembly.

(ii) Due to vertical axis symmetry, it eliminates yaw control requirement for its rotor to capture wind energy.

(iii) The rotor blades can accept the wind from any direction.

(iv) Due to vertical shaft, accessibility and serviceability are easier.

(v) As pitch control is not needed for synchronous operation, it saves costs.

(vi) The tip speed ratio and power coefficient are higher than those of the S-rotor.

Disadvantages

(i) Darrieus machine requires external mechanical aid for start up. With small machines, the problem can be solved by attaching S-rotors at the top and bottom of the vertical (rotational) axis.

(ii) Since Darrieus rotor is usually placed near the ground, it experiences lower velocity wind compared to a tower-mounted conventional wind energy conversion system of comparable projected rotor disc area. This yields less energy output.

(iii) Rotor power output efficiency of a Darrieus wind energy conversion system is lower than that of a conventional horizontal rotor.

(iv) As a Darrieus rotor cannot be yawed out of the wind and its blades are feathered special high torque braking system must be incorporated.

(v) Darrieus rotor encounters greatly varied local flow conditions per revolution, greater vibratory stresses are encountered which will affect rotor system life.

3. *Magnus effect rotor*

This concept was first demonstrated by Magnus in 1912. It consists of spinning cylinders. When cylinders are spun in wind stream, translational forces are produced perpendicular to the wind stream by the Magnus effect. Such a device can be used to propel ships or land vehicles.

Principle

When a stationary horizontal cylinder is rotated about its axis in a cross wind, it will experience a lift-force. The effect is equally applicable to a vertical cylinder being rotated about its axis in a cross wind. It will experience a force perpendicular to its axis. This will cause it to move in a direction perpendicular to that of the wind. The resultant of the lift and drag forces (more in the direction of lift because of its larger value) propels the cylinders horizontally along a track (Figure 14.49).

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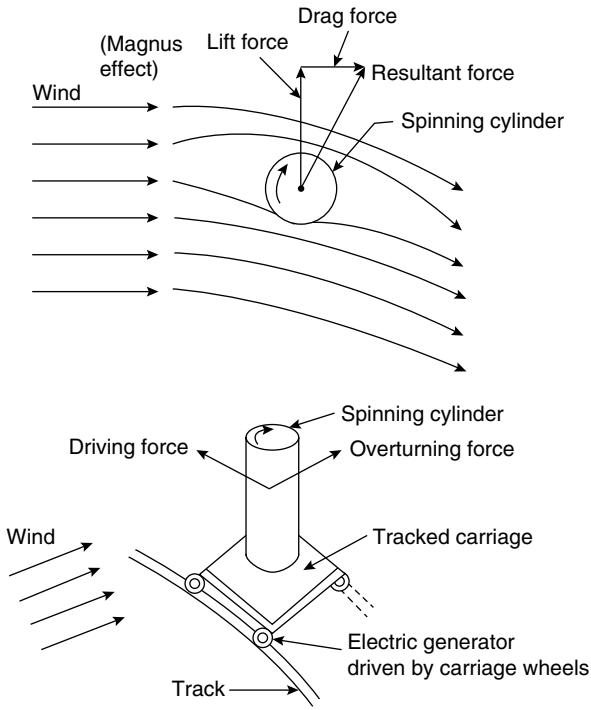


Fig. 14.49 *The Magnus Concept for Generating Electricity*

14.3.7 Aerodynamic Considerations of Wind Mill Design

The horizontal axis mills generally have better performance. They have been used for various applications including electric power generation and pumping water. The latter introduces some complexity into the design as the mechanical energy has to be transmitted over a distance. Also, in some cases, the rotor motion has to be converted to reciprocating motion. The individual components, such as the rotor transmission, generator and tower, should be as efficient as possible, and must function effectively in combination. Some of the main design considerations are outlined below.

Rotor: A wind turbine rotor can have any number of blades that may be made from wood, metal or composites of several materials including glass-reinforced plastic blades or carbon fibre, which combines light weight and flexibility with immense strength.

Horizontal axis rotors can be either lift or drag devices. Lift devices are generally preferred, since for a given swept area high rotational speeds and more output power can be developed by lift than by drag forces.

Drag devices are often capable of generating high torques (twisting forces) and are ideal for some uses such as water pumping. They are less suitable for electricity generation, which requires very high rotational speed (several thousand rpm).

All horizontal axis rotors depend mainly on lift to rotate them. Maximum efficiency is achieved when the angle of incidence approaches the maximum possible angle of attack prior to stalling (i.e. when the lift drag ratio is as high as possible). Given that a rotor's tip travels faster than the

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points nearer to the axis, the angle at which the wind meets the plane of rotation decreases in linear proportion to the radius. Hence, an efficient wind rotor requires the blades to be twisted so that the optimum angle of incidence is obtained at all radii when the machine is running at its designed speed. The ratio of the speed of the rotor tip to the speed of wind is called as tip speed ratio. Maximum efficiency is obtained with higher TSR. The tip speed ratio is given by

$$TSR \text{ (tip speed ratio)} = \frac{V_{\text{tip}}}{V}$$

where V_{tip} = speed of the rotor tip and V = free wind speed.

As the TSR increases, the number of blades decreases. Solidity is normally defined as the fraction of the total circumferential (either at the tip or sometimes at two-thirds of the tip radius) that contains blades. Numerically, it can be given by

$$S = NC/\pi D$$

where N is the number of blades, C is the average breadth of a blade and D is the diameter of the circle described by the blade.

A windmill rotor is characterized by comparing its curve of efficiency with its TSR. The efficiency is usually expressed as the C_p, known as power coefficient or performance coefficient, which is the fraction of wind energy passing through the rotor disc that is converted into shaft power. It is given by

$$C_p = 0.5\rho A V^3$$

Every type of rotor has a unique C_p curve. This is because shape of the curve is that the optimum C_p occurs when the rotor blades are operating at their maximum LDR lift drag ratio.

(i) At lower TSR, the blades stall progressively and although they can generate a lot of lift, they also produce much more drag, which dissipates energy.

(ii) At a higher than optimum TSR, they produce less than their maximum lift (and the lift forces act at a less favourable angle) because the angle of incidence decreases.

(iii) At runaway speed, the angle of incidence approaches zero, and the lift and drag forces balance, so that no net shaft power is produced and the efficiency once again is zero.

Windmills normally operate on the part of the curve to the right of the maximum C_p. They depend on an increasing wind speed to accelerate the rotor from rest through the peak C_p point.

Sometimes about from $C_p - TSR$ curve, the torque characteristic $C_T - TSR$ curve is also used to characterize wind rotors. A wind turbine will always operate at a speed that produces enough torque to balance the load's torque requirement exactly. Hence, if the characteristics of both the load and the rotor torque speed are known, the system performance can be defined based on torque coefficient.

Torque coefficient, C_T, is defined as the ratio of shaft torque to torque at maximum efficiency.

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$$C_T = \frac{T}{T_{max}} \tag{1}$$

where T = shaft torque and T_{max} = torque at maximum efficiency.

The maximum conceivable torque T on a turbine rotor occurs if the maximum thrust could be applied at the blade tip farthest from the axis. For a propeller turbine of radius R

$$T_{max} = F_{max}\, R \tag{2}$$

$$\text{and } F_{max} = \frac{1}{2}\rho A_1 V_1^2 \tag{3}$$

$$T_{max} = \frac{1}{2}\rho A_1 V_1^2 R \tag{4}$$

$$T = C_T T_{max}$$

The tip speed ration λ is defined as the ratio of the outer blade tip speed V_i to the unperturbed wind speed V_i;

$$\lambda = \frac{V_t}{V_i} = \frac{R\omega}{V_i} \tag{5}$$

where R is the outer blade radius and ω is the rotational frequency. Substituting for R in Equation (4)

$$T_{max} = \frac{\rho A V_i^2 \times V_i \lambda}{2\omega}$$

$$= P_a \lambda/\omega \tag{6}$$

where P_a is the power in the wind. If power derived from the turbine is P_t,

$$P_t = \tau.\omega \tag{7}$$

But $$P_t = C_p P_a$$

$$\therefore\ C_p P_a = C_T T_{max}\omega$$

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$$C_p P_a = C_T P_a \lambda$$

$$C_p = \lambda C_T \tag{8}$$

By the Betz criterion, the maximum value of C_p is 0.593, so in the 'ideal' case

$$C_{Tmax} = \frac{0.593}{\lambda} \tag{9}$$

Machines with higher speeds have a slightly higher maximum C_p but a much lower C_T, particularly for starting.

Number of blades: Wind turbines have been built with upto six propeller-type blades, but two- and three-bladed propellers are most common. A one-bladed rotor with a balancing counter weight has some advantages, including lower weight and cost and simpler controls, over the multi-bladed rotor. However, starting requires high wind speeds and vibration and other forces can be huge.

Turbines with three blades are used to avoid vibrations that occur due to the turning (or yawing) of the rotor in order to face it towards the wind. However, this problem can be overcome by controlling the yaw rate. Two-bladed systems are receiving major attention because they cost less to fabricate in large sizes and are capable of operating with a high tip speed ratio.

Blade design: Wind turbine blades have an airfoil-type cross section and a variable pitch as shown in Figure 14.50. They are slightly twisted from the outer tip to the root in order to reduce the tendency for the rotor to stall. The blades can also have a constant chord length. Better performance is obtained with blades that are narrower at the tip than at the root as shown in Figure 14.50.

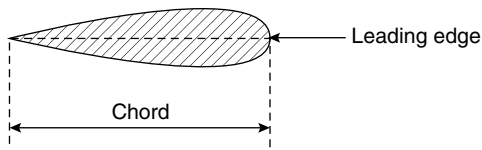


Fig. 14.50 *Cross-Section of Airfoil-Type Wind Rotor Blade*

The force that propels the blades of a conventional wind mill comes from the chord of the airfoil, being tilted away from the direction of motion. At the tip of the blades of a modern wind turbine the velocity is about six times the wind velocity. This means that the blades are set flat at a small angle with the plane of the rotation and almost at right angles to the direction of wind such that the effective wind properly approach from ahead of the leading edge. At other parts of the blade, between the tip and the axle, the velocity and the ideal set of the airfoil is at a greater angle to the plane of rotation. Twisted blades are ideal but not practical (Figure 14.51).

Wind stream variation: Wind speed variability must be considered in the design of the larger WEC machines.

Vertical axis rotors have the great advantage of not having to be turned into the wind stream as the wind direction changes, because their operation is independent of wind direction; vertical axis machines are called panemones (from Greek words meaning 'all winds').

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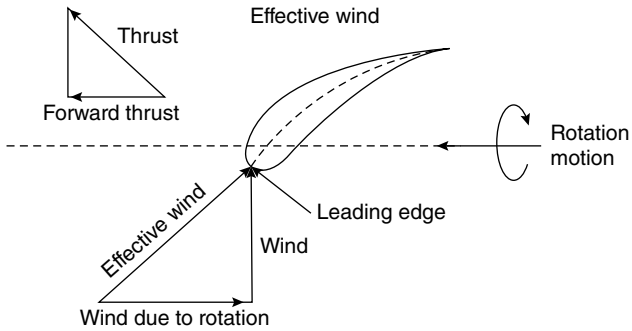


Fig. 14.51 *Wind Mill Blade as an Airfoil*

The advantages of vertical axis wind turbines over horizontal axis wind turbines are as follows:

(i) They can react to wind from any direction and do not need yawing equipment to turn the rotor into the wind.

(ii) They need minimum structural support because heavy components (such as gear box and generator) can be located at ground level.

(iii) Easy to install and maintain.

(iv) Rotor is not subjected to continuous cyclic gravity loads since the blades do not turn end over end.

Consider the aerodynamic blade shown in Figure 14.52. The blade can be treated as a typical cross-sectional element of a two-bladed aero turbine at some radius r from the axis of rotation, moving to the left. Because the blade is moving in the plane of rotation, it has tangential wind velocity, V_T, in the plane of rotation.

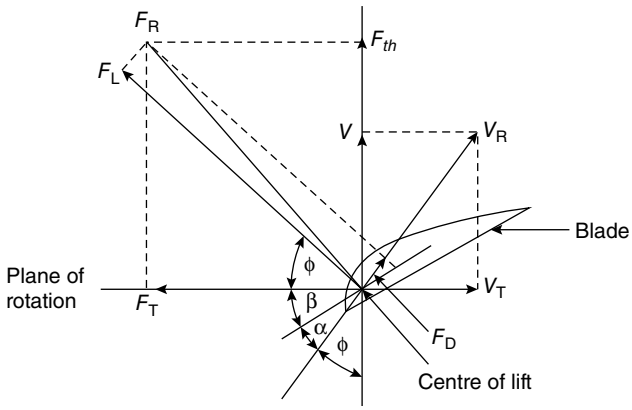


Fig. 14.52 *Forces on a Elemental Blade Reaction of an Aero Turbine (The Blade is Rotating to the Left)*

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Let

V = impinging wind velocity
V_T = wind velocity in plane of rotation due to blade turning
V_R = resultant wind velocity seen by aero turbine blade
F_L = lift force (Normal to V_R)
F_D = drag force (parallel to V_R)
F_T = torque producing component of F_R making aero turbine rotate
F_{th} = thrust force component of F_R
α = angle of attack of blade
β = blade pitch angle

This velocity component added vectorially to the incoming wind velocity gives the resulting wind velocity, V_R, seen by the rotating blade element. The lift force F_L caused by the aerodynamic shape of the blade at right angles to V_R. The drag force F_D is parallel to V_R. The vector sum of F_L and F_D is F_R, which has a torque-producing component, F_T, and a thrust-producing component. The former drives the aero turbine rotationally and the latter tends to flex the blade and overturn the aero generator. The vector diagram is centred on the centre of lift of the aerodynamic blade. α is the angle of attack of the aerodynamic element. It determines lift and drag forces and hence speed and torque output of the aero-turbine. These quantities can be varied by changing the blade pitch angle, β.

As V_T increases linearly with r, on an inclined aero-turbine blade, it is necessary to adjust β with r so as to have a positive angle of attack and to maintain reasonable stress levels within the blade. At larger r, β is made small whereas at smaller r, β is large. Thus, the blade 'bites' the air more near the base than the tips.

14.3.8 Coefficient of Performance of Wind Mill Rotor

As WECS is a capital intensive technology, it is desirable for the overall wind electric plant to have the highest efficiency possible optimally utilizing capital resources and minimizing the electric energy cost.

The overall conversion efficiency η_0 of an aerogenerator of the general type is

$$\eta_0 = \frac{\text{Useful output power}}{\text{Wind power input}} \tag{1}$$

$$= \eta_a, \eta_g, \eta_c, \eta_{gen}$$

where η_a = efficiency of the aero turbine, η_g = efficiency of the gearing, η_c = efficiency of the mechanical coupling and η_{gen} = efficiency of the generator.

Equation (1) is an application of cascaded energy conversion wherein overall efficiency will be strongly determined by the lowest efficiency converter in the cascade. For the aerogenerator, this is the aero turbine; the efficiency of the remaining three elements can be made quite high but less than 100 per cent.

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Consider an arbitrary aero turbine of cross-sectional area a driven by the wind. Its efficiency is given by

$$\eta_a = \frac{\text{Useful shaft power output}}{\text{Wind power input}}$$

$$= C_P$$

= Coefficient of performance

The coefficient of performance of an aero turbine is the fraction of power in the wind through the swept area, which is converted into useful mechanical shaft power. C_P for horizontal axis wind machine has theoretical maximum value = 0.593.

The convertible power of energy is proportional to the cube of the wind speed. Thus, if the wind speed decreases by 20 per cent, the power of output is reduced by almost 50 per cent. The wind speed may very considerable from day to day and from season to season. The efficiency of a wind generator depends first and foremost on the design of wind rotor and rotation speed, expressed as the ratio of blade tip speed to wind speed V_T/V. If n is the rotation frequency (rotations per second) and if is a rotor of diameter D metres, the tip speed is πnD m/sec (Figure 14.53).

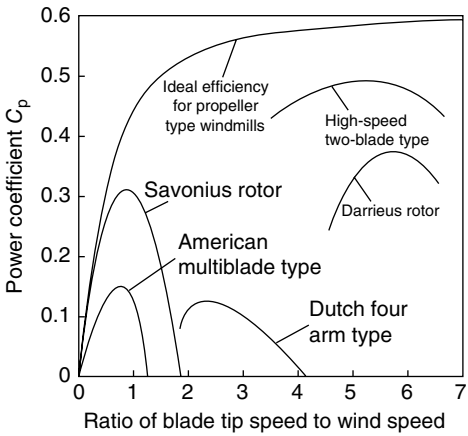


Fig. 14.53 *Typical Performance of Wind Machine*

Golding has derived the expression for aero-turbine efficiency as

$$\eta_a = \frac{1 - k\dfrac{V_T}{V}}{1 + k\dfrac{V}{V_T}} = C_p \qquad (2)$$

where $k = \dfrac{F_D}{F_L} = \dfrac{\text{Drag}}{\text{Lift}}$, V_T = wind velocity of blade element in the plane of rotation due to blade turning and V = impinging wind velocity.

If there were no drag, that is, $k = 0$, then the efficiency would be unity. In actuality, k can be made very small depending on the airfoil chosen and the angle of attack. Also, efficiency would be low if k were very large or small.

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14.3.9 Availability of Wind Energy in India

The development of wind power in India began in the 1990s and has significantly increased in the last few years. Although a relative newcomer to the wind industry compared with Denmark or the United States, a combination of domestic policy support for wind power and the rise of Suzlon (a leading global wind turbine manufacturer) have led India to become the country with the fourth largest installed wind power capacity in the world, and the wind energy leader in the developing world. The Indian wind energy sector has an installed capacity of 8757.2 MW (as on 31 March 2008). In terms of wind power-installed capacity, India is ranked fourth in the world. Today, India is a major player in the global wind energy market.

The potential is far from exhausted. Indian Wind Energy Association has estimated that with the current level of technology, the 'on-shore' potential for utilization of wind energy for electricity generation is of the order of 65,000 MW. The unexploited resource availability has the potential to sustain the growth of wind energy sector in India in the years to come. The following map shows the wind power density in India. As of September 2007, the installed capacity of wind power in India was 7660.2 MW, mainly spread across Tamil Nadu (3457.5 MW), Maharashtra (1484.9 MW), Karnataka (849.4 MW), Rajasthan (469.9 MW), Gujarat (667.1 MW), Andhra Pradesh (121.8 MW), Madhya Pradesh (57.8 MW), Kerala (2 MW), West Bengal (1.6 MW) and other states (1.6 MW).

The worldwide installed capacity of wind power reached 74,223 MW by the end of 2006. Germany (20,621 MW), Spain (11,615 MW) and the United States (11,603 MW) are ahead of India. The short gestation periods for installing wind turbines, and the increasing reliability and performance of wind energy machines has made wind power a favoured choice for capacity addition in India.

Suzlon, an India-owned company, emerged on the global player in the past decade, and by 2006 had captured almost 8 per cent of market share in global wind turbine sales. Suzlon is currently the leading manufacturer of wind turbines for the Indian market, holding some 50 per cent of market share in India. Suzlon's success has made India the developing country leader in advanced wind turbine technology.

1. *Tamil Nadu (3457.5 MW)*

India is keen to decrease its reliance on fossil fuels to meet its energy demand. Shown here is a wind farm in Muppandal, Tamil Nadu.

Tamil Nadu is the state with the highest wind-generating capacity: 3216 MW at the end of 2006. Not far from Aralvaimozhi, the Muppandal Wind Farm, the largest in Asia, is located near the once impoverished village of Muppandal, supplying the villagers with electricity for work. The village had been selected as the showcase for India's $2 billion clean energy programme that provides foreign companies with tax breaks for establishing fields of wind turbines in the area.

2. *Maharashtra (1484.9 MW)*

Maharashtra is second only to Tamil Nadu in terms of generating capacity. Suzlon has been heavily involved and operates what was once Asia's largest wind farm, the Vankusawade Wind Park (201 MW), near the Koyna reservoir in Satara district of Maharashtra (Figure 14.54).

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Fig. 14.54 *Wind Power Density in India*

Karnataka (849.4 MW)
Rajasthan (969.9 MW)
Gujarat (700.1 MW)
Andhra Pradesh (100.8 MW)
Madhya Pradesh (57.8 MW)
Kerala (2 MW)
Wind farm is present at Kanjikode, Palakkad
West Bengal (1.6 MW)
Others (1.6 MW)
1.6 MW is installed in other states.

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14.3.10 Wind Power by Country

The following tables 14.3 and 14.4 respectively give installed wind power capacity and annual wind power generation world wide.

Table 14.3 *Installed Wind Power Capacity (MW)*

		End 2012	New 2013	Total (end of 2013)
AFRICA & MIDDLE EAST	Ethiopia	81	90	171
	Egypt	550	–	550
	Morccco	291	–	291
	Tunisia	104	–	104
	Iran	91	–	91
	Cape Verde	24	–	24
	Other(1)	24	–	24
	Total	**1,165**	**90**	**1,255**
ASIA	PR China	75,324	16,088	91,412
	India	18,421	1,729	20,150
	Japan	2,614	50	2,661
	Taiwan	571	43	614
	South Korea	483	79	561
	Thailand	112	111	223
	Pakistan	56	50	106
	Sri Lanka	63	–	63
	Mongolia	–	50	50
	Other(2)	71	16	87
	Total	**97,715**	**18,216**	**115,927**
EUROPE	Germany	31,270	3,238	34,250
	Spain	22,784	175	22,959
	UK	8,649	1,883	10,531
	Italy	8,118	444	8,552
	France	7,623	631	8,254
	Denmark	4,162	657	4,772
	Portugal	4,529	196	4,724
	Sweden	3,746	724	4,470
	Poland	2,496	894	3,390

(*Continued*)

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Table 14.3 (*Continued*)

		End 2012	New 2013	Total (end of 2013)
	Turkey	2,312	646	2,959
	Netherlands	2,391	303	2,693
	Romania	1,905	695	2,600
	Ireland	1,749	288	2,037
	Greece	1,749	116	1,865
	Austria	1,378	308	1,684
	Rest of Europe[3]	4,956	832	5,737
	Total Europe	**109,817**	**12,031**	**121,474**
	of which EU-28[4]	106,454	11,159	117,289
LATIN AMERICA & CARIBBEAN	*Brazil	2,508	953	3,461
	Chile	205	130	335
	Argentina	142	76	218
	Costa Rica	148	–	148
	Nicaragua	146	–	146
	Honduras	102	–	102
	Dominican Republic	33	52	85
	Uruguay	56	4	59
	Carribean[5]	136	–	136
	Others[6]	54	20	74
	Total	**3,530**	**1,235**	**4,764**
NORTH AMERICA	USA	60,007	1,084	61,091
	Canada	6,204	1,599	7,803
	Mexico	1,537	380	1,917
	Total	**67,748**	**3,063**	**70,811**
PACIFIC REGION	Australia	2,584	655	3,239
	New Zealand	623	–	623
	Pacific Islands	12	–	12
	Total	**3,219**	**655**	**3,874**
	World total	**283,194**	**35,289**	**318,105**

Source: GWIC

[1] Israel, Jordan, Kenya, Libya, Nigeria, South Africa.

[2] Bangladesh, Philippines, Vietnam.

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[3] Bulgaria, Cyprus, Czech Republic, Estonia, Finland, Faroe Islands, FYROM, Hungary, Iceland, Latvia, Liechtenstein, Lithuania, Luxembourg, Malta, Norway, Romania, Russia, Switzerland, Slovakia, Slovenia, Ukraine.
[4] Austria, Belgium, Bulgaria, Cyprus, Croatia, Czech Republic, Denmark, Estonia, Finland, France, Germany, Greece, Hungary, Ireland, Italy, Latvia, Lithuania, Luxembourg Malta, Netherlands, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, UK.
[5] Caribbean: Aruba, Bonaire, Curacao, Cuba, Dominica, Guadalupe, Jamaica, Martinique, Granada.
[6] Bolivia, Colombia Ecuador, Peru, Venezuela.

Table 14.4 *Annual Wind Power Generation (TWh)/Total Electricity Consumption (TWh)*

Rank	Nation	2005	2006	2007
1.	Germany	27.225/533.700	30.700/569.943	39.500/584.939
2.	United States	/4049.8	26.3[35]/4104.967	/4179.908
3.	Spain	23.166/254.90	29.777/294.596	/303.758
4.	India	/661.64		
5.	China	/2474.7	2.70/2834.4	/3255.9
6.	Denmark (& Faeroe Islands)	6.614/34.30	7.432 /44.24	/37.276
7.	France	/547.8	2.323/550.063	/545.289
8.	United Kingdom	0.973/407.365	/383.898	/379.756
9.	Portugal	/35.0	4.74/48.876	
		World total (TWh)	'[36]	/16790

14.4 TIDAL ENERGY

The tides are one of the sources of energy from the oceans. This energy can be tapped from coastal waters by building dams. Dams entrap the water at high tide and release it at low tide back to the sea. Power can then be generated by turbines from both in–out flows of the water. The amount of energy available is very large but only in a few parts of the world.

The tides are rhythmic but not constant. They do not occur on a regular daily schedule. Their occurrence is due to the a balance of forces, mainly the gravitational force of the moon but also that of the sun, both acting together with that of the earth to balance the centrifugal force on the water to the earth's rotation. The result is the rhythmic rise and fall of water. The tides are characterized by their schedule and range *R*.

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The moon rotates around the earth every 24 h, 50 min. During this time, the tide rises and falls twice, resulting in a tidal cycle that lasts 12 h, 25 min.

$$R = \text{water elevation at high tide} - \text{water elevation at low tide}$$

The range is not constant. It varies during the 29.5 day lunar month, being maximum at the time of new and full moons, called the spring tides, and minimum of the time of the first and third quarter moons, called neap tides. The spring neap tidal cycle lasts one-half of a lunar month. A typical mean range is roughly one-third of the spring range.

The variations in daily periodicity and monthly and seasonal ranges must be taken into account in the design and operation of tidal power plants. The tides, however, are usually predictable, and fairly accurate tide tables are usually available.

The total tidal power that is dissipated throughout the world is estimated at 2.4×10^6 MW. Of these some 10^6 MW are dissipated in shallow seas and coastal areas and are not recoverable. Because of the very high capital costs of dams and other structures associated with tide energy conversion systems, only a small fraction of the rest can be satisfied by tidal energy (Figure 14.55).

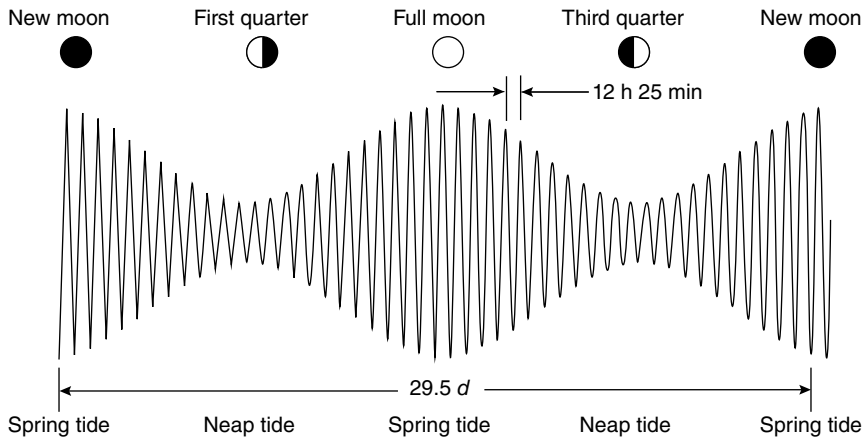


Fig. 14.55 *Relative High and Low Tides Showing Variations in Range During Lunar Month*

14.4.1 The Simple Single Pool Tidal System

The simple single pool tidal system has one pool or basin behind a dam that is filled from the ocean at high tide and emptied to it at low tide. Both filling and emptying processes take place during short periods of time. The filling takes place when the ocean is at high tide, while the water in the pool is at low tide; the emptying takes place when the ocean is at low tide and the pool at high tide. The flow of water in both directions is used to drive various reversible water turbines, each driving an electrical generator. Electric power would thus be generated during two short periods during each tidal period of 12 h, 25 min or once every 6 h, 12.5 min (Figure 14.56).

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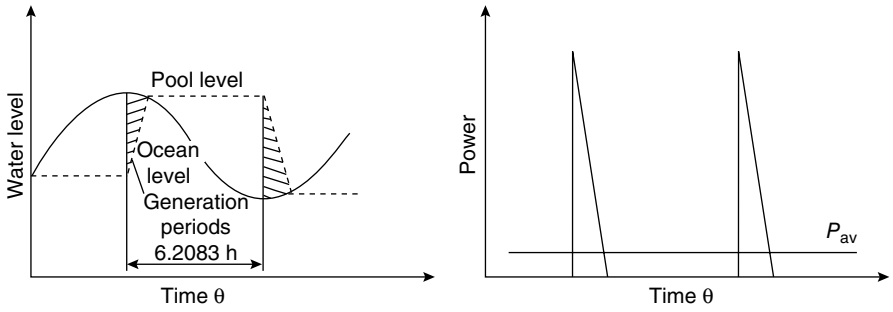


Fig. 14.56 *Ocean and Pool Levels and Power Generated in a Simple Single Basis System*

The maximum energy that can be generated during one generation period can be determined as shown in Figure 14.57, which shows the case of the pool beginning at high tide, emptying through the turbine to the ocean, which is at low tide.

For a tidal range, R, and an intermediate head, h, at a given time during the emptying process, the differential work done by the water is equal to its potential energy at the time, or

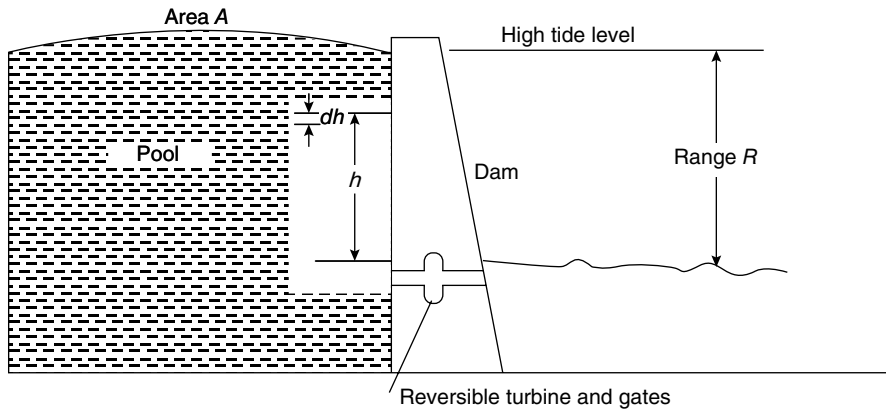


Fig. 14.57 *Tidal Energy Generation During One Generation Period*

the differential work done by the water is equal to its potential energy at the time, or

$$\text{we have, } dW = \frac{g}{g_c} dm \times h$$

$$\text{But } dm = -\rho A \times dh$$

$$\therefore \quad dW = \frac{g}{g_c} \rho A h \times dh \qquad (1)$$

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where

W = work done by the water, J
g = gravitational acceleration, 32.2 ft/s^2 or 9.81 m/s^2
g_c = conversion factor 1.0 kg/(N – s^2)
m = mass flowing through turbine, kg
h = head, m
ρ = water density, kg/m^3
A = surface area of pool, considered constant, m^2

The total theoretical work during a full emptying (or filling) period is obtained by integrating Equation (1)

$$W = \int_R^0 dW = -\frac{g}{g_c}\rho A \int_R^0 h dh$$

$$W = \frac{1}{2} \times \frac{g}{g_c} \times \rho A R^2$$

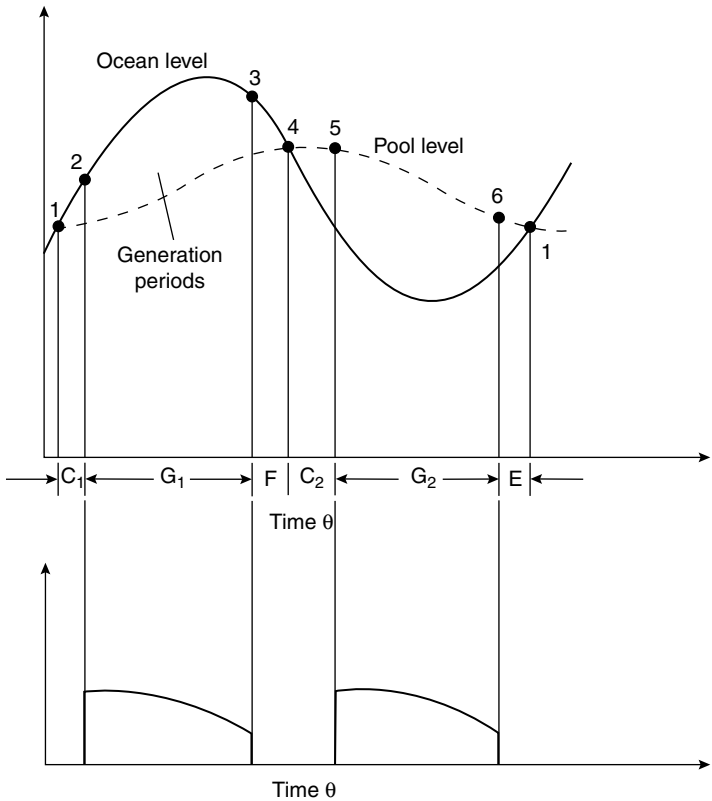


Fig. 14.58 *Ocean and Pool Levels in a Modulated Single-Pool Tidal System*

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14.4.2 The Modulated Single Pool Tidal System

In the simple single pool system two high-peak, short-duration power outputs occur during every tidal period. Such peaks require large turbine generators that remain idle much of the time. The power peaks also occur at different times every day (50 min later each successive day), at times of high and low tides. This will not always correspond to times of peak power demand and pose a burden on the electric power grid they are connected to.

The modulated single-pool tidal system partially corrects for these deficiencies by generating power more uniformly at a lower average head, although still with some periods of no generation. Because the average head h is lower and work and power are proportional to h^2, the turbine generators are much smaller and operate over much longer periods. The resulting total work is reduced.

In this system, shown in Figure 14.58, the reversible turbines are allowed to operate during pool filling and emptying instead of at high and low levels only. They do not operate when the head is too low for efficient operation. Period C_1 begins with both pool and ocean at low tide level (i), the ocean at the beginning of tide rise, and all gates closed. When the head is sufficient (ii), gates to the turbines are opened and water from the ocean is allowed through. Power is generated during period G_1 as both ocean and pool levels rise. The ocean level reaches its peak and begins to decrease but the pool level still increases until at 3 the head is too low for efficient generation. The gates to the turbines are closed and bypass gates are opened so that the pool is allowed to till up during period F to 4. At 4, all gates are once again closed and the pool level remains constant while the ocean level decreases during period C_2. At 5, the head is once again sufficient to allow for turbine water flow in the opposite direction and a second-generation period G_2 begins. At 6, generation ceases but the pool is allowed to empty during period E and the system returns back to point 1, repeating the cycle.

14.4.3 The Two-Pool Tidal System

Figure 14.59 shows a two pool tidal system. The two-pool tidal system is less dependent on tidal fluctuation but is more complex and involves more costly dam construction. An inland basin is enclosed by dam A and is divided into a high pool and a low pool by dam B. By proper gating in dam A, the high pool gets periodically filled at high tide from the ocean and the low pool gets periodically emptied at low tide. Water flows from the high pool to the low pool through the turbines in dam B. The capacities of these two pools are large enough in relation to the water flow between them that the fluctuations in the head are minimized resulting in continuous and more uniform power generation.

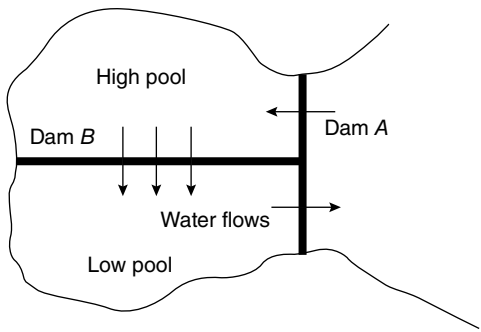


Fig. 14.59 *Schematic of a Two-Pool Tidal System*

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14.5 OCEAN THERMAL ENERGY CONVERSION

14.5.1 Principle of Working

Solar energy, which may be used directly, creates other forms of energy that can also be harnessed to generate power. Absorption of solar radiation by the seas and oceans causes ocean currents and moderate temperature gradients from the water surface downward especially in tropical waters. The ocean and seas constitute some 70 per cent of the earth's surface area representing a rather large storage reservoir of the solar input.

The temperature gradient can be utilized in a heat engine to generate power. This is called ocean temperature energy conversion (OTEC).

14.5.2 Ocean Temperature Differences

Considering deep waters in general, the high temperatures are at the surface, whereas deep water remains cool. In the tropics, the ocean surface temperature often exceeds 25°C (77°F), whereas 1 km below the temperature is usually no higher than 10°C (50°F). Water density decreases with an increase in temperature. Thus, there will be no thermal convection currents between the warmer, lighter water at the top and the deep cooler, heavier water. Thermal conduction heat transfer between them, across the large depths, is too low and thus mixing is retarded, so the warm water stays at the top and the cool water stays at the bottom. Hence, in tropical waters, there are two essentially infinite heat reservoirs, a heat source at the surface at about 27°C (81°F) and a heat sink some 1 km directly below, at about 4°C (39°F); both reservoirs are maintained annually by solar incidence.

The concept of OTEC is based on the utilization of this temperature difference in a heat engine to generate power (first recognized by the Frenchman d'Arsonval in 1881). The maximum temperature difference on earth is in the tropics and is about 15°C (59°F). Ocean currents carry the 27–28°C warm tropical surface waters on a journey to the arctic circles during which they are gradually cooled to 4°C and maximum density (Figure 14.60). In the arctic circle, they then settle below the surface and a surface deep water siphon is created that keeps cold water below the surface.

The maximum possible efficiency of a heat engine operating between two temperature limits cannot exceed that of a Carnot cycle operating between the same temperature limits. For heat source and sink temperatures, T_1 and T_2, the Carnot efficiency η_c is given by

$$\eta_c = \frac{T_1 - T_2}{T_1}$$

It is thus important to have large temperature difference $T_1 - T_2$. In OTEC systems, this may average 20°C (68°F) compared with about 500°C (932°F) for

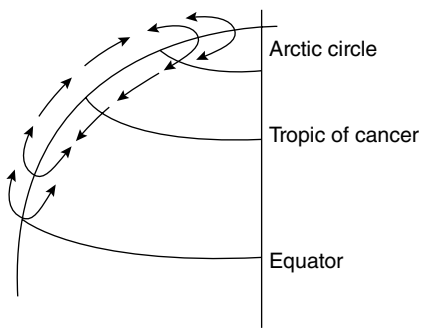


Fig. 14.60 *Ocean Surface Temperature as a Function of (a) Latitude and (b) The Seasons, in Tropical Waters*

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modern fossil power plants. Taking the temperature difference of 20°C and a surface temperature of 27°C, the Carnot efficiency would be

$$\eta_c = 0.0667 = 6.67\%$$

Because of temperature drops in steam or other vapour generator and condenser in an actual system, inefficiencies in turbine and pumps pressure drops of the large volumes of water and working fluid flows (internal irreversibility), and inefficiencies of other components the efficiency of a real OTEC power plant seldom exceeds 2 per cent (Figure 14.61).

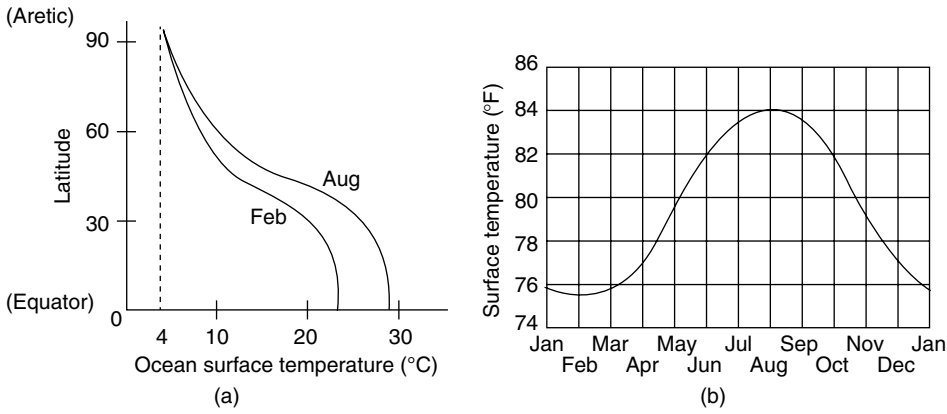


Fig. 14.61 *Variation of Ocean Surface Temperature with Latitude and Months of a Year*

The extremely low efficiency of an OTEC system implies extremely large power e plant heat exchangers and components. At 2 per cent efficiency, the heat exchangers must handle 50 times the net output of the plant. In addition to the large size per unit power generation, the developmental problems and the uncertainties of market penetration make the financial risks associated with the development of large OTEC technologies so high as to effectively preclude most utilities.

There are two basic types for OTEC systems: the open cycle also known as the Claude cycle and the closed cycle also known as the Anderson cycle.

14.5.3 The Open or Claude Cycle

An open cycle also called the Claude cycle in which seawater itself plays the multiple role of heat source, working fluid and heat sink.

In the cycle shown in Figure 14.62, warm surface water at 27°C is admitted into an evaporator in which the pressure is maintained at a value slightly below the saturation pressure corresponding to that of water temperature. Water entering the evaporator gets itself 'superheated' at the new pressure. For example, in Figure 14.63, the warm water at 27°C has a saturation pressure of 0.0356 bar at point 1. The evaporator pressure is 0.0317 bar, which corresponds to 25°C

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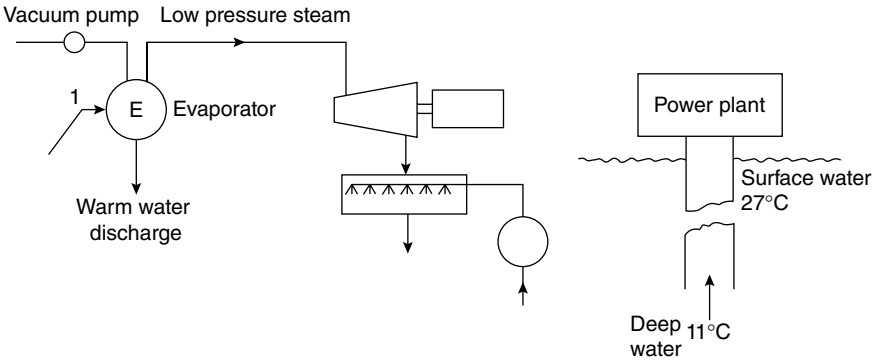


Fig. 14.62 *A Claude (Open Cycle) OTEC Power Plant*

saturation temperature. This temporarily superheated water undergoes volume boiling causing water to partially flash to steam to an equilibrium two-phase condition at the new pressure and temperature of 0.0317 bar and 25°C point 2. Process 1–2 is a throttling and hence constant enthalpy process. The low pressure in the evaporator is maintained by a vacuum pump that also removes the dissolved non-condensable gases from the evaporator.

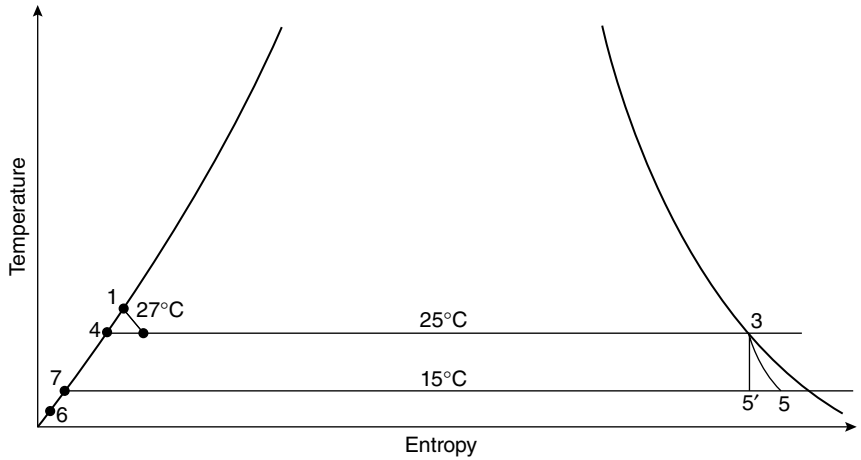


Fig. 14.63 *T–S Diagram of a Claude (Open Cycle) Cycle*

The evaporator contains a mixture of water and steam of very low quality at 2. The steam is separated from the water as saturated vapour at 3. The remaining water is saturated at 4 and is discharged back to the ocean. The steam at 3 is a very low pressure (0.0317 bar) and very high specific volume working fluid (43.40 m^3/kg) expands in a specially designed turbine to 5. In the figure, the condenser pressure and temperature at 5 are 0.017 bar and 15°C. Since the turbine exhaust system will be discharged back to the ocean in the open cycle, a direct contact condenser is used in which the exhaust at 5 is mixed with cold water from the deep cold water pipe at 6, which results in near saturated water at 7. That water is discharged to the ocean.

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The cooling water reaching the condenser at 13°C is obtained from deep water at 11°C. This rise in temperature is caused by heat transfer between the progressively warmer outside water and the cooling water inside the pipe as it ascends the cold water pipe.

14.5.4 The Closed or Anderson OTEC Cycle

A closed cycle utilizes the ocean's warm surface and cool deep waters as heat source and sink, respectively, but requires a separate working fluid that receives and rejects heat to the source and sinks via heat exchangers (boiler and surface condenser) as shown in Figure 14.64.

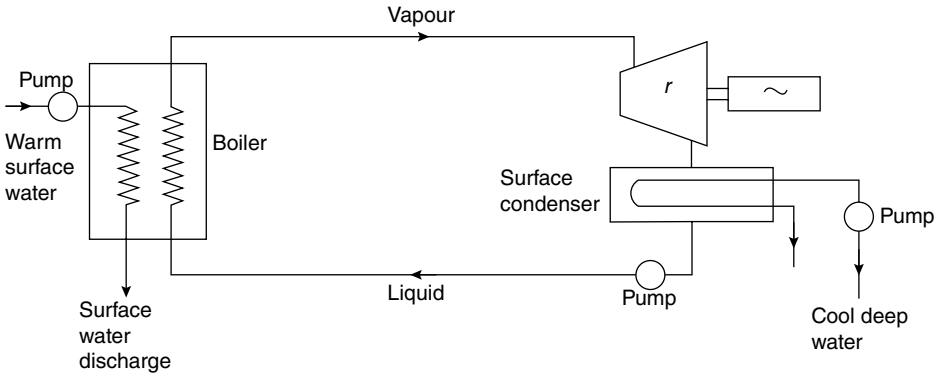


Fig. 14.64 *Schematic of a Closed Cycle OTEC Plant*

The working fluid may be ammonia, propane or Freon. The operating (saturation) pressures of such fluids at the boiler and condenser temperatures are much higher than those of water, being roughly 10 bar at the boiler, and their specific volumes are much lower, being comparable to those of steam in conventional power plants.

Such pressures and specific volumes result in turbines that are much smaller and hence less costly than those that use the low-pressure steam of the open cycle. The closed cycle also avoids the problems of the evaporator. But it requires the use of very large heat exchangers (boiler and condenser) because, for an efficiency of about 2 per cent, the amounts of heat added and rejected are 50 times the output of the plant. In addition, the temperature differences in the boiler and condenser must be kept as low as possible to allow for the maximum possible temperature difference across the turbine, which also contributes to the large surfaces of these units.

The closed cycle approach was first proposed by Barjot in 1926, but the most recent design was by Anderson and Anderson in the 1960s. The closed cycle is sometimes referred to as the Anderson cycle. The Anderson chose propane as the working fluid with a 20°C temperature difference between warm surface and cool water, the latter some 600 m deep. Propane is vapourized in the boiler at 10 bar or more and exhausted in the condenser at about 5 bar.

Anderson OTEC system employs thin plate-type heat exchangers instead of the usual heavier and more expensive shell-and-tube heat exchanges. To help reduce the thickness of the

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plates, the heat exchangers are placed at depths where the static pressure of the water in either exchanger roughly equals the pressure of the working fluid.

In the international arena, many countries have recently been active in OTEC research and development. These include Japan, France, Sweden, West Germany and the Netherlands. A 100 kW land-based plant in the Pacific Ocean equatorial island nation of Nauru is being built by the Japanese.

14.6 WAVE ENERGY

Like the wind and OTEC, ocean and sea waves are caused indirectly by solar energy. Waves are caused by the wind, which in turn is caused by the uneven solar heating and subsequent cooling of the earth's crust and the rotation of the earth. Wave energy at its peak can be much more concentrated than incident solar energy even at the latter's peak. Devices that convert energy from waves can therefore produce much higher power densities than solar devices.

The main advantages of power from waves

(i) It is a free and renewable energy source. Wave power devices unlike solar or wind devices, do not use up large land masses are relatively pollution free.

(ii) Because they remove energy from the waves, they leave the water in a relatively placid state in their wakes. Thus, a string of devices are situated where there is a large amount of wave activity can in addition to producing electricity protect coastlines from the destructive action of such waves, minimize erosion and even help create artificial harbours.

Disadvantages

(i) Waves lack dependability and there is relative scarcity of accessible sites of large wave activity.

(ii) Most devices that have been proposed are relatively complicated and lack the necessary mechanical strength to withstand the enormous power of stormy seas.

(iii) Economic factors, such as the capital investment, costs of maintenance, repair and replacement as well as problems of biological growth of marine organisms, are all relatively higher.

14.6.1 Energy and Power from Waves

The total energy of a wave is the sum of its potential and kinetic energies (Figure 14.65).

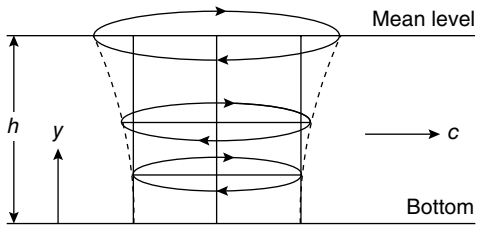


Fig. 14.65 *Elliptical Paths of Water Particles at Different*

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1. *Potential energy*

Potential energy arises from the elevation of the water above the mean level ($y = 0$) (Figure 14.66).

The potential energy density per unit area is PE/A, where $A = \lambda L$, in J/m^2, is then given by

$$\frac{PE}{A} = \frac{1}{4}\rho a^2 \frac{g}{g_c}$$

where g = gravitational acceleration, m/s^2, g_c = conversion factor, 1.0 kg m/(N-s^2), ρ = water density, kg/m^3 and a = amplitude, m.

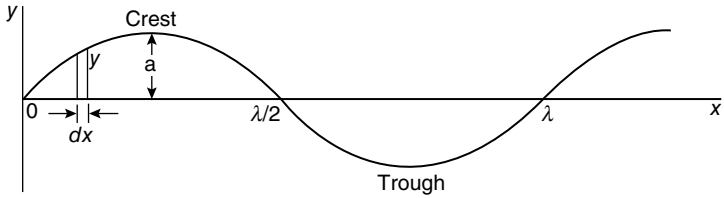


Fig. 14.66 *Energy and Power from Waves*

2. *Kinetic energy*

The kinetic energy of the wave is that of the liquid between two vertical planes perpendicular to the direction of wave propagation x and placed one wavelength apart.

The kinetic energy density is given by

$$\frac{KE}{A} = \frac{1}{4}\rho a^2 \frac{g}{g_c}$$

3. *Total energy and power*

It can be seen that the potential and kinetic energies of a progressive sine wave are identical, so that the total energy E is half potential and half kinetic. The total energy density is thus given by

$$\frac{E}{A} = \frac{1}{2}\rho a^2 \frac{g}{g_c}$$

The power P per unit time is given for a wave by energy times frequency. Thus, the power density, W/m^2, is given by

$$\frac{P}{A} = \frac{1}{2}\rho a^2 f \frac{g}{g_c}$$

where $f = \dfrac{1}{t}$ = wave frequency

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14.6.2 Wave Energy Conversion by Floats

Wave motion is primarily horizontal, but the motion of water is primarily vertical. This latter motion is made use of by floats to obtain mechanical power. A large float is driven up and down by the water within relatively stationary guides. This reciprocating motion is converted to mechanical and then electric power. Figure 14.67 shows a wave energy conversion system using floats.

The square float moves up and down with the water, guided by four vertical manifolds that are part of a platform. The platform is stabilized within the water by four large underwater floatation tanks so that it is supported by buoyancy forces. Due to this, no significant vertical or horizontal displacement of the platform due to wave action occurs. Damping fins may be used to further reduce motion if necessary. The platform is therefore relatively stationary in space even in heavy seas. The platform can be made of moulded plastic with a foamed plastic core to arrive at the required density and strength.

A piston attached to the float moves up and down inside a cylinder attached to the platform and is therefore relatively stationary. This piston cylinder arrangement is used as a reciprocating air compressor. The downward motion of the piston draws air into the cylinder via an inlet check valve. The upward motion compresses the air and sends it through an outlet check valve to the four underwater floatation tanks via the four manifolds. The four floatation tanks thus serve the dual purpose of buoyancy and air storage and the four vertical manifolds serve the dual purpose of manifolds and float guides (Figure 14.67).

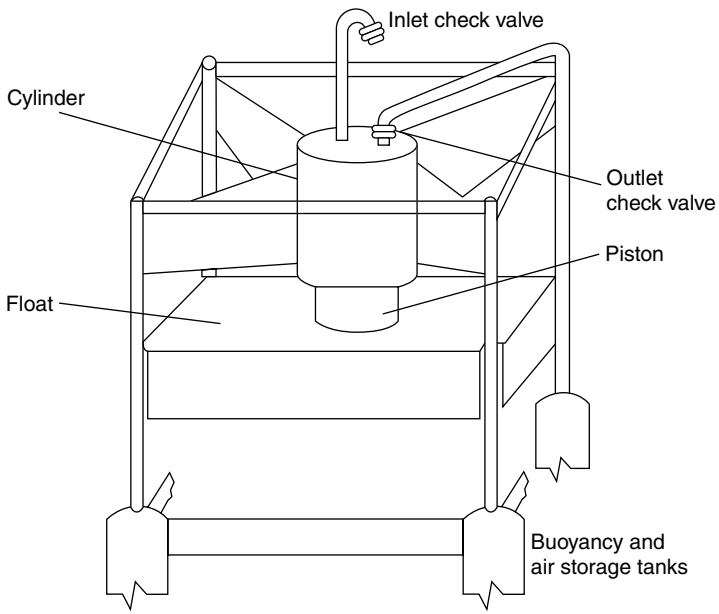


Fig. 14.67 *Wave Energy Conversion by Floats*

The compressed air in the buoyancy storage tanks is in turn used to drive an air turbine that drives an electrical generator. The electric current is transmitted to the shore via an underwater cable.

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14.6.3 High-Pressure Accumulator Wave Machines

In these machines, instead of compressing air the water itself is pressurized and stored in a high-pressure accumulator or pumped to a high-level reservoir from which it flows through a water turbine electrical generator. This is done by transforming large volumes of low-pressure water at wave crest into small volumes of high-pressure water by the use of a composite piston. This piston is composed of a large diameter main piston and a small diameter piston at its centre.

Figure 14.68 shows a hydraulic accumulator in which the main large piston moves inside a submerged cylindrical generator while the small piston moves inside a power cylinder. Wave water enters and leaves through openings at the bottom hence causing the main piston to move up and down. A closed water loop exists above the small piston. On the upstroke, the pressure on the main piston is magnified on the small piston by the inverse ratio of the square of their diameters.

The high-pressure water is passed through a one-way up valve to a hydraulic accumulator at the top of the generator. Two air (or other gas) volumes in the accumulator counterbalance and act as cushions in a chamber above the main piston and in a sealed compartment of the hydraulic accumulator. The latter also maintains high water pressure. Part of the high-pressure water flows through a Pelton wheel or Francis hydraulic turbine that drives an electrical generator and is then discharged to a storage chamber below the turbine.

On the trough of the wave, the composite piston is pushed downward by the gas pressure above the main piston, which thus acts also as a spring. The turbine's exhaust water in the storage volume is sucked into the pump cylinder via a one-way return valve while the up valve is closed and the cycle is repeated. The hydraulic accumulator is large enough to permit continuous turbine operation even though the waves are cycle.

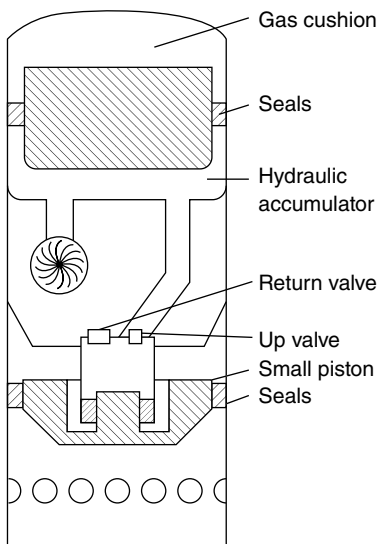


Fig. 14.68 *High-Pressure Accumulator Wave Machines*

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Figure 14.69 shows a high-level reservoir wave machine in which similar pressure magnification piston is used. The pressurized water is elevated to a natural reservoir above the wave generator near a shoreline or to an artificial water reservoir. The water in the reservoir is made to flow through a turbine back to sea level. A 20 m diameter generator of this type can produce 1 MW.

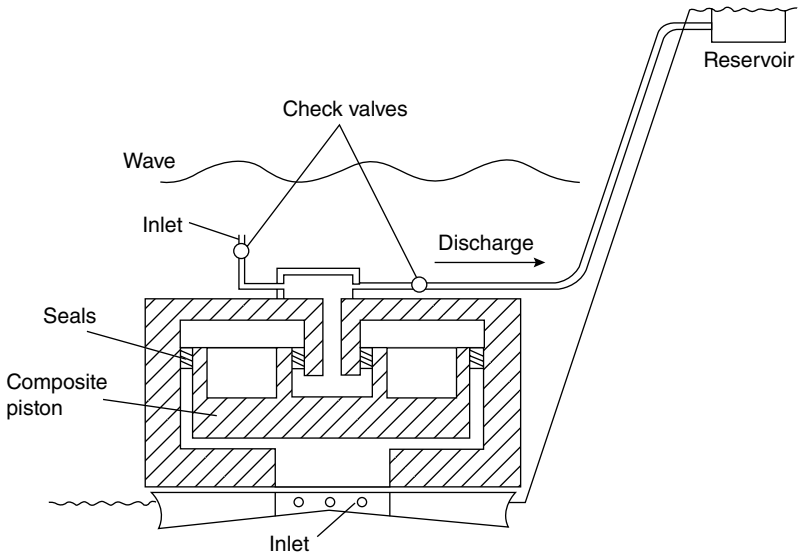


Fig. 14.69 *High-Level Reservoir Wave Machine*

14.7 FUEL CELLS

14.7.1 Working Principle

As discussed early in the chapter, a fuel cell directly converts a hydrogen-rich fuel into electricity using a highly efficient electrochemical process. In a fuel cell, the reverse electrolysis process takes place to produce water and electricity from hydrogen and oxygen. The hydrogen is normally produced from natural gas or methanol. Chemical energy is stored outside the fuel cell, and the reaction products are commonly rejected from the cells.

Figure 14.70 schematically shows a fuel cell. The reverse electrolysis process is performed by controlled continuous chemical reactions at the anode and cathode of the fuel cell. In a typical hydrogen–oxygen fuel cell, hydrogen is the fuel and oxygen is the oxidant. The electrodes are porous and are connected to the load. At the anode, the hydrogen fuel splits into positive hydrogen ions and electrons according to the following reaction:

$$H_2 \rightarrow 2H^+ + 2e^-$$

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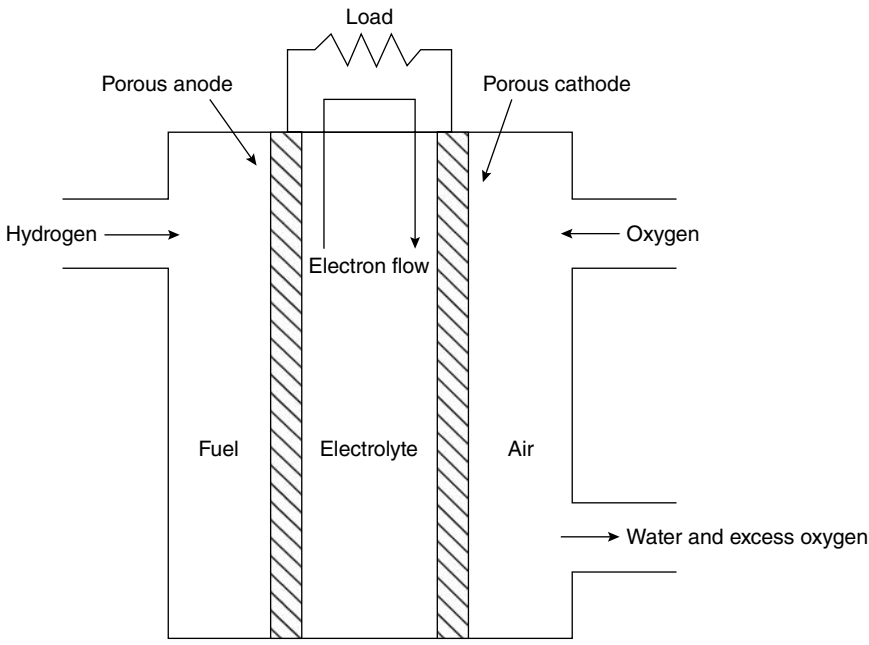


Fig. 14.70 *Fuel Cell Working Principle*

The electrons and the hydrogen ions combine with oxygen at the cathode after flowing through the load circuit, according to the following reaction:

$$2H^+ + 2e^- + \frac{1}{2}O_2 \rightarrow H_2O$$

The overall reaction in the fuel cell is given by

$$2H_2 + O_2 \rightarrow H_2O$$

The oxidation product is water, which is removed from the cell.

The ideal electromotive force (emf) E of a fuel cell is given by

$$E = -\frac{\Delta G}{23.06n} \text{ expressed in volts}$$

where ΔG = the energy change is theoretically available from the fuel cell is ΔG

= the Gibbs free energy change, in kilocalories per gram mole

N = the number of electrons transferred per molecule of fuel oxidized

If ΔH = the net amount of energy liberated is the change in enthalpy

= the difference between the enthalpy of the reactants and the enthalpy of the products.

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Then, the maximum possible emf for a fuel cell is given by

$$\eta = \frac{\Delta G}{\Delta H}$$

Example 14.8

For the standard fuel cell reaction, the value for the Gibbs free energy is $\Delta G = -56.7$ kcal at 25°C and the value for the enthalpy is $\Delta H = -68.3$ kcal. Determine the maximum possible emf for a fuel cell.

Solution: We have the maximum possible emf for a fuel cell

$$\eta = \frac{\Delta G}{\Delta H} = \frac{-54.7}{-68.1} = 0.803$$

$$E = -\frac{\Delta G}{23.06n} = -\frac{54.7}{23.06 \times 2} = 1.19 \text{ V.}$$

In fuel cell power plants, a number of fuel cells are used. A fuel-processing section in the plant converts natural gas or other hydrocarbon fuels into a hydrogen-rich fuel through a steam catalytic reforming process. This fuel is then fed to the power section. Here, it reacts with oxygen in fuel cells to produce DC electricity in addition to by-products such as hot water or steam. Individual fuel cells are connected in series in a fuel cell 'stack' to generate usable amounts of electricity. The power conditioning section converts DC to utility grade AC.

14.7.2 Types of Fuel Cells

Fuel cells are classified based on the type of electrolyte used, and by their operating temperature. Low-temperature fuel cells operate in the range of 60–200°C and include phosphoric acid fuel cells, alkaline fuel cells and solid polymer fuel cells. High-temperature fuel cells operate in the range of 650–1000°C and include molten carbonate fuel cells and solid oxide fuel cells. In general, fuel cells may be of the following types:

(i) Polymer Electrolyte Membrane (PEM) Fuel Cells
(ii) Direct Methanol Fuel Cells
(iii) Alkaline Fuel Cells (AFCs)
(iv) Phosphoric Acid Fuel Cells (PAFCs)
(v) Molten Carbonate Fuel Cells (MCFCs)
(vi) Solid Oxide Fuel Cells (SOFCs)

1. *PEM fuel cells*

PEM fuel cells are also called as proton exchange membrane fuel cells. They deliver high-power density coupled with low weight and volume. PEM fuel cells use a solid polymer as an electrolyte

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and porous carbon electrodes containing a platinum catalyst. They take only hydrogen, oxygen from the air and water to operate and do not require any corrosive fluids like in fuel cells. The fuelling is done with pure hydrogen supplied from storage tanks or on-board reformers.

These fuel cells operate at relatively low temperatures, around 80°C, which enables quick start (less warm-up time) and results in less wear on system components, resulting in better durability. However, it is costly as it requires a noble-metal catalyst (typically platinum) to separate the hydrogen's electrons and protons, adding to system cost.

In addition, the platinum catalyst is also extremely sensitive to CO poisoning and requires an additional reactor to reduce CO in the fuel gas if the hydrogen is derived from an alcohol or hydrocarbon fuel.

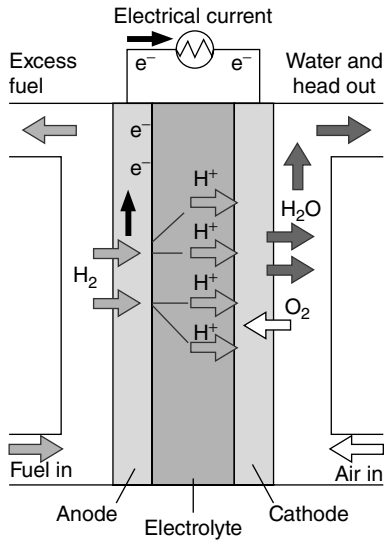


Fig. 14.71 *Polymer Electrolyte Membrane (PEM) Fuel Cells*

PEM fuel cells are used primarily for transportation applications and some stationary applications. As these fuel cells offer fast start-up time, low sensitivity to orientation and favourable power-to-weight ratio, they are particularly suitable for use in passenger vehicles, such as cars and buses (Figure 14.71).

2. *Direct methanol fuel cells*

Unlike most fuel cells that are powered by hydrogen, direct methanol fuel cells use methanol that has a higher energy density than hydrogen and has no storage issues. Methanol is also easier to transport and supply, such as gasoline. Direct methanol fuel cell technology is relatively new compared with that of fuel cells powered by pure hydrogen.

3. *AFCs*

AFCs were one of the first fuel cell technologies developed (Figure 14.72). These fuel cells use a solution of potassium hydroxide in water as the electrolyte and can use a variety of non-precious metals as a catalyst at the anode and cathode. High-temperature AFCs operate at temperatures between 100 and 250°C, whereas some newer AFC designs operate at lower temperatures of roughly 23–70°C.

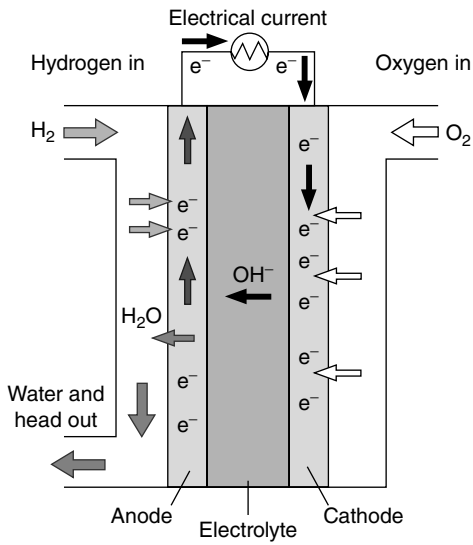


Fig. 14.72 *Alkaline Fuel Cell (AFC)*

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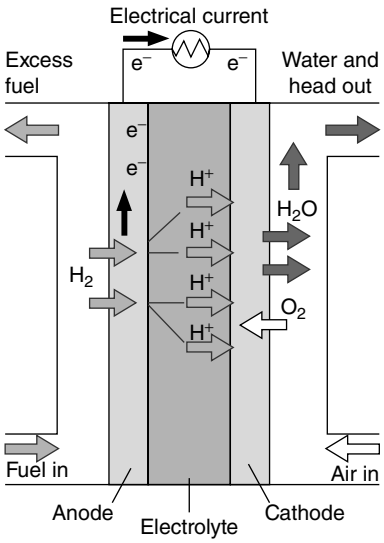


Fig. 14.73 *Phosphoric Acid Fuel Cells*

AFCs have high performance due to the rapid rates of chemical reactions that take place in the cell. They also have efficiencies near 60 per cent in space applications. However, they are easily poisoned by carbon dioxide (CO_2). They are costly and these fuel cells will have to become more cost effective.

4. *PAFCs*

These fuel cells use liquid phosphoric acid as an electrolyte. The acid is contained in a Teflon-bonded silicon carbide matrix and porous carbon electrodes containing a platinum catalyst. Figure 14.73 shows a typical PAFC.

Advantages

(i) PAFCs have more capability to absorb impurities in fossil fuels compared to other types, which are easily 'poisoned' by carbon monoxide because carbon monoxide binds to the platinum catalyst at the anode, decreasing the fuel cell's efficiency.

(ii) They are 85 per cent efficient when used for the co-generation of electricity and heat.

Disadvantages

(i) Less efficient at generating electricity alone (37–42 per cent).

(ii) Less powerful than other fuel cells, for the same weight and volume.

(iii) They are typically larger and heavier.

(iv) PAFCs are expensive as they require an expensive platinum catalyst.

5. *MCFCs*

MCFCs are used for natural gas and coal-based power plants for electrical utility, industrial and military applications (Figure 14.74). They are high-temperature fuel cells that use an electrolyte composed of a molten carbonate salt mixture suspended in a porous, chemically inert ceramic lithium aluminium oxide ($LiAlO_2$) matrix. Due to extremely high operating temperatures of 650°C and above, non-precious metals can be used as catalysts at the anode and cathode. This reduces the fuel costs.

Advantages

(i) Higher efficiency and less costly compared to PAFCs.

(ii) Molten carbonate fuel cells, when coupled with a turbine, can have efficiencies approaching 65 per cent. With the waste heat capturing, overall fuel efficiencies can be as high as 85 per cent.

(iii) MCFCs do not require an external reformer to convert more energy-dense fuels to hydrogen unlike alkaline, phosphoric acid and PEM fuel cells. Due to the high temperatures at which MCFCs operate, these fuels are converted to hydrogen within the fuel cell itself by a process called internal reforming, which also reduces cost.

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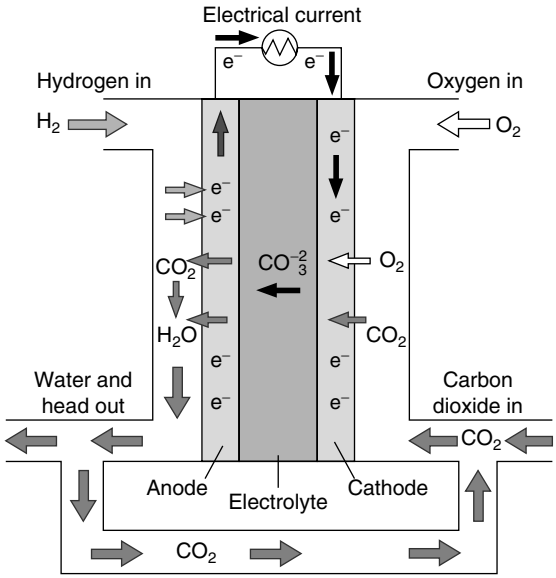


Fig. 14.74 *Molten Carbonate Fuel Cells (MCFCs)*

(iv) They are not prone to carbon monoxide or carbon dioxide 'poisoning'.

(v) They can even use carbon oxides as fuel enabling fuelling with gases made from coal.

Disadvantages

(i) MCFCs have shorter durability.

(ii) Due to the higher operating temperatures, they are corrosive electrolyte used accelerate component breakdown and corrosion, decreasing cell life.

6. *SOFCs*

SOFCs make use of a hard, non-porous ceramic compound as the electrolyte (Figure 14.75). Because of the solid electrolyte, the cells need not be constructed in plate-like form like other fuel cell types.

Advantages

(i) They have 50–60 per cent conversion efficiency. In applications with waste heat recovery arrangement, this could be up to 80–85 per cent.

(ii) Solid oxide fuel cells operate at very high temperatures (1000°C), enabling the usage of low-cost metal catalyst, thereby reducing cost.

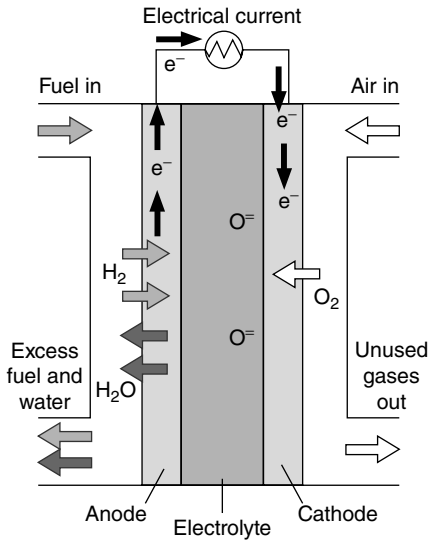


Fig. 14.75 *Solid Oxide Fuel Cells (SOFCs)*

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(iii) A variety of fuels can be used that reduce the cost associated with adding a reformer to the system.

(iv) Due to higher resistance to sulphur, they can tolerate several orders of magnitude more of sulphur than other cell types.

(v) These cells are not poisoned by carbon monoxide (CO) and can even be used as fuel.

(vi) They allow the usage of other gases made from coal.

Disadvantages

(i) It results in a slow start-up and requires significant thermal shielding to retain heat and protect personnel. Except for utility applications, it is not convenient for transportation and small portable applications.

(ii) Due to the high operating temperatures durability of materials is lower.

7. *Regenerative Fuel Cells*

Regenerative fuel cells produce electricity from hydrogen and oxygen and generate heat and water as by-products, just like other fuel cells. In addition, regenerative fuel cell systems can also use electricity from solar power or some other source to divide the excess water into oxygen and hydrogen fuel by electrolysis.

14.8 THERMOELECTRIC AND THERMIONIC POWER

14.8.1 Thermoelectric Power

It is a device that converts heat directly into electric power. It eliminates the conversion of heat into kinetic energy of gas or steam flow. Its principle is based on Seebeck effect: If two dissimilar materials are joined to form a loop and the two junctions are maintained at different temperatures, an emf will be developed around the loop. The magnitude of the emf (E) developed by the above process is proportional to the temperature difference between the two junctions.

$$E \alpha (T_2 - T_1) = \alpha(T_h - T_c)$$

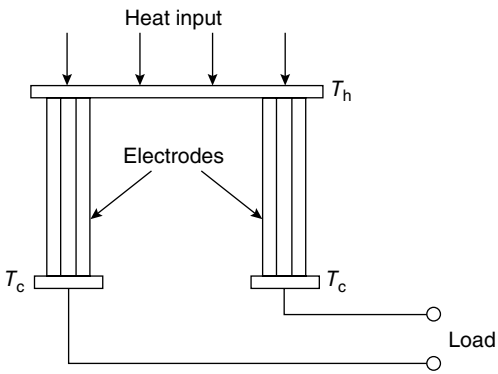


Fig. 14.76 *Thermoelectric Generator*

where T_h = temperature of hot junction and T_c = temperature of cold junction and α = Seebeck coefficient.

The hot junction is maintained at a temperature T_h by the applied heat source, which may be small oil or gas burner, a nuclear reactor or direct solar radiation by paraboloidal concentrator, and the cold junction is maintained at T_c by either water cooling or radiative heat transfer (Figure 14.76).

A thermoelectric module used for power generation is similar to a conventional thermocouple in certain aspects. Consider a single

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thermoelectric couple with an applied temperature difference as shown in Figure 14.77.

With no load (R_L not connected), the open circuit voltage as measured between points a and b is given by

$$V = \alpha \times \Delta T$$

where V = output voltage from the couple (generator) in volts, α = average Seebeck coefficient in volts/°K and ΔT = temperature difference across the couple in °K where $\Delta T = T_h - T_c$.

When a load is connected to the thermoelectric couple, the output voltage (V) drops as a result of internal generator resistance. The current passing through the load is given by

$$I = \frac{\alpha \times \Delta T}{R_C + R_L}$$

where I = generator output current in amperes, R_c = average internal resistance of the thermoelectric couple in ohms and R_L = load resistance in ohms.

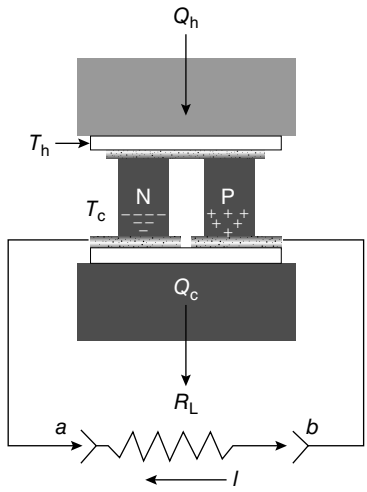


Fig. 14.77 *Single Thermoelectric Couple where $T_h > T_c$*

The total heat input to the thermocouple (Q_h) is

$$Q_h = (\alpha \times T_h \times I) - (0.5 \times I^2 \times R_c) + (K_c \times \Delta T)$$

where Q_h = the heat input in watts, K_c = the thermal conductance of the couple in watts/°K and T_h = the hot side of the couple in °K.

The efficiency of the generator (η_g) is given by $\eta_g = \dfrac{V \times I}{Q_h}$

Considering a complete module made of a number of couples, it is necessary to rewrite our equation for an actual module as follows:

$$V_o = \alpha_M \times \Delta T = I \times (R_M + R_L)$$

where V_o = the generators output in volts, α_M = module's average Seebeck coefficient in volts/°K and R_M = the module's average resistance in ohms.

In the above equation, the values of α_M, R_M and K_M must be selected at the average module temperature T_a given by

$$T_a = \frac{T_h + T_c}{2}$$

The power output (P_o) from the module in watts is given by

$$P_o = V_o I$$

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14.8.2 Thermionic Converter

It consists of two electrodes held in a container filled with ionized caesium vapour. Heating one electrode 'boils out' electrons that travel to the opposite colder side electrode. The positive ions in the gas neutralize the space charge effect of the electrons that normally prevent the flow of electrons. Ionized gas offsets space–charge effect that tends to repel the migration of electrons.

Electrons that are emitted by heating cathode are migrated to cooler anode collector and flow through outer circuit to develop electric power. Low work function materials such as barium and strontium oxides are used for anodes, whereas high work function materials such as tungsten impregnated with Barium is used for cathodes (Figures 14.78 and 14.79).

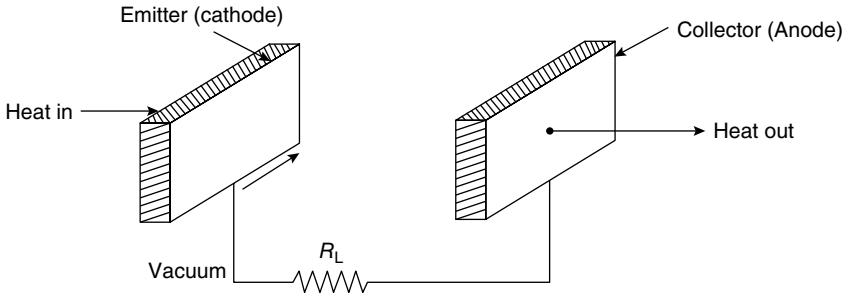


Fig. 14.78 *Thermionic Converter*

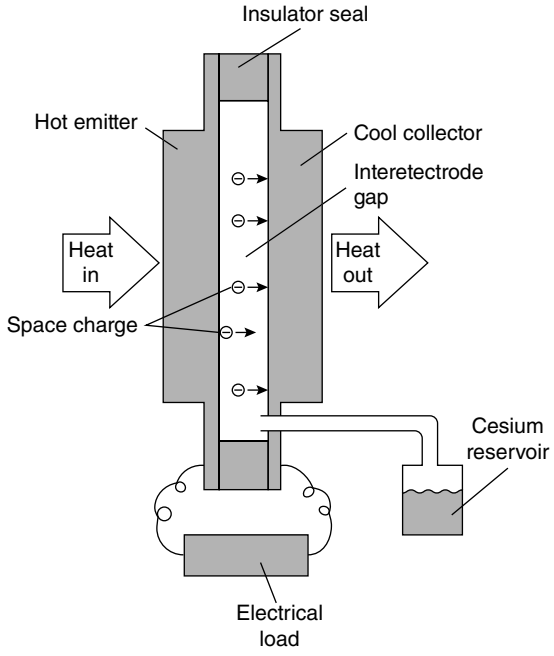


Fig. 14.79 *A Typical Thermionic Converter*

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14.9 MHD GENERATION

In a MHD, super-hot plasma is created by ionizing the atoms of the fuel mixture. The magnetic field deflects positive and negative charges in different directions. DC voltage is given by the collecting plates that can be converted into AC voltage (Figure 14.80).

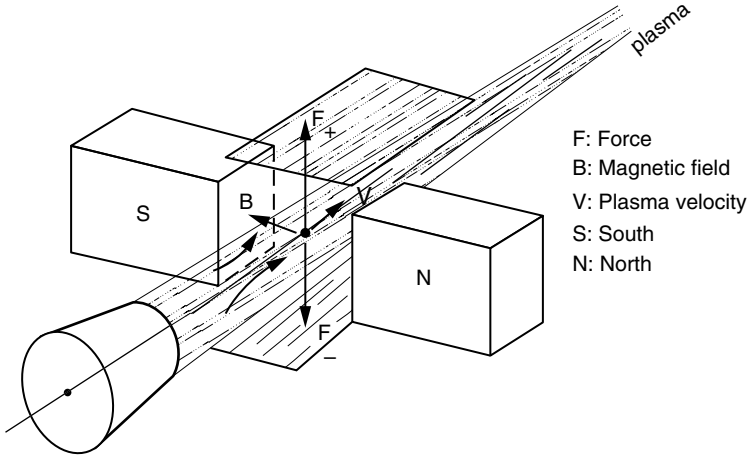


Fig. 14.80 *Working of a Magneto Hydrodynamic Generator*

Principle of Operation:

MHD generators are used for direct conversion of thermal energy into electrical energy. They work on Faraday principle. When an electric conductor moves across a magnetic field, a voltage is induced in it, which produces an electric current. In MHD generators, the solid conductors are replaced by a fluid that is electrically conducting. The working fluid may be either an ionized gas or liquid metal. The hot, partially ionized and compressed gas is expanded in a duct and forced through a strong magnetic field; electrical potential is generated in the gas. Electrodes placed on the side of the duct pick up potential generated in the gas. The DC thus obtained can be converted into AC using an inverter (Figure 14.81).

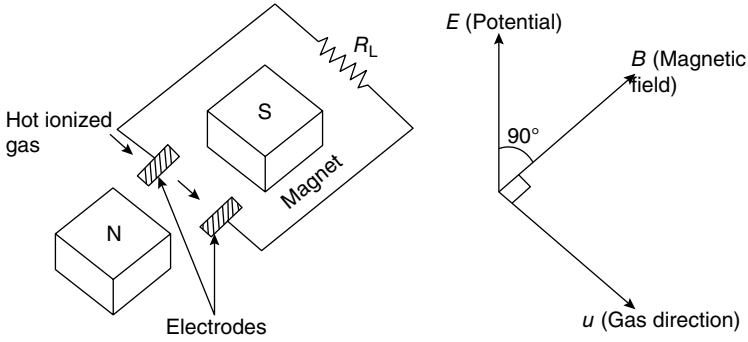


Fig. 14.81 *Principle of MHD Power Generation*

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The system is simple with large power- and temperature-handling capacity without any moving parts. It is highly reliable and can be brought to full load within 45 seconds. Power output can be changed from no load to full load in fraction of a second.

Allowing for the inefficiencies introduced by finite heat transfer rates and component inefficiencies in real heat engines, a system employing an MHD generator offers the potential of ultimate efficiency in the range of 60–65 per cent. This is much better than the 35–40 per cent efficiency that can be achieved in a modern conventional plant. In addition, MHD generators produce fewer pollutants than conventional plants. However, the higher construction costs of MHD systems have limited their adoption.

14.10 GEOTHERMAL ENERGY

Geothermal energy is heat transported from the interior of the earth, which is recoverable in the form of steam or hot water.

14.10.1 A Typical Geothermal Field

Figure 14.82 shows a typical geothermal field. The hot magma near the surface (A) solidifies into igneous rock (B). The heat of the magma is conducted upward to the igneous rock. Groundwater that finds its way down to this rock through fissures in it will be heated by the heat of the rock or by mixing with hot gases and steam evaporating from the magma. The heated water then rises upwards into a porous and permeable reservoir (C) above the igneous rock. This reservoir is covered by a layer of impermeable solid rock (D), which traps the hot water in the reservoir. The solid rock has fissures (E) that act as vents of the giant underground boiler. The vents are seen at the surface as geysers, fumaroles (F) or hot springs (G). A well (H) taps steam from the fissure for use in a geothermal power plant.

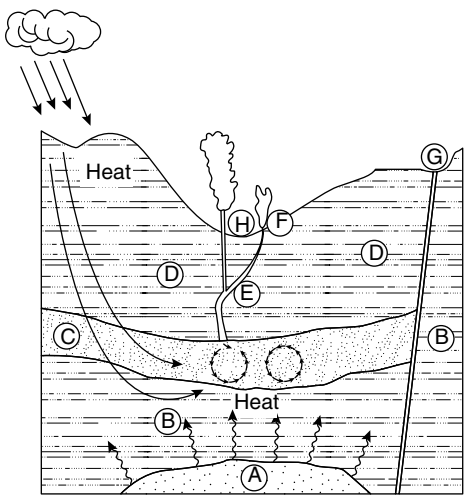


Fig. 14.82 *A Typical Geothermal Plant*

The geothermal steam is of the following two kinds:

(i) Originating from the magma itself, called magmatic steam.

(ii) Originating from groundwater heated by the magma called meteoritic steam. This is the largest source of geothermal steam.

Geothermal sources may be of the following three types:

(i) Hydrothermal

(ii) Geopressured

(iii) Petrothermal

14.10.2 Hydrothermal Systems

In Hydrothermal systems, water is heated by contact with solid rock (Figure 14.83). Hydrothermal systems may be (1) vapour dominated and (2) liquid dominated.

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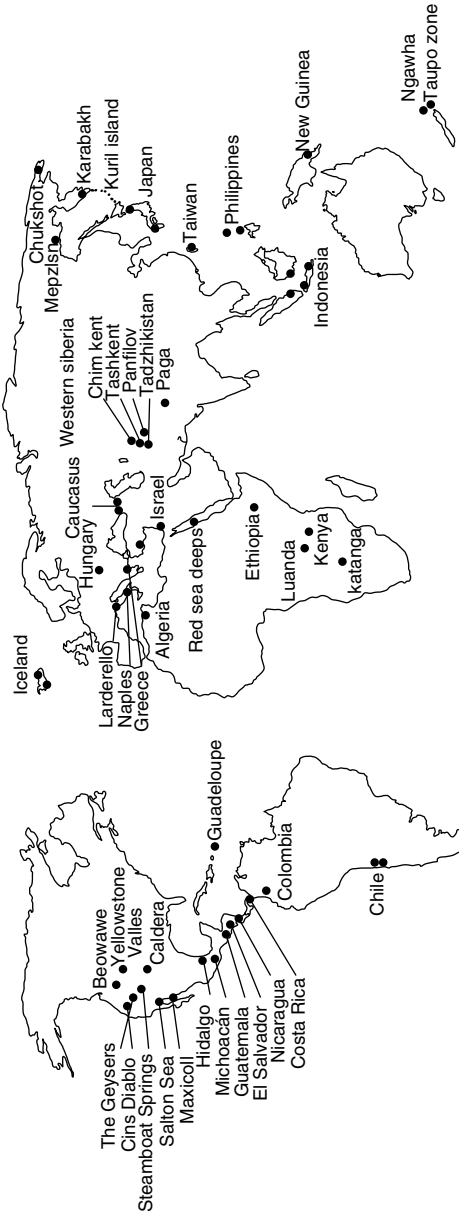


Fig. 14.83 *Worldwide Available Geothermal Fields*

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1. *Vapour-dominated systems*

Figure 14.84 shows a schematic and T–s diagram of a vapour-dominated power system. Dry steam from the well (1) at around 200°C and almost saturated at the bottom of the well at 35 bar is passed through the well head (2). Pressure drops through the well cause it to slightly superheat at the well head (2) at (~7 bar). It then goes through a centrifugal separator to remove particulate matter and then enters the turbine after an additional pressure drop (3). Processes 1–2 and 2–3 are throttling processes with constant enthalpy. The steam expands through the turbine and enters the condenser at 4.

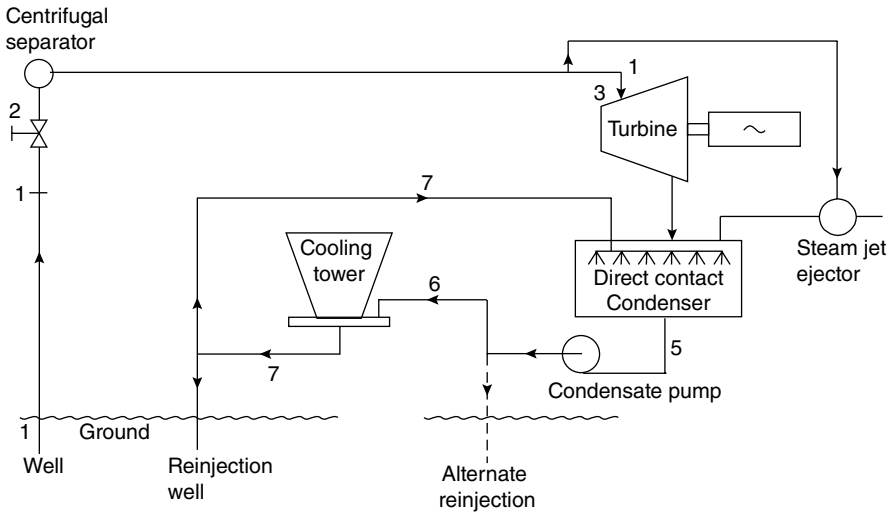


Fig. 14.84(a) *Schematic of a Vapour-Dominated Power Plant*

Because turbine flow is not returned to the cycle but re-injected back into the earth, a direct contact condenser of the barometric or low level may be used. The turbine exhaust steam at 4 mixes with the cooling water (7) that comes from a cooling tower. The mixture of 7 and 4 is saturated water at (5) that is pumped to the cooling tower (6). Some portion of cooled water is re-circulated back, while small portion is re-injected back to the ground.

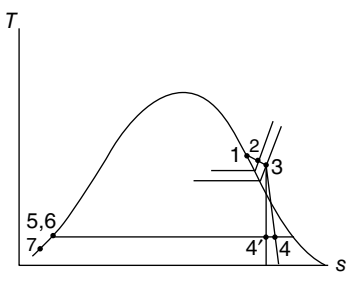


Fig. 14.84(b) *T–s Diagram of a Vapour-Dominated Power Plant*

2. *Liquid-dominated systems*

(i) *Flashed steam system*

This system is used for water in the higher temperature range. Water from the underground reservoir at 1 reaches the well head at 2 at a lower pressure. Process 1–2 is essentially a constant enthalpy throttling process that results in a two-phase mixture of low quality at 2. This is throttled further in a flash separator resulting in a still low but slightly higher quality at 3. This mixture is now separated into dry saturated steam at 4 and saturated brine at 5. The latter is re-injected into the ground.

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The dry steam, a small fraction of the total well discharge and at pressures below 8 bar, is expanded in a turbine to 6 and mixed with cooling water in a direct-contact condenser with the mixture at 7 going to a cooling tower in the same fashion as the vapour-dominated system. The balance of the condensate after the cooling water is re-circulated to the condenser and is re-injected into the ground (Figure 14.85).

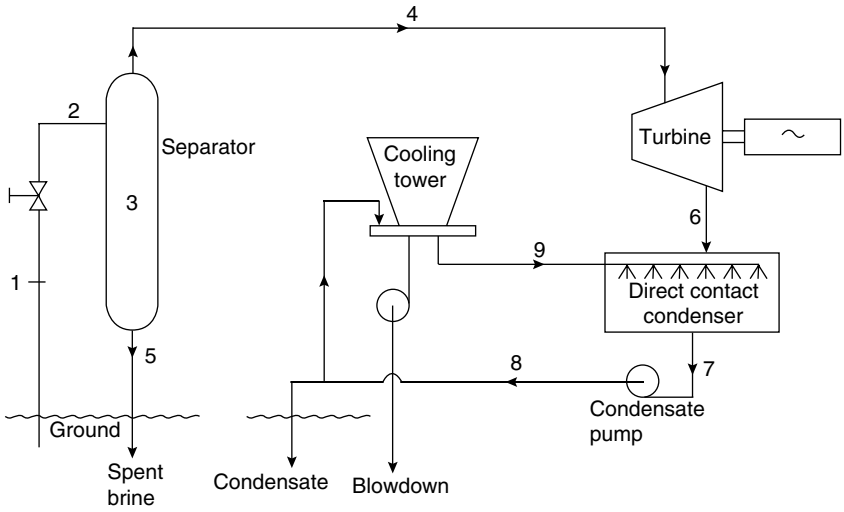


Fig. 14.85(a) *Liquid-Dominated Single-Flash Steam System*

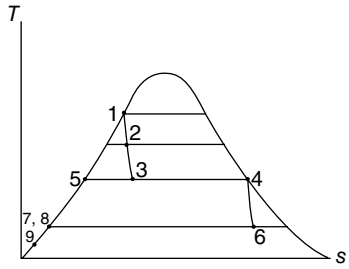


Fig. 14.85(b) *T–s Diagram of a Liquid-Dominated Single-Flash Steam System*

The flashed steam system is a more difficult proposition than the vapour-dominated system for several reasons:

(i) Much larger total mass flow rates through the well.

(ii) A greater degree of ground surface subsidence as a result of such large flows.

(iii) A greater degree of precipitation of minerals from the brine resulting in the necessity for design of valves, pumps, separator internals and other equipment for operation under scaling conditions and (4) greater corrosion of piping, well casing and other conduits.

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(ii) *Total flow concept*

In flashed steam system, some useful energy is discarded with the separated brine regardless of how many stages of separation are used. Thermodynamically, therefore, direct expansion of the fluid from the well head to the condenser has the potential of converting the greatest fraction of available energy in the fluid to mechanical work. This means that the total well-head flow is to be expanded to the condenser pressure. Hot brine from the well at 1 is throttled to 2, where it becomes a two-phase mixture of low quality. Instead of separating the two phases at this point, the full flow is expanded to 3 condensed to 4, and re-injected into the ground at 5. The flow in the flashed steam turbine and hence the work per unit flow from the well head is only a small fraction equal to the quality in the flash separator, x_3, of the total flow and work that would occur in the latter (Figure 14.86).

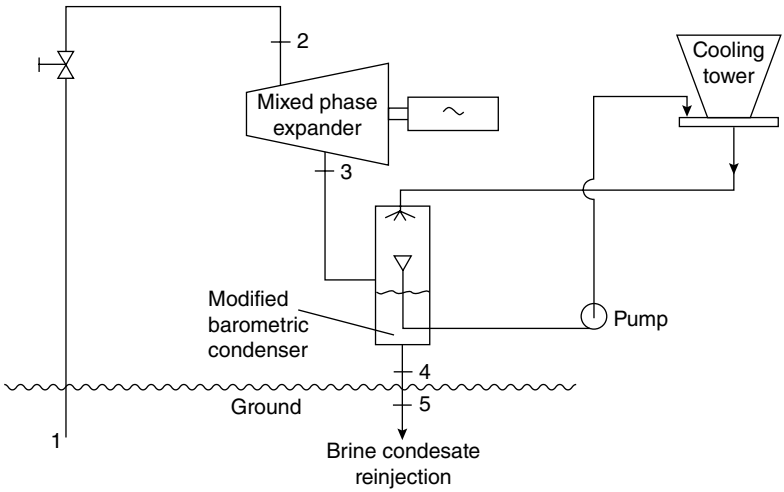


Fig. 14.86(a) *Liquid-Dominated Total Flow Concept*

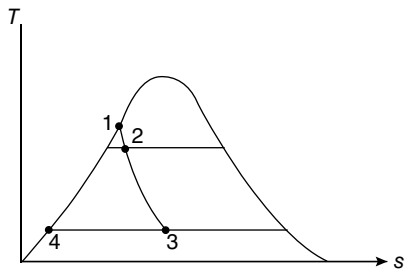


Fig. 14.86(b) *T–s Diagram of a Liquid-Dominated System*

14.10.3 Petro Thermal Systems

Petro thermal systems are composed of hot dry rock (HDR) but no underground water. They are the largest geothermal resources available. The rock occurring at moderate depths has very low permeability and needs to increase its heat transfer surface.

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The thermal energy of the HDR is extracted by pumping water through a well that has been drilled to the lower part of the fractured rock. The water moves through the fractures, picking up heat. It then travels up a second well drilled to the upper part of the rock and finally back to the surface. Further, it is used in a power plant to produce electricity. It envisages a fracture zone in crystalline rock with steam pipe at about 2300 m below the surface.

14.10.4 Hybrid Geothermal Fossil Systems

The hybrid geothermal fossil fuel system utilizes the relatively low-temperature heat of geothermal sources in the low-temperature end of a conventional cycle and the high-temperature heat from fossil-fuel combustion in the high-temperature end of the cycle. The concept thus combines the high efficiency of a high-temperature cycle with a natural source of heat for a part of the heat addition, thus reducing the consumption of the expensive and non-renewable fossil fuel.

There are two possible arrangements for hybrid plants. These are (i) geothermal preheat suitable for low-temperature liquid-dominated systems and (ii) fossil superheat, suitable for vapour-dominated and high-temperature liquid-dominated systems.

Fossil-superheat hybrid systems

In these systems, the vapour-dominated steam or the vapour obtained from a flash separator in a high-temperature liquid-dominated system is superheated in a fossil-fired super heater.

Figure 14.87 shows schematic flow and T–s diagrams of a proposed system. It comprises a double-flash geothermal steam systems. Steam produced at 4 in the first-stage flash separator is preheated from 4 to 5.

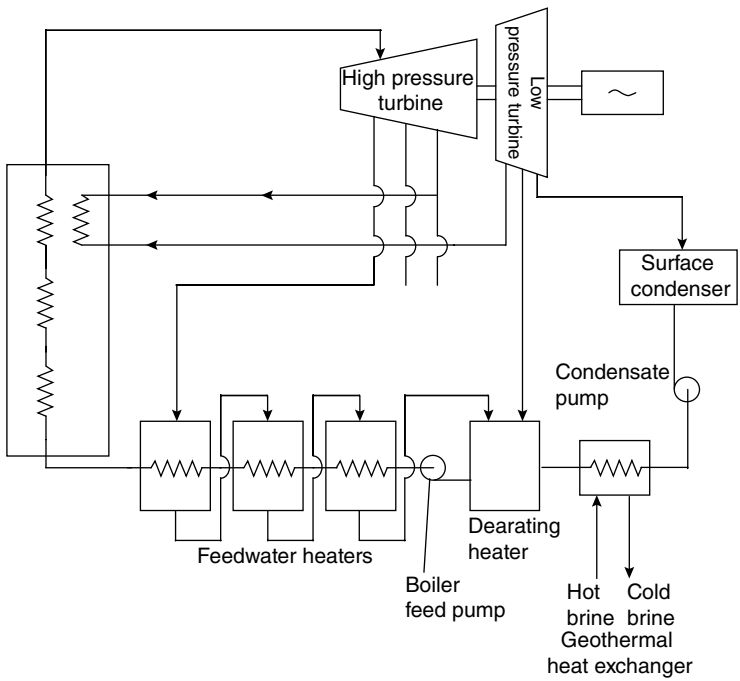


Fig. 14.87 *Schematic of a Geothermal Preheat Hybrid System*

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Figure 14.88(a) shows a fossil-superheat hybrid system with two-stage flash evaporation, regenerator and fossil-fired superheater.

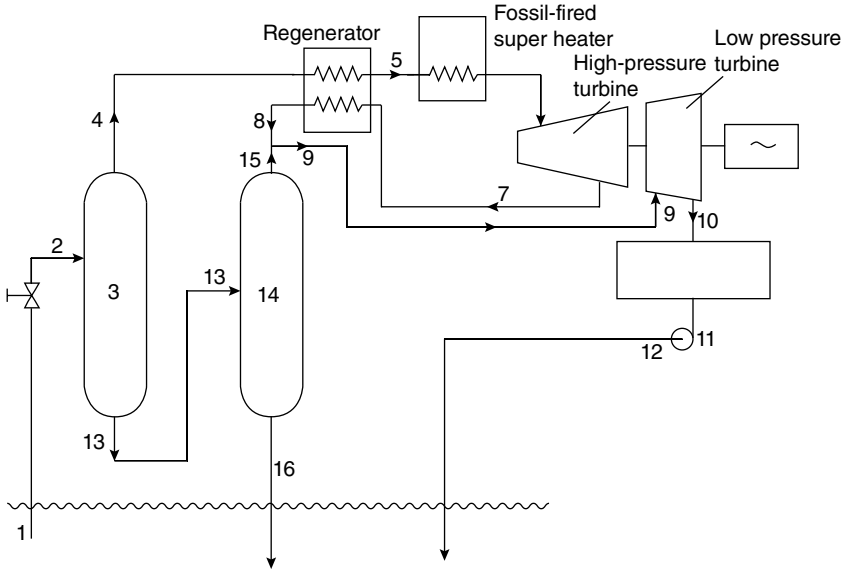


Fig. 14.88(a) *Schematic of a Fossil-Superheat Hybrid System*

Figure 14.88(b) shows flow and T–s diagrams of a proposed system. It comprises a double-flash geothermal steam systems. Steam produced at 4 in the first-stage flash separator is preheated from 4 to 5 in a regenerator by exhaust steam from the high pressure turbine at 7. It is then superheated by a fossil fuel-fired superheater to 6 and expands in the high-pressure turbine to 7 at a pressure neat that of the second-stage steam separator. It then enters the regenerator and leaves it at 8, where it mixes with the lower-pressure steam produced in the second-stage flash separator at 15, and produces steam at 9, which expands in the lower pressure turbine to 10. The condensate at 11 is pumped and re-injected into the ground at 12. The spent brine from the second-stage evaporator is also re-injected into the ground at 16.

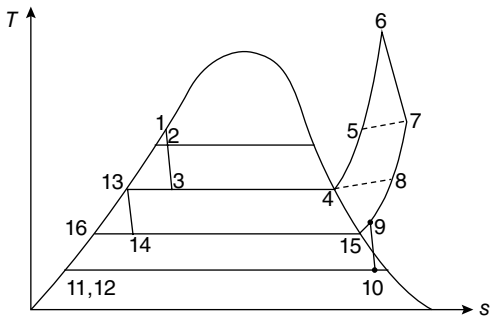


Fig. 14.88(b) *T–s Diagram of Cycle*

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14.10.5 Problems Associated with Geothermal Conversion

(i) Steam and water from both hydrothermal systems are present along with the dissolved solids in the water, entrained solid particles and non-condensable gases. The entrained solids must be removed as much as possible, usually by centrifugal separators at the well head, before they enter plant equipment and by strainers.

(ii) The non-condensable gas content varies from 0.2 to 4.0 per cent depending on the particular well and its age. The younger the well the higher the percentage as the non-condensables tend to vent out a bit faster than the H_2O. The non-condensables themselves are mostly CO_2 (about 80 per cent) plus varying amount of methane, CH_4; hydrogen, H_2; nitrogen, N_2; ammonia, NH_3; and hydrogen sulphide, H_2S. Besides finding their way with the fluid into the plant equipment, the non-condensables also partly escape to the atmosphere through the particle centrifugal separators, the condenser ejectors and in some cases the cooling towers.

(iii) The presence of the non-condensable gases has the following effects. A large quantity of these gases, relative to non-condensables in conventional steam systems necessitates the careful design of adequate gas ejectors to maintain vacuum in the condenser. The presence of acid-forming carbon steels and their corrosive effect in wet conditions forces the use of stainless steel in all equipments exposed to wet steam or condensate. Such equipment includes turbine corrosion shields and shaft seals, exhaust duct lining, condenser lining condensate lines and pump and metal parts in cooling towers.

(iv) Non-condensables are environmentally undesirable because they partly escape into the atmosphere. Most are corrosive in the normally damp atmosphere of the plant and are noxious and toxic and hence are major air pollutants. The most objectionable are H_2S and NH_3.

(v) Environmental problem caused by geothermal plants is land surface subsidence because of the extraction of large quantities of underground fluids. Large extractions and re-injections also have a possibility of seismic disturbances.

(vi) Noise pollution is another problem. Exhausts, blow downs and centrifugal separators are some of the sources of noise that necessitate the installation of silencers on some equipment.

(vii) Geo-pressured water carries large quantities of sand when high flows are required. This results in increased corrosion and scaling problems.

14.11 ELECTRICITY FROM CITY REFUGE

Electricity can be produced by burning 'municipal solid waste' (MSW) as a fuel. MSW power plants, also called waste to energy (WTE) plants, are designed to dispose of MSW and to produce electricity as a by-product of the incinerator operation.

The term MSW describes the stream of solid waste ('trash' or 'garbage') generated by households and apartments, commercial establishments, industries and institutions. MSW consists of everyday items such as product packaging, grass clippings, furniture, clothing, bottles, food scraps, newspapers, appliances, paint and batteries. It does not include medical, commercial and industrial hazardous or radioactive wastes, which must be treated separately.

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MSW is managed by a combination of disposal in landfill sites, recycling and incineration. MSW incinerators often produce electricity in WTE plants.

Burning MSW can generate energy while reducing the volume of waste by up to 90 per cent, an environmental benefit. Ash disposal and the air polluting emissions from plant combustion operations are the primary environmental impact control issues.

MSW contains a diverse mix of waste materials, some benign and some very toxic. Effective environmental management of MSW plants aim to exclude toxics from the MSW-fuel and to control air pollution emissions from the WTE plants.

Toxic materials include trace metals such as lead, cadmium and mercury, and trace organics, such as dioxins and furans. Such toxics pose an environmental problem if they are released into the air with plant emissions or if they are dispersed in the soil and allowed to migrate into ground water supplies and work their way into the food chain. The control of such toxics and air pollution are key features of environmental regulations governing MSW-fuelled electric generation.

Figure 14.89 shows a Nisargruna Biogas plants developed by BARC Mumbai based on thermophilic microorganisms and microbial processes, designed for MSW management. The plant is completely gravity based.

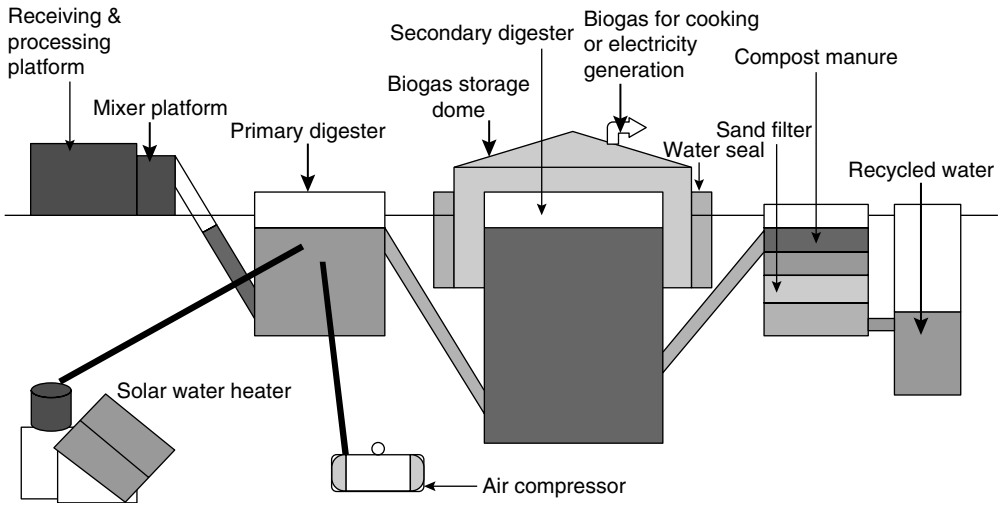


Fig. 14.89 *A Typical Nisargruna Biogas Plant (MSW)*

In this plant, the segregated wet garbage (food waste) brought to the plant site is loaded on a sorting platform. The residual plastic, metal, glass and other non-biodegradable items are further segregated and the waste is loaded into a waste crusher along with water, which is mounted on the platform. The food waste slurry mixed with hot water is directly charged into the primary digester.

The primary digester serves the purpose of a hydrolysis cum acidification tank for the treatment of suspended solids. Compressed air from a compressor is used for agitation of slurry for breaking the slag. Compressed air will also help in increasing aeration since bacteria

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involved in this tank are aerobic in nature. Once the system reaches equilibrium in initial the four to five days, the fresh slurry entering the tank will displace equal amount of digested matter from top into the main digester tank.

Main digester tank serves the purpose as a methane fermentation tank and BOD reduction takes place here. The treated overflow from this digester is connected to the manure pits. This manure can be utilized by farmers, municipal gardens and local gardens.

The biogas is collected in a dome (gas holder) is a drum-like structure, fabricated of either mild steel sheets or fibre glass-reinforced plastic. It fits like a cap on the mouth of digester where it is submerged in the water and rests in the ledge, constructed inside the digester for this purpose. The drum collects gas, which is produced from the slurry inside the digester as it gets decomposed and rises upwards, being lighter than air. The biogas produced can be used for cooking, heating and power generation purpose.

Some of the suitable locations for installation of plant are hotel premises, army/big establishment canteens (private/government), residential schools/colleges, housing colonies, religious places/temple trusts, hospitals, hotels, sewage treatment plants, villages, etc. Table 14.5 gives the technical details of a biogas plant.

Table 14.5 *Technical Details of Biogas Plant*

Capacity (tons/day)	Methane generation (m^3)	Manure production (kg/day)	Required area (m^2)	Cooking fuel (equivalent to LPG cylinder/day) or electricity generation
1	60	100	300	2–3 (5 KVA)
2	120	200	400	4–5 (10 KVA)
3	180	300	400	4–5 (10 KVA)
4	240	400	500	8–10 (20 KVA)
5	300	500	500	12–14 (25 KVA)
10	600	1000	800	22–25 (50 KVA)

14.12 QUESTIONS

14.12.1 Objective Questions

1. The commercial sources of energy are
   (a) solar, wind and biomass
   (b) fossil fuels, hydropower and nuclear energy
   (c) wood, animal wastes and agriculture wastes
   (d) none of the above
2. The main source of production of biogas is
   (a) human waste
   (b) wet cow dung
   (c) wet livestock waste
   (d) all of the above

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3. In fuel cell, _____ energy is converted into electrical energy.
   (a) mechanical (b) chemical
   (c) heat (d) sound
4. Solar thermal power generation can be achieved by
   (a) using focusing collector or heliostats (b) using flat plate collectors
   (c) using a solar pond (d) any of the above systems
5. The energy radiated by sun on a bright sunny day is approximately
   (a) 700 W/m^2 (b) 800 W/m^2
   (c) 1 kW/m^2 (d) 2 kW/m^2
6. Solar constant is
   (a) 140 Wm^{-2} (b) 1.4 Wm^{-2}
   (c) 1.4 kWm^{-2} (d) 1.4 MWm^{-2}
7. The radiation in the sunlight that gives us the feeling of hotness is
   (a) visible radiation (b) infra-red
   (c) red (d) ultra-violet
8. Which of the following is not a bio-mass source?
   (a) Gobar gas (b) Coal
   (c) Wood (d) Nuclear energy
9. The energy which is not derived from the sun is
   (a) biomass (b) fossil fuels
   (c) nuclear energy (d) geothermal energy
10. Harmful radiation emitted by the sun is
    (a) visible (b) infrared
    (c) ultraviolet (d) radio waves
11. Fuel formed under the earth's surface by the decomposition of organic matter is called
    (a) organic fuel (b) biogas
    (c) fossil fuel (d) underground fuel
12. Which of the following causes the least pollution when burnt?
    (a) Petrol (b) Diesel
    (c) Coal (d) Natural gas
13. The main constituent of LPG is
    (a) methane (b) butane
    (c) hydrogen (d) propane
14. The main constituent of CNG is
    (a) methane (b) butane
    (c) ethane (d) propane
15. Which of these is not a renewable source of energy?
    (a) Sun (b) Natural gas
    (c) Wind (d) Ocean tidal energy

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16. The radiations absorbed by ozone layer are
    (a) infrared (b) ultraviolet
    (c) visible (d) gamma rays
17. The efficiency of the solar cooker can be increased by placing a
    (a) plane mirror (b) convex mirror
    (c) convex lens (d) concave lens
18. The maximum temperature that can be attained in a box-type solar cooker is
    (a) 200°C (b) 140°C
    (c) 80°C (d) 250°C
19. A solar cell is made up of
    (a) silicon (b) titanium
    (c) magnesium (d) teflon
20. Which of the following is not combustible?
    (a) oxygen (b) hydrogen
    (c) butane (d) methane
21. In a solar panel, the metal used is
    (a) gold (b) copper
    (c) silver (d) nickel
22. A solar cell converts
    (a) heat energy into electrical energy
    (b) solar energy into electrical energy
    (c) heat energy into light energy
    (d) solar energy into light energy
23. Floating generators are used in the sea to harness
    (a) tidal energy (b) wave energy
    (c) hydel energy (d) energy from OTEC power plant
24. The temperature difference between the upper layers and the deeper layers of the ocean should be _____ to install an OTEC power plant.
    (a) 40°C (b) 50°C
    (c) 20°C (d) 30°C
25. The molten material mixed with gases in the mantle of the earth is called
    (a) core (b) lava
    (c) geyser (d) magma
26. The source of energy of the sun is
    (a) nuclear fission (b) chemical reaction
    (c) nuclear fusion (d) photoelectric effect

Answers

1. b	2. d	3. b	4. d	5. c	6. c	7. c	8. d	9. c	10. c
11. c	12. c	13. c	14. c	15. c	16. c	17. c	18. c	19. c	20. c
21. c	22. c	23. c	24. c	25. c	26. c				

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14.12.2 Review Questions

Solar energy

1. Briefly discuss the different non-conventional energy resources.
2. Explain the importance of solar energy power generation with reference to present world energy scenario.
3. Define solar constant.
4. Differentiate between beam and diffuse radiation.
5. Define the following terms:
   (a) Incident angle
   (b) Zenith angle
   (c) Azimuth angle
   (d) Declination angle
   (e) Latitude angle
   (f) Hour angle
6. Differentiate between a pyrheliometer and pyranometer.
7. Explain the principle of conversion of solar energy into heat.
8. With a neat sketch explain the function of a flat plate solar collection.
9. What are the advantages and disadvantages of concentrating collectors over flat-plate collections?
10. Explain the working principle of a photovoltaic cell.
11. Calculate the sunset hour angle and day length at a location latitude of 40°N on February 14.
12. Estimate the daily global radiation on a horizontal surface at 22°13′N, 73°13′E during the month of May where a = 0.28; b = 0.48 and average sunshine hours for day.

Wind energy

1. Explain the working principle of wind energy conversion system.
2. Discuss the availability of wind energy in India.
3. For a horizontal axis wind machine show the maximum power obtained.
   $$P_{max} = \rho A V^3$$
   When exit velocity is one-third of wind velocity.
4. With a neat sketch explain the working of a wind energy conversion system with main components.
5. How wind energy systems are classified?
6. Discuss the advantages and disadvantages of wind energy conversion system.
7. With a neat sketch explain a horizontal axis wind machine.
8. With a neat sketch explain the working of a vertical axis wind mill.
9. Discuss on aerodynamic considerations of a wind turbine.
10. Define the following terms:
    (a) Torque coefficient
    (b) Tip speed ratio
    (c) Power coefficient

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11. Discuss briefly about 'yaw control'.
12. Write short notes on the following:
    (a) Savonius rotor
    (b) Darrius rotor
    (c) Problems associated with wind power
    (d) Availability of wind energy in India
13. Discuss the major problems associated with wind power.

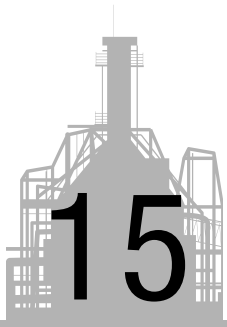
Tidal energy

1. Briefly explain tidal energy conversion systems.
2. What is wave energy? Explain the working principle of a wave energy conversion system.
3. With a neat sketch explain the following:
    (a) Wave energy conversion by floats
    (b) High-pressure accumulator wave machine
4. Sketch and explain the following:
    (a) Single-pool tidal system
    (b) Double-pool tidal system
5. Explain the working principle of an OTEC plant.
6. With a neat sketch explain the working of a
    (a) Clade OTEC plant
    (b) Anderson OTEC plant
7. Enlist different geothermal energy sources.
8. With a neat sketch explain a typical geothermal field.
9. What are the problems associated with geothermal conversion?
10. With a neat sketch explain the working of the following systems:
    (a) A vapour-dominated power plant
    (b) A liquid-dominated power plant
11. With a neat sketch explain a liquid-dominated single-flash steam system.
12. Write short notes on petrothermal systems.
13. An OTEC power plant of Claude type has an ideal turbine operating between 25 and 15°C. For a turbine output of 2 MW, calculate
    (a) the mass flow rate of steam
    (b) the mass flow rate of warm water if the evaporator temperature drop is 2°C
14. A 3 m wave has a g–s period occuring at the surface of water 100 m deep. Find the
    (a) wavelength
    (b) wave velocity
    (c) energy and power densities

    Take $\rho_w = 1025$ kg/m^3

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Hydroelectric Power Plant



15

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15.1 Introduction to water power
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15.7 Classification of hydro-plant
15.8 Surge tanks
15.9 Control gates and valves
15.10 Classification of dams and spillways
15.11 Brief description of some of the important hydel installations in india
15.12 Water turbines
15.13 Micro hydel developments
15.14 Project cost of hydroelectric plant
15.15 Advantages of hydro-power plant
15.16 Questions

15.1 INTRODUCTION TO WATER POWER

In the development of a country, the hydro power plays a very important role because it provides power at cheapest rate. Nearly, 20 per cent of the total world power is generated using hydro plants. To countries in the world, 90 per cent of the nation's power requirement is met by hydro power. Countries such as Russia and Nepal have vast hydro resources that are yet to be harnessed. As per the estimate of the World Power Organization (WPO), the world hydro potential is roughly 5,000 GW, whereas only 4 per cent of it (200 GW) is currently developed.

Rain falling upon the earth's surface has potential energy relative to oceans towards which it flows. This energy can be converted into shaft work by hydraulic prime mover and finally into electrical energy. The shaft power developed by the water passing through the prime mover is given by the following relation:

$$P = \frac{mgH}{1000} \times \eta_h \times \eta_m \times \eta_g$$

where m = rate of water flow in kilogram/second
H = height of fall in meters

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η_h = hydraulic efficiency of prime mover
η_m and η_g are mechanical and generating efficiencies, respectively

or $$P = \frac{mgH}{1000} \times \eta_{overall}$$

where

overall efficiency, $= \eta_h \, \eta_m \, \eta_g$

Generation of power depends on the quantity of water available and head. The available head depends on the selection of site for hydraulic power plant. The site of the power plant is always selected for the highest available head when other things are in favour of site selection.

The quantity of water available at the selected site depends on the hydrological cycle of nature. The quantity of water available can be determined from the study of rainfall and run-off in that area. As the availability of water depends on the natural phenomenon of rain, the maximum capacity of hydraulic generating plant is usually fixed on the basis of minimum quantity of water available. Usually, storage reservoirs are constructed for such plants in order to store the water during peak periods of run-off and supply the same during off-peak periods of run-off.

15.2 HYDROLOGICAL CYCLE

It is the science that deals with rainfall and run-off. The evaporation of the water from the surfaces of river and oceans and its precipitation on the earth is known as hydrological cycle. Based on hydrological aspect, the distribution of precipitation on the earth surface and beneath the earth is calculated.

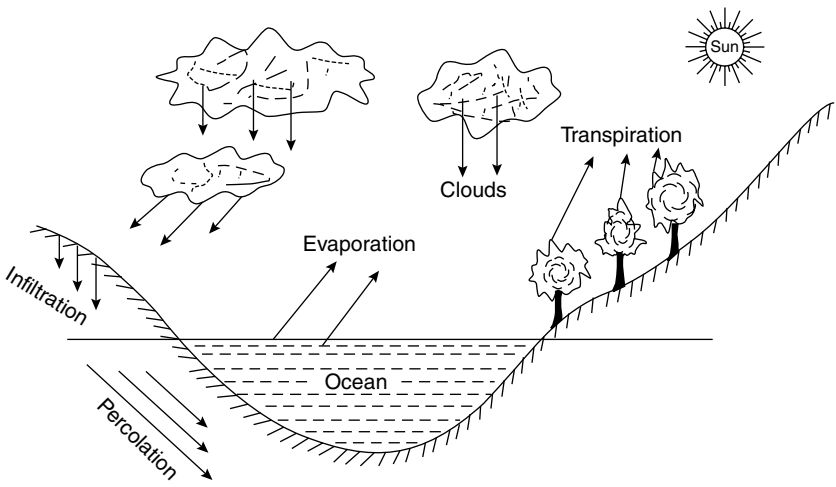


Fig. 15.1 *Hydrological Cycle*

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Water is evaporated from plants, rivers and oceans and carried with the air in the form of vapour, which is known as clouds. When the vapours in the atmosphere are cooled below dew point temperature, it falls in the form of water or snow depending on the atmospheric temperature. This evaporation and precipitation is a natural continuous process, and it therefore constitutes a perennial source of energy. Figure 15.1 shows the processes of evaporation and precipitation.

Note:

(i) *Precipitation:* This is the total condensation of moisture that reaches earth in any form, e.g. rainfall, snow, hail, etc.

(ii) *Evaporation:* This is the total portion of rainfall that (does not reach earth's surface but) evaporates back to atmosphere. From land and water surface, it includes the following:

(a) total evaporation from land and water surfaces

(b) evaporation by transpiration from vegetation

(c) atmospheric evaporation

The portion of total precipitation that reaches the catchment area is known as run-off. Therefore, run-off = total precipitation − total evaporation.

(iii) *Catchment area:* Catchment area of a hydro-site is the total area behind the dam, draining water into the reservoir.

Part of precipitation is absorbed by soil and seeps into the ground and finally reaches the catchment area through underground channels.

Total run-off = direct run-off over the land surface + run-off through seepage

The unit of run-off is cubic meter/second or day–second–metre

1 day–sec–m = discharge collected in the catchment area at the rate of 1 m^3/s or for 1 day

$= 1 \times 24 \times 3{,}600 = 86{,}400$ m^3/day.

15.3 RAINFALL AND ITS MEASUREMENT

The 'rainfall', also known as 'precipitation', is a natural process of converting atmospheric vapour into water.

During summer, the evaporation loss from free water surface is considerably large, and this evaporated water finds room in the air mass. The water holding capacity of air in the form of vapour is also considerably large in hot weather. When this highly moist air mass cools down by some means below the dew point temperature, the water vapour from the air precipitates in the form of water and reaches the earth, which is known as rainfall.

Rainfall in general may be defined as the total condensation of moisture from the atmosphere that reaches the earth, including all forms of rains, ice and snow. The rate of rainfall is expressed in centimetres of water during a time period. A rainfall of 1 cm is the quantity of water collected on a certain area due to rainfall, which becomes 1 cm in height. It is assumed that there are no losses due to evaporation, seepage and whatsoever, and there is no run-off also during the accumulation of water.

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The rainfall varies widely from one part of the world to another, ranging from desert regions to the hilly regions where the average annual rainfall may be over 1,100 cm. In some regions, there is a slight seasonal variation and monthly rainfalls are relatively uniform. In other regions, there is wide seasonal variation and monthly rainfalls also differ widely.

The annual rainfall at any given station varies irregularly from year to year. The range of this variation marks the reliability of the rainfall and has great importance in the design of storage reservoirs.

15.3.1 Intensity of Rainfall

The intensity of rainfall is expressed as an amount of precipitation in a stated period. During heavy rainstorm, the intensity of rainfall varies widely from minute to minute and can be measured only with a continuously recording gauge. Intense rainfall is usually of limited extent and duration. The greater the area under consideration, the smaller the average intensity of rainfall. Further, the longer the duration of rainstorm, the smaller the intensity of average rainfall.

The relation between the area of rainstorm and its average intensity is used to assess the amount of rain, which may be expected to fall upon a catchment area within a given period. The intensity of rainfall is equally important in the design of spillways at the time of heavy rainfall.

The intensity of rainfall is given by I (Intensity in inches/hour) = where, T is the duration of rain storm in hours and R and C are the constants quoted by different authorities for different areas in the world. The intensity also depends on the area selected for the measurement (Figure 15.2).

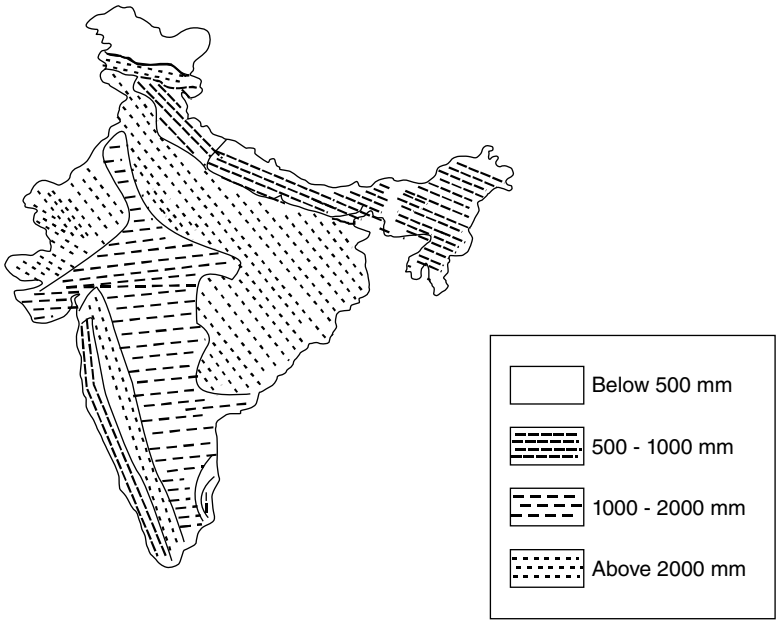


Fig. 15.2 *Yearly Rainfall in India in Millimetres*

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15.3.2 Measurement of Rainfall

All forms of precipitation are measured on the basis of vertical depth of water that would accumulate on the level surface if all the precipitation remained where it fell. The rainfall is generally measured with the help of rain gauges. Rain gauges are mainly of two types: non-recording and recording.

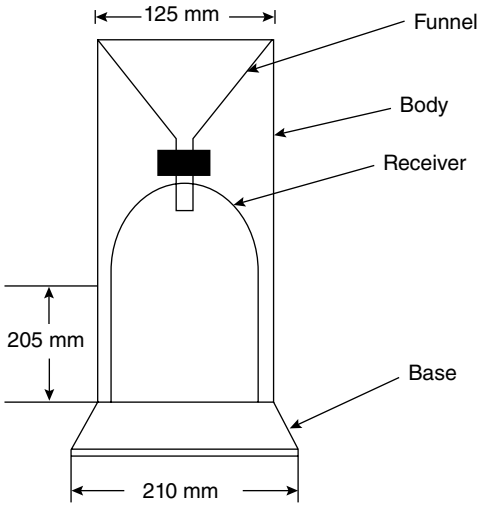


Fig. 15.3 *Non-Recording-Type Gauge*

(i) *Non-recording-type gauge*

If the water from the rainfall is collected before the losses take place, then the depth of the water over the small area can be accurately determined to find out the rainfall in millimetre. The area selected for setting the gauge should represent the large area whose meteorological characteristics are the same. The non-recording gauge usually consists of a standard funnel discharging into a receiver that is large enough to hold the maximum possible day's rainfall. The funnel and receiver are placed in a metal casing with suitable packing. At regular intervals, the water is collected into a measuring vessel that is so calibrated in relation to the area of the funnel mouth that directly reads the rainfall in centimetres or millimetres. The interval used for measuring the rainfall generally depends on the intensity of rainfall. The arrangement of the rain gauge can be seen in Figure 15.3. The base of the gauge is permanently fixed to the concrete block at a site where rainfall is to be measured. The precaution is taken at the time of fixing the level perfectly. The gauge is fixed to the block in such a way that the top of the gauge will be 30 cm above the natural surface level. The gauging station is generally protected by wire fencing with gate.

(ii) *Recording-type gauge*

The recording-type gauge is shown in Figure 15.4. A standard funnel is provided on the top of a rectangular box as shown in the figure. A float is adjusted in the same box. The float is connected to a pin point by means of a float rod. The pin point touches the graph paper mounted on a rotating drum as shown in the figure. The electric motor helps the drum to rotate continuously. Generally, the drum makes one rotation in a 24-hr period. When the rainfall occurs, water passes into the rectangular box. The float rises as the level in the box rises. As the float rises, the pin-point moves up on the graph paper to plot the mass curve (cumulative) of rainfall.

When the box gets filled to a level shown by AA the siphon starts working, and all the collected water is drained out from the box. The graph on the rotating drum gives the mass curve of the rainfall.

In the design of hydroelectric projects, the rainfall record of nearly 35 yr is required. This period is accepted as giving a reasonably accurate assessment of mean annual rainfall.

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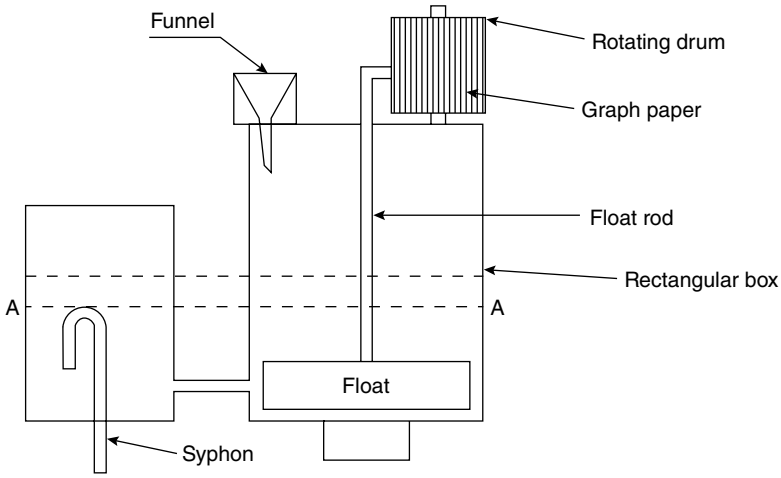


Fig. 15.4 *Recording-Type Gauge*

Many times, it is not possible to wait many years for rainfall data; and in such cases, shorter periods are accepted by design engineers with reserve and allowance is made for possible error.

15.3.3 Location of Rain Gauge

The following points are taken into consideration while selecting the site for the rain gauge station:

(i) The rain gauge station should be easily accessible to the observer.

(ii) The gauge should be erected on a level ground.

(iii) A wind shelter should be provided to reduce the effect of wind.

(iv) The rain gauging station should not be close to the trees and buildings. The distance of the gauge from every object should not be less than twice the height of the object above the gauge.

(v) The location of rain gauge should be true representative of the area whose average rainfall is to be measured.

15.3.4 Average or Mean Depth of Rainfall

The primary intention of rain gauging in hydroelectric work is to determine the monthly and annual rainfalls over the catchment area. When the rain gauging stations are more than one for a particular basin, the following methods are used for calculating the average of rainfall depending on the area of the basin: arithmetic mean method, Thiesson method and isohyetal method.

(i) *Arithmetic mean method*

In this method, the values of rainfalls of all the stations are added, and the sum is divided by the number of gauging stations. This can be represented mathematically as follows:

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$$h_a = \frac{h_1 + h \ldots + h_n}{n} = \frac{\sum h}{n}$$

If the gauges are uniformly distributed and the variation of individual gauge reading from the mean is not large, this method gives the reading as accurate as any other method. This method is conveniently used when the area of the basin in less than 500 km.

(ii) *Thiesson method*

This method is very accurate and is generally used when the catchment area lies between 500 and 5,000 km^2. The use of this method is best explained using the following example. Assume that there are stations in the given basin named as 1, 2, 3, 4, 5, 6, 7 and 8. It is very important to divide the total basin area in such a way that each located station in the basin represents that area in the true sense.

In this method, the gauging stations are marked on the basin area as shown in Figure 15.5. All the stations are joined to each of the adjacent stations as indicated by dotted lines so as to form a system of triangles. Each rain gauging station forms a vertex of a triangle. After forming the triangles, the perpendicular bisectors of each of the sides of all triangles are drawn by full lines as shown in Figure 15.5. This construction divides the whole basin area into eight polygons. Each polygon encloses only one gauging station, and it is the domain of that rain gauging station. After forming the polygons, the area of each polygon is calculated using a planimeter.

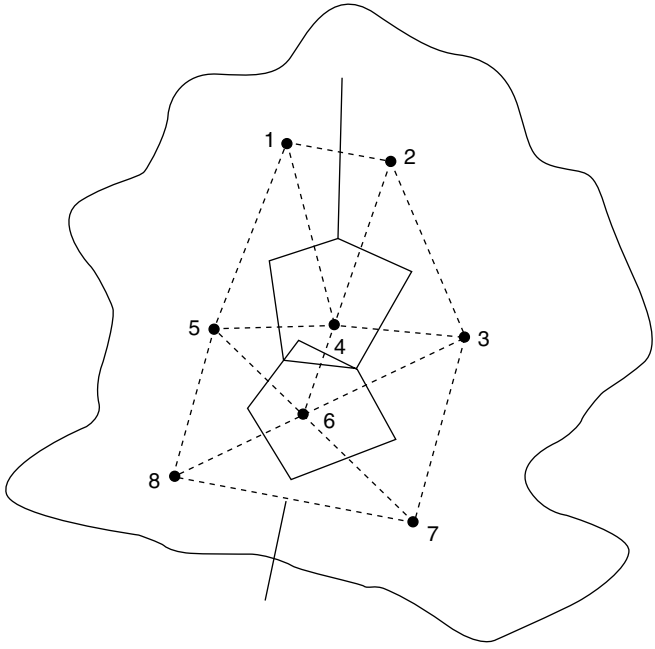


Fig. 15.5 *Construction of Thiesson Method*

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The measured gauge readings and areas of the polygons enclosing the gauges are tabulated as follows:

Station	Rainfall (cm)	Area of polygon fall in polygon	Total volume in rainfall in polygon
(a)	(b)	(c)	(d) = b × c
1	h_1	A_1	$h_1 \times A_1$
2	h_2	A_2	$h_2 \times A_2$
3	h_3	A_3	$h_3 \times A_3$
4	h_4	A_4	$h_4 \times A_4$
5	h_5	A_5	$h_5 \times A_5$
6	h_6	A_6	$h_6 \times A_6$
7	h_7	A_7	$h_7 \times A_7$
8	h_8	A_8	$h_8 \times A_8$

Column (a) represents the number of station, column (b) represents the average rainfall at each gauging station and column (c) represents the area of each corresponding polygon.

The mean rainfall of the basin is given by

$$h_a = \frac{\sum d}{\sum c}$$

$$= \frac{h_1A_1 + h_2A_2 + h_3A_3 + h_4A_4 + h_5A_5 + h_6A_6 + h_7A_7 + h_8A_8}{A_1 + A_2 + A_3 + A_4 + A_4 + A_5 + A_6 + A_7 + A_8}$$

Thiesson method gives better result than arithmetic mean method because each point on a perpendicular bisector of the line joining the two gauging stations will be equidistant from both the stations. If one moves slightly from either side of the bisector, the position will distinctly fall in the domain of that station of which ones position is now nearer.

(iii) *Isohyetal method*

Isohyete is a contour joining the points of equal rainfall in the given catchment area. Iso-hyetes have the following properties:

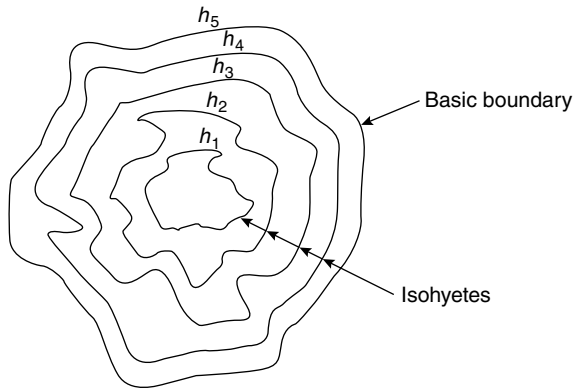


Fig. 15.6 *Isohyetes on the Catchment Area*

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(a) Two different isohyetes do not cross each other.

(b) Each isohyete closes on itself.

(c) Isohyete of higher value indicates the higher rainfall.

In this method, isohyetes are drawn in given basin by joining the points of equal rainfall as shown in Figure 15.6. The points of equal rainfall can be computed from the rainfall values of rain gauging stations.

Generally, interval of the isohyete is 1 cm. After drawing the isohyetes, the areas between each two successive isohyetes are measured using a planimeter. The rest or the procedure of finding out the mean rainfall is similar to that described in the second method. The highest spot rainfall value in area is h_1. The results of the method are tabulated as follows:

Isohyetol interval	Mean	Area between two successive isohyetes	Mean × Area
(a)	(b)	(c)	(d) = b × c
$h_1 - h_2$	$(h_1 + h_2)/2$	A_1	$(h_1 + h_2)A_1/2$
$h_2 - h_3$	$(h_2 + h_3)/2$	A_2	$(h_2 + h_3)A_2/2$
$h_3 - h_4$	$(h_3 + h_4)/2$	A_3	$(h_3 + h_4)A_3/2$
$h_4 - h_5$	$(h_4 + h_5)/2$	A_4	$(h_4 + h_5)A_4/2$
h_5 – boundary	$(2h_5 - 1)/2$	A_5	$(_2h_5 - _1)A_5/2$

In this analysis, the difference between h_1 and h_2, $(h_1 - h_2)$ may not exactly be one as h_1 represents the highest rainfall. The difference between the isohyete h_5 and boundary is considered as one which is the common interval. The interval between the isohytes h_2 and h_3, h_3 and h_4, h_4 and h_5 is 1 cm.

The mean rainfall of the basin is given by

$$h = \frac{\sum d}{\sum c}$$

$$h = \frac{(h_1 + h_2)A_1 + (h_2 + h_3)A_2 + (h_3 + h_4)A_3 + (h_4 + h_5)A_4 + (2h_5 - 1)A_5}{2[A_1 + A_2 + A_3 + A_4 + A_5]}$$

This method is commonly used when the basin area is over 500 km^2.

The analysis of rainfall for power generation purposes is usually more concerned with dry years rather than wet years. For the assessment of storage requirements, it is often necessary to determine the driest period for which provision must be made.

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15.3.5 Run-off and its Measurement

As the rain falls upon the drainage basin, a portion of it is evaporated directly by the sun; another large portion is taken up by vegetation and growing crops, particularly in growing season and some percolates into the ground. The percolating water may be partly absorbed by the roots of vegetation, or it may reappear in course of time as spring water either inside or outside the catchment area. The remaining portion of rainfall flows through the catchment area on the surface of the earth and is known as run-off. The suitability of run-off for power generation depends on the magnitude and time distribution of flow and head made available by the surrounding topography. In general, the run-off is given by

$$R = P - L$$

where R (run-off) $= R_s + R_c$
P = precipitation by rainfall
L = all losses
R_s = run-off over the surface
R_c = run-off reaching the catchment area through the pervious earth

Finding out the run-off rate through the catchment area is the first phase in the design of hydroelectric power project. The unit of run-off is cubic meter/second or* day–second–metre, or it is also expressed in centimetre of water depth over the entire catchment area or** square kilometer-centimeter/hour.

In the absence of stream flow records, the recorded run-off of the neighbouring stream is used to estimate the probable amount and distribution of run-off of a stream by a comparison of the relative run-off characteristics of the two watersheds.

The factors that affect the run-off are described below:

(i) *Rainfall pattern*

The run-off is more if the rainfall is heavy. The run-off increases more rapidly with increase in rainfall because the time allowed for evaporation and percolation losses is less when the intensity of rainfall is high. If the duration of the rainfall is more, the run-off will also be prolonged because the soil tends to become saturated and lowers the rate of seepage and humid atmosphere slows evaporation.

(ii) *Nature of catchment area*

The topography, shape, its vegetal cover and nature of the surface and sub-surface geology have a great influence on the run-off characteristics of the catchment area. The steep and rocky surface gives more run-off.

(iii) *Shape and size of the catchment area*

Large catchment area gives more run-off. The run-off at the point X as shown in Figure 15.7(a) is more than that at the point X as shown in Figure 15.7(b); even the catchment area of both is same, the reason for which is obvious from the figures itself.

* one day second metre is the flow collected at the rate of 1 cum/s for one day $= 1 \times 24 \times 60 \times 60 = 86400$ m^3/day.

** km^2-cm/hr $= 1000 \times 1000 \times \frac{1}{100} \times \frac{1}{3600} = 2.78$ m^3/s.

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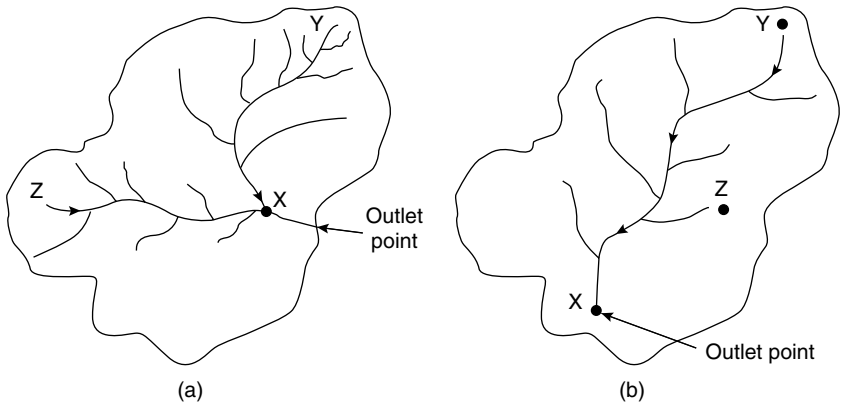


Fig. 15.7 *Distribution of Run-off over the Catchment Area*

(iv) *Vegetation*

The transpiration and interception losses depend on the nature and extent of vegetation including crops. Vegetation, particularly of forest, has considerable effect upon the run-off. It consumes a proportion of the rainfall, causes interception losses and provides physical obstruction for run-off.

(v) *Geology of the area*

The geology of the catchment area is of fundamental importance in the consideration of run-off. Rocky area gives higher run-off than softy or sandy area.

(vi) *Weather conditions*

Low temperature, high relative humidity and low winds result in high run-offs because the evaporation losses increase with the increase in temperature, decrease in relative humidity and increase in wind velocity. The run-off is also reduced by water absorbed by the hot earth.

15.3.5.1 Run-off Measurement

To determine the available energy in a given river, knowledge of the quantity of water flowing and its variation with time over a long period of years is essential. For measuring the run-off, the following methods are used:

(i) *Usage of rainfall records*

The lack of availability of direct run-off data often makes it necessary to derive the run-off characteristics from rainfall observations and estimated losses, from the records.

The run-off is calculated from the rainfall data available for sufficiently long period by multiplying with a coefficient known as 'run-off coefficient'. The value of run-off coefficient is decided considering all the factors that affect run-off. The following table gives the value of run-off coefficient (k) for different types of catchment areas.

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Values of run-off coefficients (*K*) for various surfaces		
S. no.	**Surfaces**	***K***
1.	Garden apartment	0.50
2.	Forest areas depending on soil	0.50–0.10
3.	Commercial and industrial	0.90
4.	Asphalt of concrete pavement	0.85
5.	Parks, farmland, pasture	0.05–0.30

Hence, run-off = rainfall × run-off coefficient.
This method is generally used for small catchment area due to non-availability of run-off data.

Limitations
This is an uncertain method of estimating run-off because of the following reasons:

(a) wide range of conditions encountered in a large catchment area
(b) the difficulty of measuring the losses accurately
(c) the manner in which they sometimes vary throughout the season

(ii) *Run-off formulae*
The following relations are used to calculate run-off at specific regions.

(a) *Khosla's formula*

$$R = P - 4.811\ \mathrm{T}$$

where R = run-off in millimetre
P = rainfall in millimetre
T = mean temperature in degree Celsius

(b) *Inglis formulae, for area in Maharashtra*

$$R = 0.08\ P - 304.8 \text{ for Ghat area,}$$

and

$$R = \frac{P - 177.8P}{2540} \text{ for area other than Ghat}$$

These formulae are mostly based on observations made by a person and extensive experiments conducted by individual. These formulae are useful only for specific regions.

(iii) *Actual gauging*
It is the only precise method of evaluating run-off by direct measurement using stream gauging at a given site for a long enough time.

In this method, the water flow volume through a selected channel of fixed cross section is measured by measuring the velocity of water at enough points for different water levels. The depths at different points are measured using a rope at regular intervals. The mean velocity in each section is measured with the float method or current meter directly.

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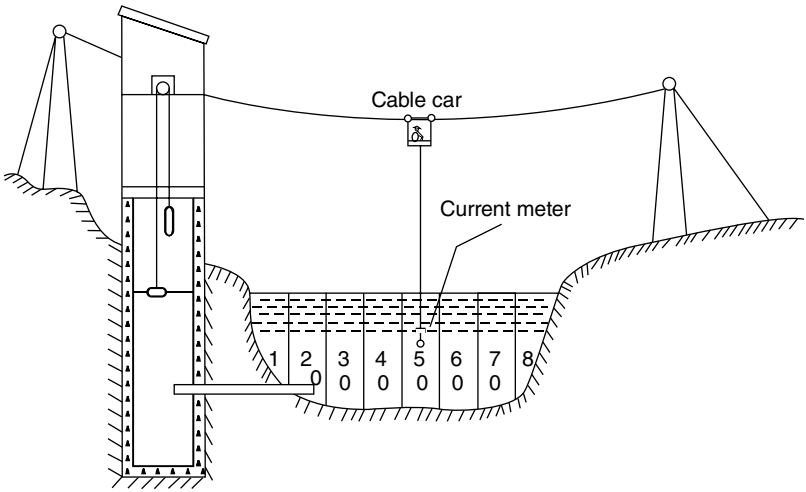


Fig. 15.8 *Actual Stream Gauging Station*

The run-off through the cross section of the river shown in Figure 15.8 is given by

$$Q = A_1V_1 + A_2V_2 + A_3V_3 + A_4V_4 + A_5V_5 + A_6V_6 + A_7V_7 + A_8V_8$$

where A_1, A_2, … A_8 are the areas of the sections considered and V_1, V_2, … V_8 are the mean velocities of water flowing through the cross-sectional areas A_1, A_2, … A_8, respectively.

Along with the annual total and average rates of flow, the maximum and minimum rates must also be known. Maximum flood flow governs the size of the headworks and the dam to be built. If the failure of the dam would cause loss of life below the site, the dam would be designed for a flood occurring once in 1,000 yr or even 10,000 yr. A dam on a remote river might be designed to hold floods occurring once in 10–20 yr. If a dam is built to resist the maximum possible flood of all time, the high cost would make the construction economically unattractive.

The minimum flow and the part that can be stored fix the amount of primary power that will be available in a power plant. The part of the capacity of a hydraulic plant available at all times when energy is needed is classed as primary power. Average flow must be known to determine the maximum marketable energy and revenue the projected plant may be expected to yield.

The maximum rate of flow through the river is either calculated by using empirical formulae or energy curves. Flood marks also used to give the maximum rate of flow.

The following empirical formulae are useful for finding maximum discharge:

(a) *Dicken's formula*

$$Q = \mathrm{CA}^{0.75}$$

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where C = 11.37 for the Northern India, 13.77–19.28 for the Central India and 22.04 for the Western India.

(b) *Inglis formula*

$$Q = \frac{123.2\text{A}}{\sqrt{\text{A} + 10.36}}$$

This is used for all types of catchment areas.

(c) *Ryves' formula*

This formula is used only in Southern India

$$Q = C.(A)^{2/3}$$

C = 6.74 for areas within 24 km from sea coast
= 8.45 for areas with 24–161 km from sea coast
= 10.1 for hilly areas.

15.4 HYDROGRAPHS

A hydrograph is a graphical representation that represents variation of discharge or flow with time. It is plotted with flow as ordinate and time intervals as abscissa as shown in Figure 15.9. A hydrograph may be plotted for several weeks or months. It indicates the power available from the water stream at different times of the day, week or year. It also gives an idea of the extreme conditions of flow. Hydrograph of stream of a river will depend on the characteristics of the catchment area and the precipitation over the catchment. The graph also gives a picture of flood flow of rivers.

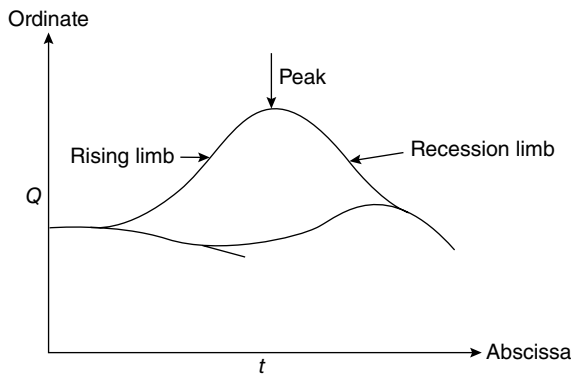


Fig. 15.9 *Hydrograph with Rising and Recession Limb*

Hydrograph is defined as a graph showing discharge (run-off) of flowing water with respect to time for a specified time. The time period for discharge hydrograph may be hour, day, week or month. The discharge may be cubic metre/second, square kilometre centimetre/hour or day–second–metre. Discharge hydrographs are known as flood or run-off hydrographs. Each hydrograph has its own reference to a particular river site. Figure 15.10 shows a typical hydrograph.

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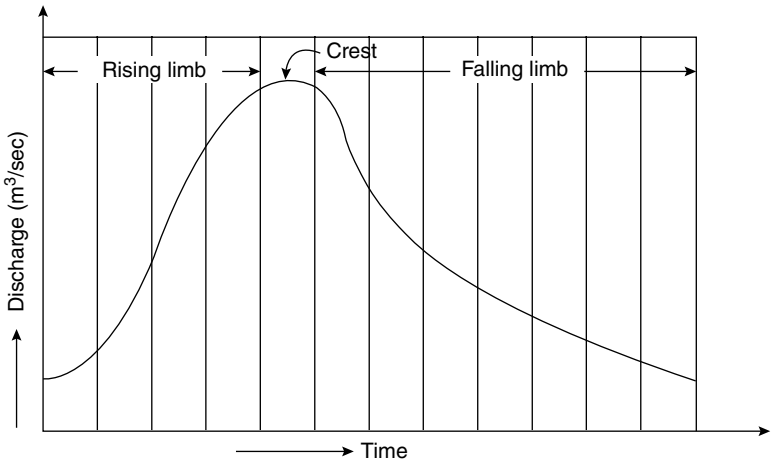


Fig. 15.10 *Hydrograph or Discharge Curve*

The following information is available from a hydrograph:

(i) Rate of flow at any instant during the duration period

(ii) Total volume of flow up to that instant as the area under the hydrograph indicates the volume of water during that duration

(iii) The mean annual run-off or mean run-off for each month of the year

(iv) The maximum and minimum run-off or mean run-off for each month

(v) The maximum rate of run-off during the floods and duration and frequency of the flood

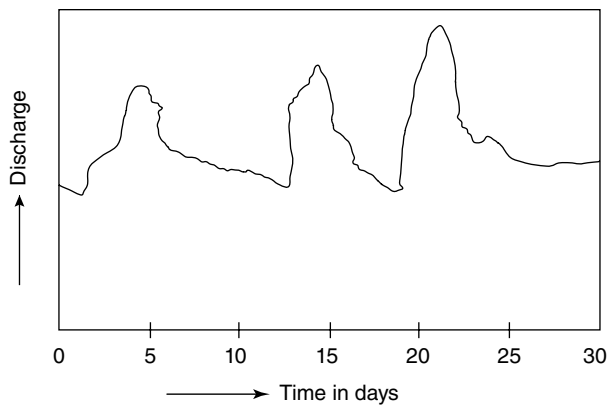


Fig. 15.11 *Hydrograph*

Figure 15.11 shows a hydrograph for a specific month. It can be noted from the figure that flood occurred thrice during this month.

The hydrograph can be drawn taking day, month or year as time axis as shown in Figure 15.12. The hydrograph on the basis of day gives an idea about the flood period during this month. The hydrograph on the basis of month gives an idea about the dry period of the year and also decides the period during which the water should be collected and to be used during the dry period of the year. The year-wise data provide information concerning the lean or draught year, which is very essential for deciding the location and size of the hydel power plant.

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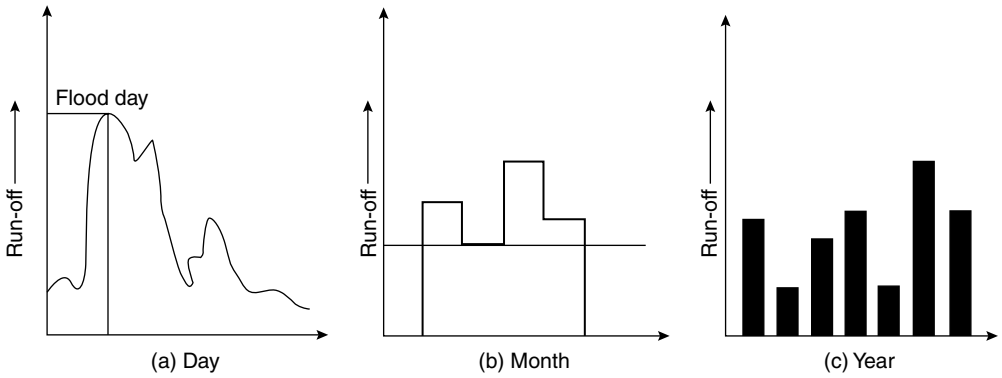


Fig. 15.12 *Different Forms of Hydrograph*

15.4.1 The Unit Hydrograph

If the rainfall distribution in the storms is similar with respect to time and area, the ordinate of each hydrograph will be proportional to the volume of run-off. The unit hydrograph is a hydrograph with a volume of 1 inch of run-off resulting from a rainfall of specified duration and a real pattern. The run-off hydrographs from two storms (rainfall) would be same if two identical rainstorms could occur covering a drainage basin with identical conditions before the rain[(1)] (Sherman, 1932). According to Sherman, all hydrographs have the same time base resulting from rainfalls of a given duration. Based on the above theory, the hydrograph for any other similar rainfall of the same duration can be constructed by multiplying the ordinates of the unit hydrograph by the storm run-off. Unit hydrograph is helpful in estimating run-off from a basin for a storm of given duration. It is also helpful in predicting the expected flood flow from a catchment if rainfall intensity in the catchment area is known.

Hydrographs of various rainfalls will be similar in shape. The ordinates are proportional to the run-off volumes within a fixed duration and similar rate and a real distribution of rainfall. However, the occurrence of identical rainfalls is very rare. The rainfalls may vary in duration, amount and area distribution. The effect of storage basin also varies. Therefore, unit hydrographs of very large floods differ somewhat from those of small rainfalls.

Construction of Unit Hydrograph

A unit hydrograph can be constructed from a hydrograph of the actual run-off when there is uniform rainfall intensity and uniform areal distribution.

The following steps are used to construct the unit hydrograph:

(i) The total volume of run-off (area under ABC) is first measured from the actual hydrograph.

(ii) The ordinates of the unit hydrograph are found by dividing the ordinates of actual run-off by the total volume of run-off in centimetre over the hydrograph for the basin.

(iii) The effective duration of run-off-producing rain for which the unit hydrograph is applicable is determined by a study of the rainfall records.

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The construction of unit hydrograph from the actual hydrograph of run-off is shown in Fig . 15.13.

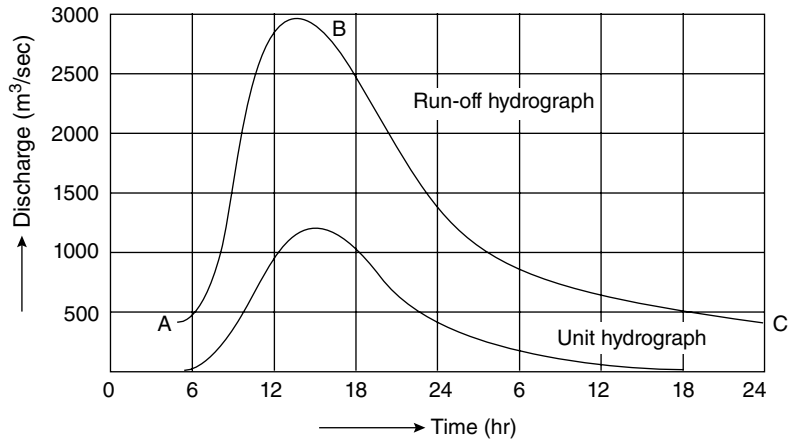


Fig. 15.13 *Derivation of Unit Hydrograph*

Use of Unit Hydrograph

The use of unit hydrograph is shown in Figure 15.14. The ordinates of the unit hydrograph multiplied by the estimated volume of direct run-off in centimetre of water over the basin gives the ordinates of the actual hydrograph.

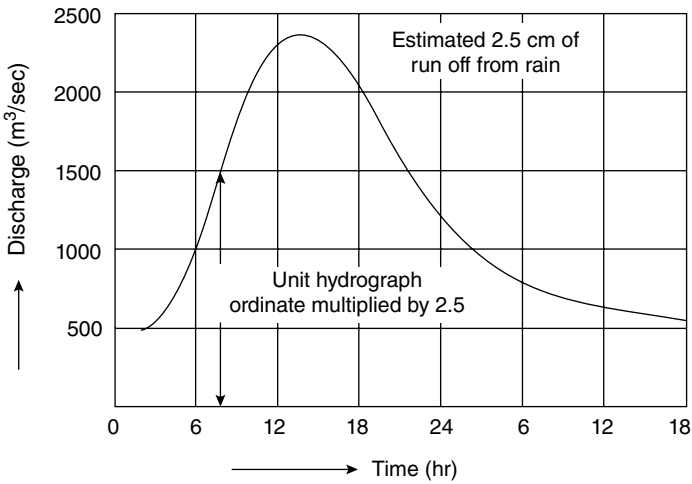


Fig. 15.14 *Use of Unit Hydrograph*

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The number of unit hydrographs for a given basin is theoretically infinite. This is because there may be one unit hydrograph for each duration and distribution pattern of rainfall in the basin. In practice, only a limited number of unit hydrographs are used for a given basin. While constructing the unit hydrograph, the variations in rainfall distribution within the basin area is generally neglected for small basins, but the variations over large areas cannot be ignored as variations are large. Hence, it is not advisable to use unit hydrograph method for basins over 5,000 km^2 area.

The unit hydrographs can be applied successfully to basin areas as large as 25,000 km^2, provided distribution patterns are classified into different types and unit hydrographs are developed for each type. It is always preferable to divide the large basin into sub-areas, utilize hydrographs for each sub-area independently and combine the resulting hydrographs.

Limitations to the use of unit hydrographs

(i) As similar rainfall distribution over a large area from storm to storm is highly unlikely, unit hydrographs are used to areas under 5,000 km^2.

(ii) Unit hydrograph method is not used to odd-shaped basins, long and narrow, as the rainfall distribution is uneven.

15.5 FLOW DURATION AND MASS DURATION CURVES

15.5.1 Flow Duration Curve

The run-off data for a given time can be represented by flow duration curve. The flow duration curve is drawn with magnitudes of run-off as ordinate against the corresponding percentage of time as abscissa. If the magnitude on the ordinate is the potential power contained in the stream flow, then the curve is known as 'power duration curve'. This curve is a very useful tool in the analysis of water power generation as it relates flows and the length of time during which they are available.

The flow duration curve is drawn with the help of hydrograph from the available run-off data. The following example explains the method used for drawing the flow duration curve.

Run-off data			
Month	**Discharge (m^3/sec)**	**Month**	**Discharge (m^3/sec)**
1	600	7	2,200
2	400	8	2,800
3	200	9	1,600
4	100	10	600
5	200	11	400
6	800	12	300

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The hydrograph can be drawn as shown in Figure 15.15 for the given run-off data.

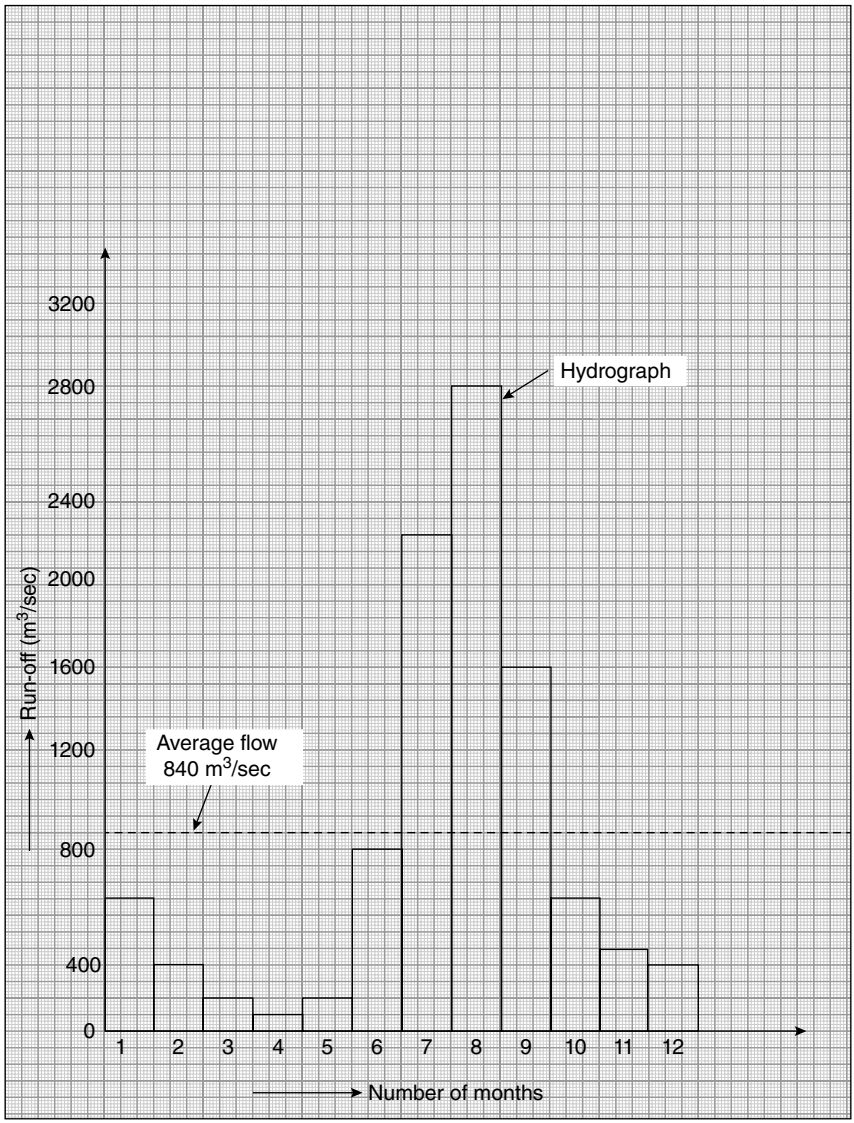


Fig. 15.15 *Hydrograph*

To obtain the flow duration curve from the hydrograph, find out the length of time during which certain flows are available either from run-off data or from hydrograph as given below:

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S. no.	Discharge (m^3/sec)	Time available (%)	Total number of months available
	(a)	(b)	c = × 100
1.	100 (and more)	12	100
2.	200 (and more)	11	91.6
3.	300 (and more)	9	76
4.	400 (and more)	8	66.6
5.	600 (and more)	6	50
6.	800 (and more)	4	33.3
7.	1,600 (and more)	3	25
8.	2,200 (and more)	2	16.6
9.	2,800 (and more)	1	8.3

Now the flow duration curve is drawn taking 100 per cent time on the *X*-axis and run-off on *Y*-axis as shown in Figure 15.16.

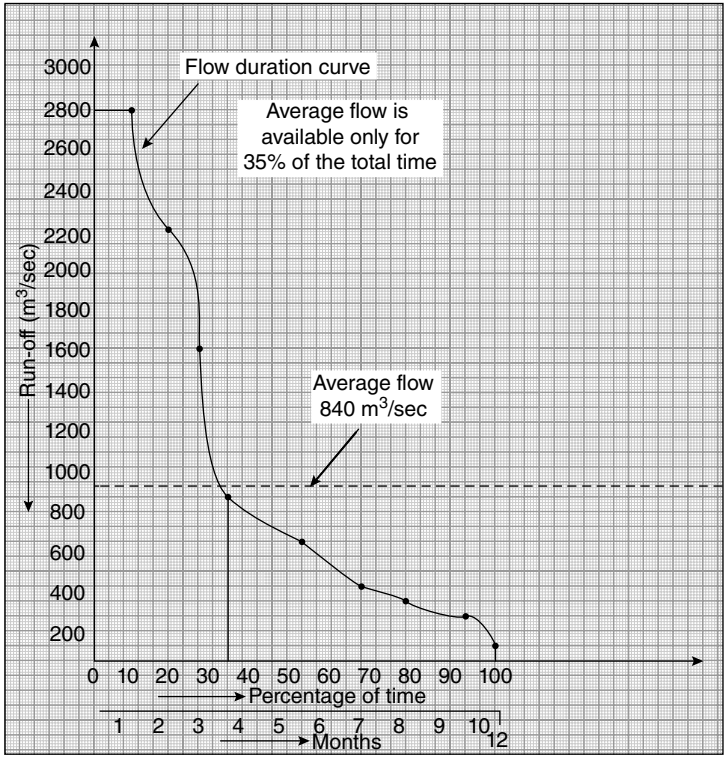


Fig. 15.16 *Flow Duration Curve*

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If the discharge head is known, the possible power developed from the water in kilo-watts can be determined using the following equation:

$$\text{Power in kW} = \frac{mgH}{1000} \times h \times \eta$$

where m = discharge, kilogram/second
H = head available, metre
η = combined efficiency of hydraulic prime mover and electric generator

The area under the flow duration curve gives the total quantity of run-off during that period.

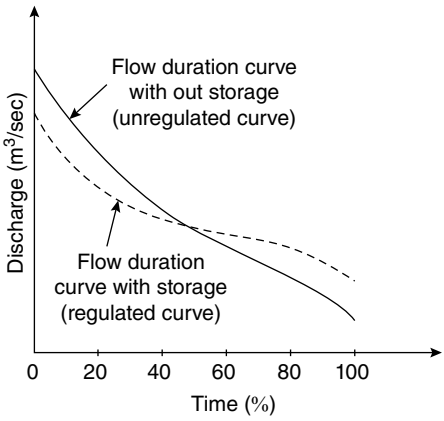


Fig. 15.17 *Effect of Storage on Flow Duration Curve*

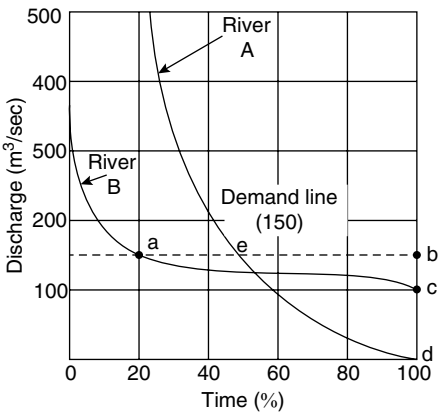


Fig. 15.18 *Comparison of Flow Duration Curves for Two Streams*

Effect of Storage on Flow Duration Curve

Storage must be provided at the power site when a large amount of natural flow is available during rainy season. The water can be stored during off-peak periods and can be made available during peak periods. Storage collects the natural flow of the stream during low-load condition (lower than the inflow) and supplies with a uniform rate if the load is constant. Storage also reduces the average flow from water surface area due to increased evaporation losses. Hence, the effective storage is the amount of water accumulated after adjusting the evaporation and other losses. The effect of storage on the flow duration is shown in Figure 15.17. It can be observed that storage facility lowers the upper portion of the curve and raises the lower portion, meaning more amount of discharge is available all the time.

With no evaporation and leakage losses, the area below the unregulated curve and above the regulated curve should be equal to the area below the regulated curve and above the unregulated curve.

In run-off river power plants without storage facilities, the unregulated flow duration curve should be used and firm power is assumed on the basis of the flow available 90 to 97 per cent of the total time. The flow duration curve should always be developed by considering longest possible or available stream flow record, to avoid undue influence by abnormally wet or dry periods. It should also be modified to correct for upstream reservoir storage.

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Uses of Flow Duration Curve

Flow duration curves are most useful for preliminary studies and for comparisons between streams. To understand this, consider the flow duration curve for river A and flow duration curve for river B as shown in Figure 15.18. The flow duration curve of river A necessitates proper storage to provide water during periods of low natural flow. River B provides minimum 100 m^3/sec throughout the year for direct use. Storage would be required on both streams to meet the constant demand of 150 m^3/sec, but the storage volume required for river B (a, b, c) is much less than that for river A (e, b, d). River B produces considerably more run-off than river A. If adequate yield facilities are provided, river B can provide a much higher yield than river A.

Limitations

The main drawbacks of the flow duration curve are as follows:

(i) It does not present the flows in natural sequence of occurrence.

(ii) It is also not possible to ascertain from flow duration curve whether the lowest flows occurred in consecutive periods or were scattered throughout the considered period.

15.5.2 Mass Curve and Storage

The graph of the cumulative values of a water quantity (run-off) against time is known as 'mass curve'. A mass curve gives the area under the hydrograph from one time to another, representing the total volume of run-off in month–metre–second during a given period. The mass curve is used to compute the capacity of the reservoir for a hydro-site by plotting cumulative volume of flow as ordinate and time (days, weeks and months) as abscissa. The abscissa of each point on mass curve in Figure 15.19 represents the total time from the beginning of the period and the ordinate of each point represents the total quantity of water available during this period.

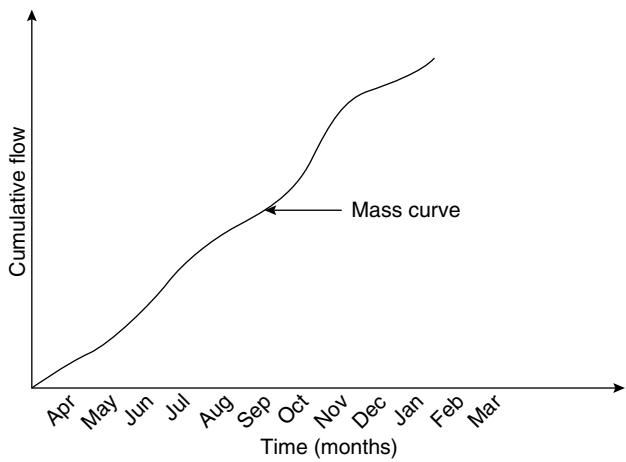


Fig. 15.19 *Mass Curve*

If V = the volume of run-off

Q_t = the discharge in cubic meter/second as a function of time

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Mathematically, the flow mass curve is expressed as follows:

$$V = \int_{t_1}^{t_2} Q_t \, dt$$

The mass curve of the river whose monthly average run-off is given in the following table is drawn as shown in Figure 15.20.

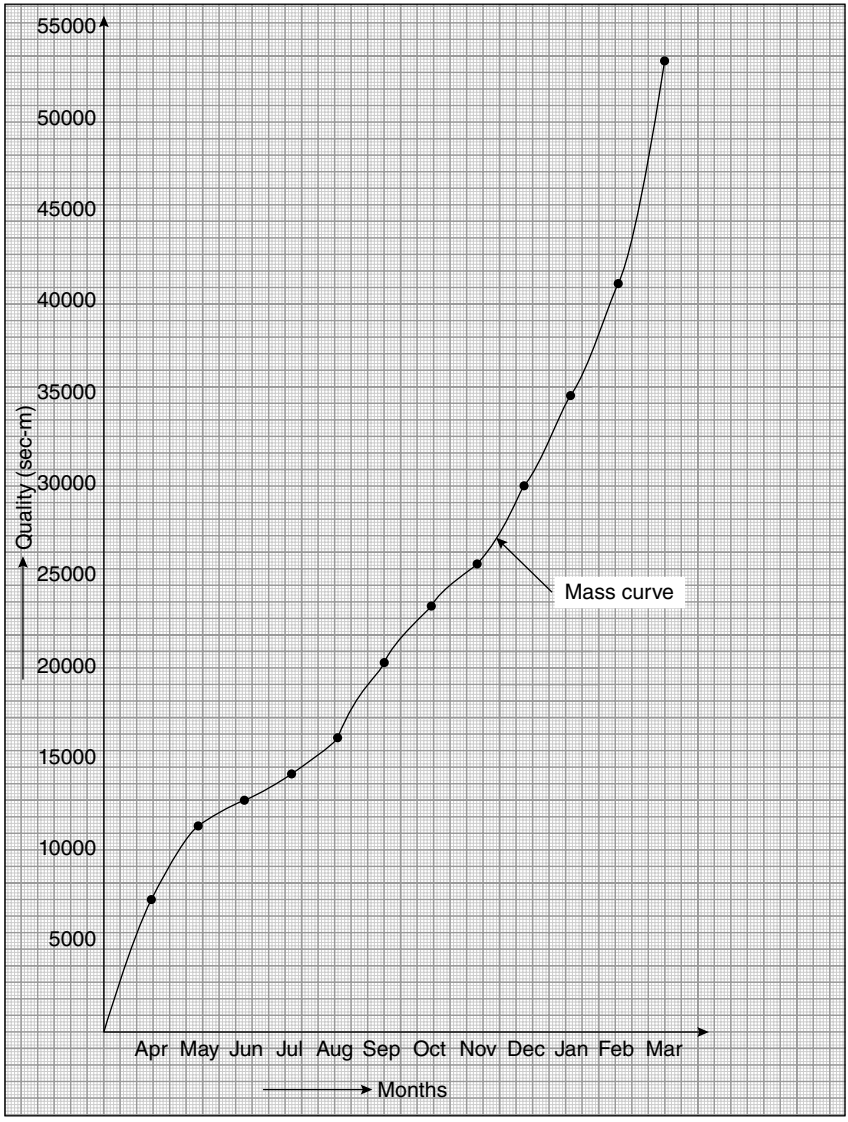


Fig. 15.20 *The Mass Curve of the River*

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Month	Second–metre mean-monthly run-off	Second–metre months cumulative
Apr	7,000	7,000
May	4,000	11,000
Jun	1,500	12,500
Jul	1,000	13,500
Aug	2,500	16,000
Sep	4,000	20,000
Oct	3,000	23,000
Nov	2,500	25,500
Dec	4,000	29,500
Jan	5,000	34,500
Feb	6,000	40,500
Mar	12,000	52,500

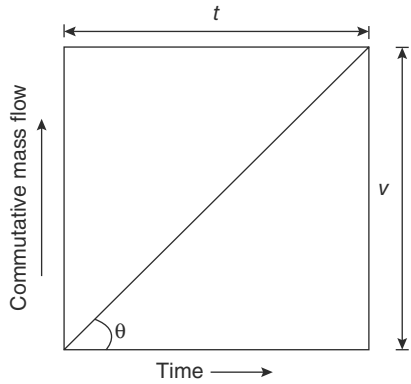


Fig. 15.21 *Mass Curve for Uniform Flow*

The mass curve plotted from mean monthly flows is correct information only at the beginning and end of each month, as the variation in flow during the month is not taken into account. For more accuracy, a summation of daily flows, instead of monthly flow, may be taken. However, such a mass curve plotted on the basis of daily flow is used in critical periods, and not often.

If the rainfall is uniform throughout the year, the mass curve for a river will be a straight line as shown in Figure 15.21 as the mean monthly flow remains constant throughout the year.

If V is total volume of the water collected during the time period t, then the slope of the curve, tan θ is given by

$$\tan\theta = V/t$$

and the rate of flow is

$$Q = \tan\theta = V/t$$

Advantages

(i) As the determination of required capacity for a reservoir is generally more complex, many problems related to storage capacity can be solved conveniently using a mass curve.

(ii) A negative inclination of mass curve means the amount of water flowing in the reservoir is less than the loss due to evaporation and seepage.

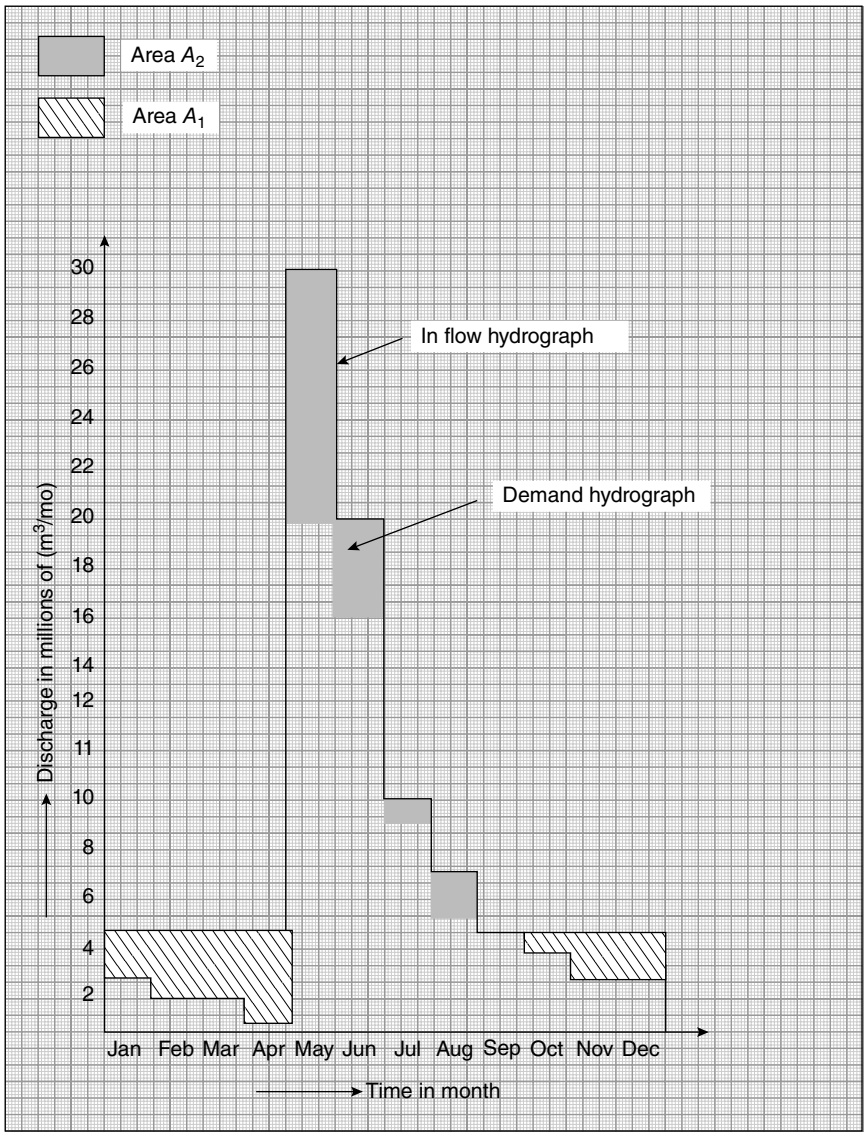
(iii) If the seasonal power demand is not constant, the flow demand line may be curved. In this case, flow mass curve represents the flow demand line.

The procedure to find out the reservoir capacity required when the inflow varies with respect to time and demand also varies with respect to time is explained using the following example.

It is evident from the figure that area A_1 gives the excess of the natural discharge over the requirements, whereas area A_2 represents the deficiency of the discharges. Therefore, it is clear

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that the reservoir must be filled during the period from May to August (both inclusive) when there is surplus of water in the river and must be emptied from October to April to compensate for the insufficiency of the natural discharge in these months.



From the given hydrographs, we can find out the cumulative inflow as well as cumulative demand for the time considered (i.e. 12 months).

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The cumulative values of the inflow and demand for 12 mo are as listed in the table below. From the cumulative values of inflow and demand, the mass curves for the inflow and demand can be drawn as shown in Figure 15.22.

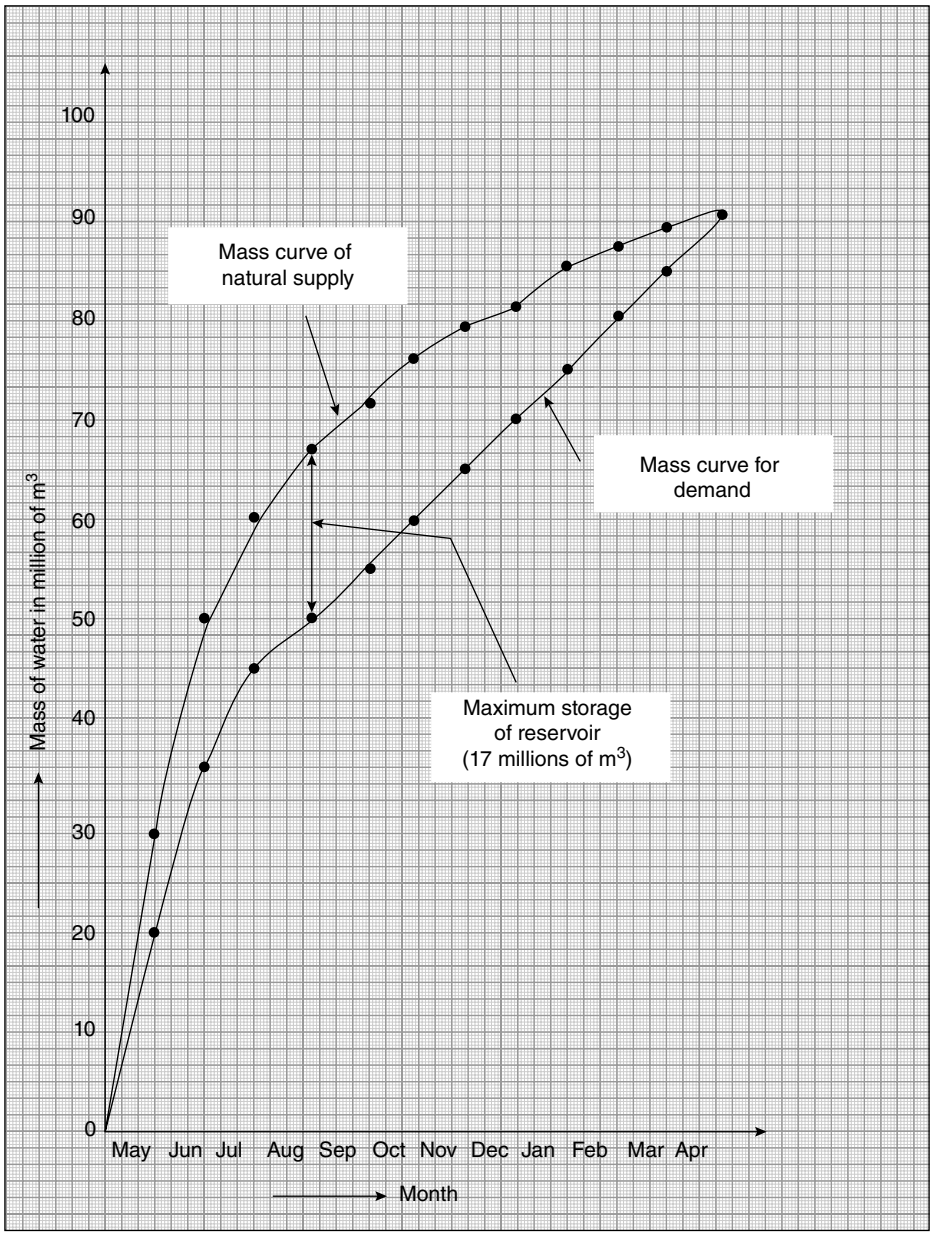


Fig. 15.22 *Mass Curves for Inflow and Demand*

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Month	Monthly discharge (millions m^2/mo)	Total mass water discharged (millions m^3)	Monthly demand (millions m^3/mo)	Total mass of water required (millions m^3)
May	30	30	20	20
Jun	20	50	16	36
Jul	10	60	9	45
Aug	7	67	5	50
Sep	5	72	5	55
Oct	4	76	5	60
Nov	3	79	5	65
Dec	3	82	5	70
Jan	3	85	5	75
Feb	2	87	5	80
Mar	2	89	5	85
Apr	1	90	5	90

From the mass curves shown in Figure 15.22, it is obvious that the difference between the amount that entered the reservoir and that left the reservoir must represent the amount retained in the reservoir. The maximum difference between the ordinates of two mass curves represents the maximum contents of the reservoir.

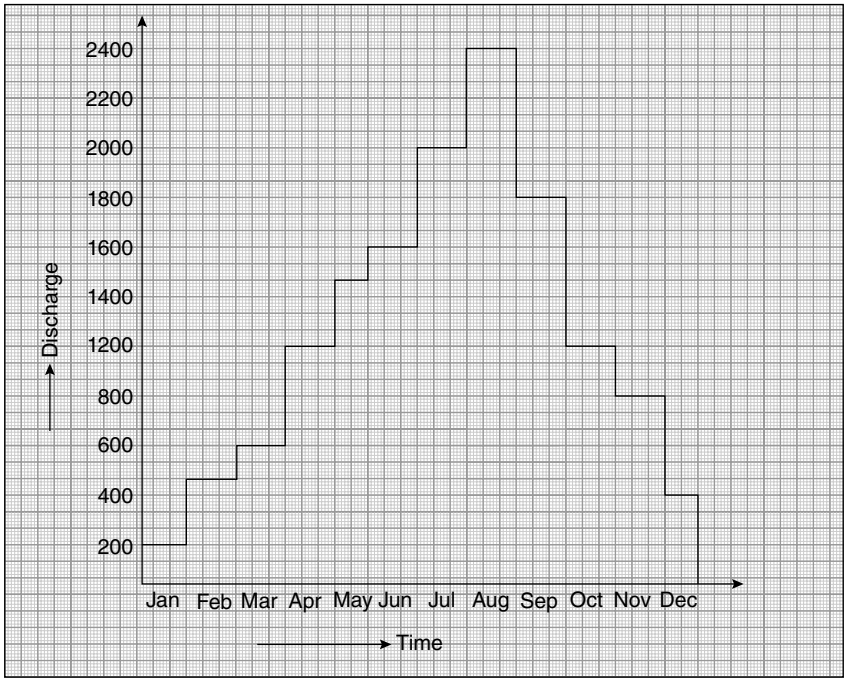
Example 15.1

The mean monthly discharge at a particular site is tabulated below. Draw the hydrograph and flow duration curve.

Month	Discharge (m^3/s)	Month	Discharge (m^3/s)
Jan	200	Jul	2,000
Feb	450	Aug	2,400
Mar	600	Sep	1,800
Apr	1,200	Oct	1,200
May	1,500	Nov	800
Jun	1,600	Dec	400

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Solution: The hydrograph for different discharges at different months is shown below:

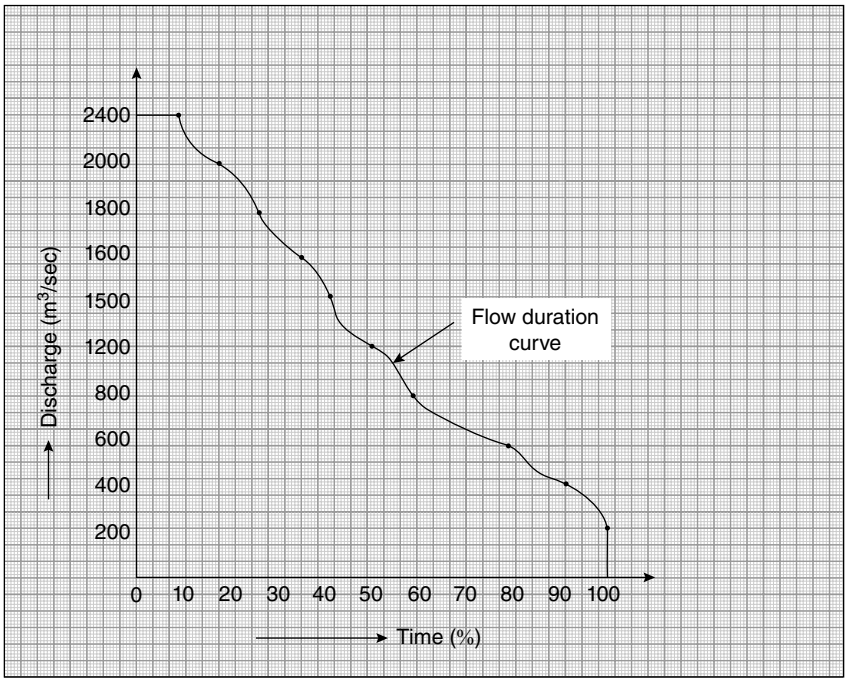


Flow duration curve can be drawn by knowing the percentage of time of different discharges. This is tabulated below:

Discharge (m^3/s)	Length of time (mo)	(%) of time
200	12	100.00
400	11	91.50
450	10	83.33
600	9	75.00
800	8	58.20
1,200	7	50.20
1,500	5	41.66
1,600	4	33.33
1,800	3	25.00
2,000	2	16.67
2,400	1	8.34

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The flow duration curve is plotted on the following graph sheet:



Example 15.2

A hydroelectric plant has a catchment area 50 mi^2 and run-off is 70 per cent. The average rainfall is 60 inch and the head is 400 m. Find the power available if the overall efficiency of the plant is 80 per cent.

Solution: Flow available $= 50 \times \frac{60}{12} \times \frac{0.70}{1.13}$

$= 154.87$ cusecs

$= 4.42$ cumecs

Note: 1 Cusecs = 1.13 mi^2, ft/yr.
One foot of water over an area of 1.13 mi^2 will give a discharge of 1 cusecs throughout the year assuming 100 per cent run-off.

Power available,

$$= WQH\eta_o \times 10^{-3} \text{ kW}$$

$$= 9{,}810 \times 4.42 \times 400 \times 0.8 \times 10^{-6} \text{ MW}$$

$$P = 13.875 \text{ MW}$$

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Example 15.3

(i) Define pondage factor.

(ii) Determine the capacity of a run-off river hydro power plant to be used as 10-hr peaking plant assuming daily flow in a river to be constant at 20 m^3/s. Calculate the pondage factor and pondage if the head of the plant is 12 m and overall efficiency is 80 per cent.

Solution: It is the ratio of total inflow hours in a given period to the total number of hours the plant is running during the same period.

(a) *Pondage factor*

$$\text{PF} = \frac{T_1}{T_2} = \frac{\text{Total number of hours in one day}}{\text{Total number of hours plant is running}}$$

$$= \frac{24}{10} = 2.4$$

(b) *Capacity of plant*

Capacity of the plant without pondage:

$$P = \text{WQH}\eta_o \times 10^{-3} \text{ kW}$$
$$= 9{,}810 \times 20 \times 12 \times 0.80 \times 10^{-6} \text{ MW}$$
$$= 1.8835 \text{ MW}$$

Capacity of the plant with pondage:

$$P_1 = \text{PF} \times P$$
$$= 2.4 \times 1.8835$$
$$= 4.52 \text{ MW}$$

Pondage = (24 − 10) = 14-hr

$$= 14 \times 60 \times 60 \times 20$$
$$= 1{,}008 \times 10^3 \text{ m}^3$$

Example 15.4

The catchment area of a hydel power plant is 2,500 km^2, with an average rainfall of 160 cm. The percolation and evaporation losses account for 19 per cent. If the available head is 150 m find the power developed by the plant. Assume that $\eta_g = 0.91$ and $\eta_t = 0.86$.

Solution: Amount of water available for power generation:

$$Q_a = A \times H \times (1 - y)$$
$$= 2{,}500 \times 10^6 \times \frac{160}{100}(1 - 0.19)$$
$$= 3.04 \times 10^{11} \text{ m}^3$$

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Flow rate of water:

$$Q = \frac{Q_a}{365 \times 24 \times 60 \times 60}$$

$$= \frac{3.04 \times 10^{11}}{365 \times 24 \times 60 \times 60}$$

$$= 9639.8 \text{ m}^3/\text{s}$$

Power developed:

$$= \text{WQH} \times \eta_t \times \eta_g \times 10^{-6} \text{ MW}$$

$$= 9{,}810 \times 9639.8 \times 0.86 \times 0.91 \times 10^{-6}$$

$$= 74 \text{ MW}$$

Example 15.5

The average daily stream flows for 7 day resulting from a heavy storm on a basin of 3,340 km^2 is tabulated below. Find out the total flow volume in day–sec–meter, km^2-m, cm and millions m^3.

Days	Mean daily flow (m^3/sec)
1	100
2	300
3	200
4	120
5	50
6	30
7	20

Solution: Total flow volume for 7 day

$$= 24 \times 3{,}600 \ (100 + 300 + 200 + 120 + 50 + 30 + 20)$$

$$= 70.848 \times 10^6 \text{ m}^3$$

$$= 70.848 \text{ million m}^3$$

$$= 820 \text{ day–sec–m}$$

$$= 2.121 \text{ cm}$$

$$= 70.848 \text{ km}^2\text{-m} \ (1 \text{ km}^2\text{-m} = 1 \text{ million m}^3)$$

Example 15.6

The mean monthly discharge at a particular site is given below. Draw the hydrograph and flow duration curve.

Month	Discharge (m^3/sec)
Jan	2,000
Feb	4,500
Mar	6,000
Apr	12,000
May	15,000
Jun	16,000
Jul	20,000
Aug	24,000
Sep	18,000

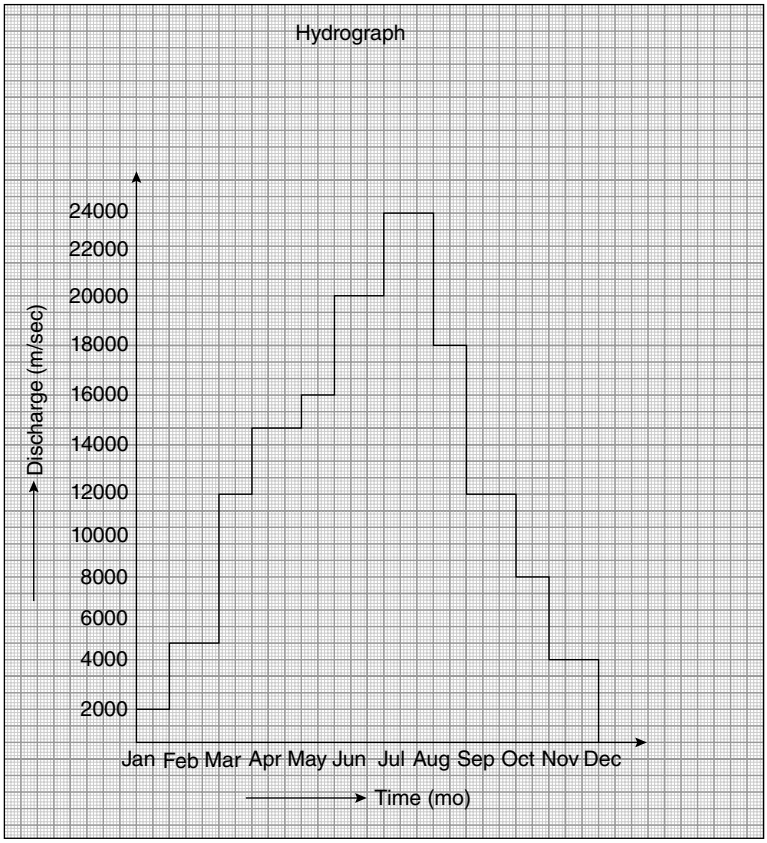
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Month	Discharge (m^3/sec)
Oct	12,000
Nov	8,000
Dec	4,000

Solution: The hydrograph is drawn based on the given discharge in the following graph sheet:



To draw the flow duration curve, it is essential to determine the percentage of time the given flow occurs. The following table is formed to draw the flow duration curve.

Discharge (m^3/s)	Length of time (mo)	Time (%)
2,000	12	100
4,000	11	91.5
4,500	10	83.33

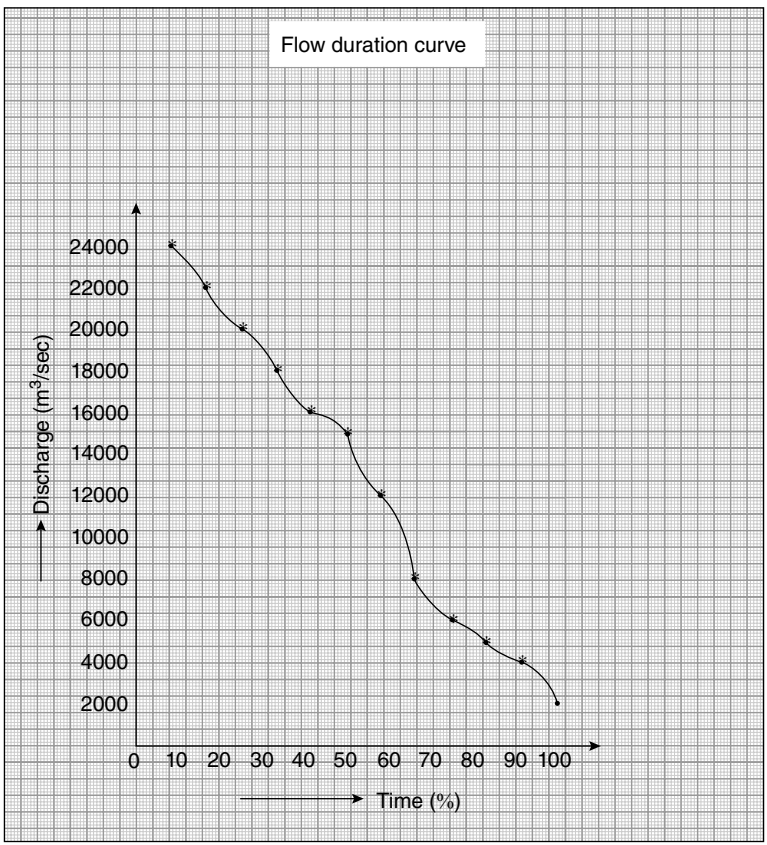
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Discharge (m^3/s)	Length of time (mo)	Time (%)
6,000	9	75.0
8,000	8	66.7
12,000	7	58.2
15,000	6	50.0
16,000	5	41.66
18,000	4	33.33
20,000	3	25.0
22,000	2	16.67
24,000	1	8.34

Using the data given the above table, the flow duration curve is drawn as shown in the following graph sheet.



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Example 15.7

Draw the mass curve for the run-off data given below and determine the capacity of the reservoir.

Month	Discharge (m^3/sec)
Jan	2,000
Feb	1,500
Mar	1,000
Apr	1,050
May	1,050
Jun	1,300
Jul	3,200
Aug	3,500
Sep	1,700
Oct	1,000
Nov	1,100
Dec	2,100

(Bangalore University, Oct-1999)

Solution: To draw the mass curve, the cumulative flow in cubic meter/second is first calculated for different months as shown in the following table. Based on the available data, the mass flow curve is drawn as shown in the graph sheet.

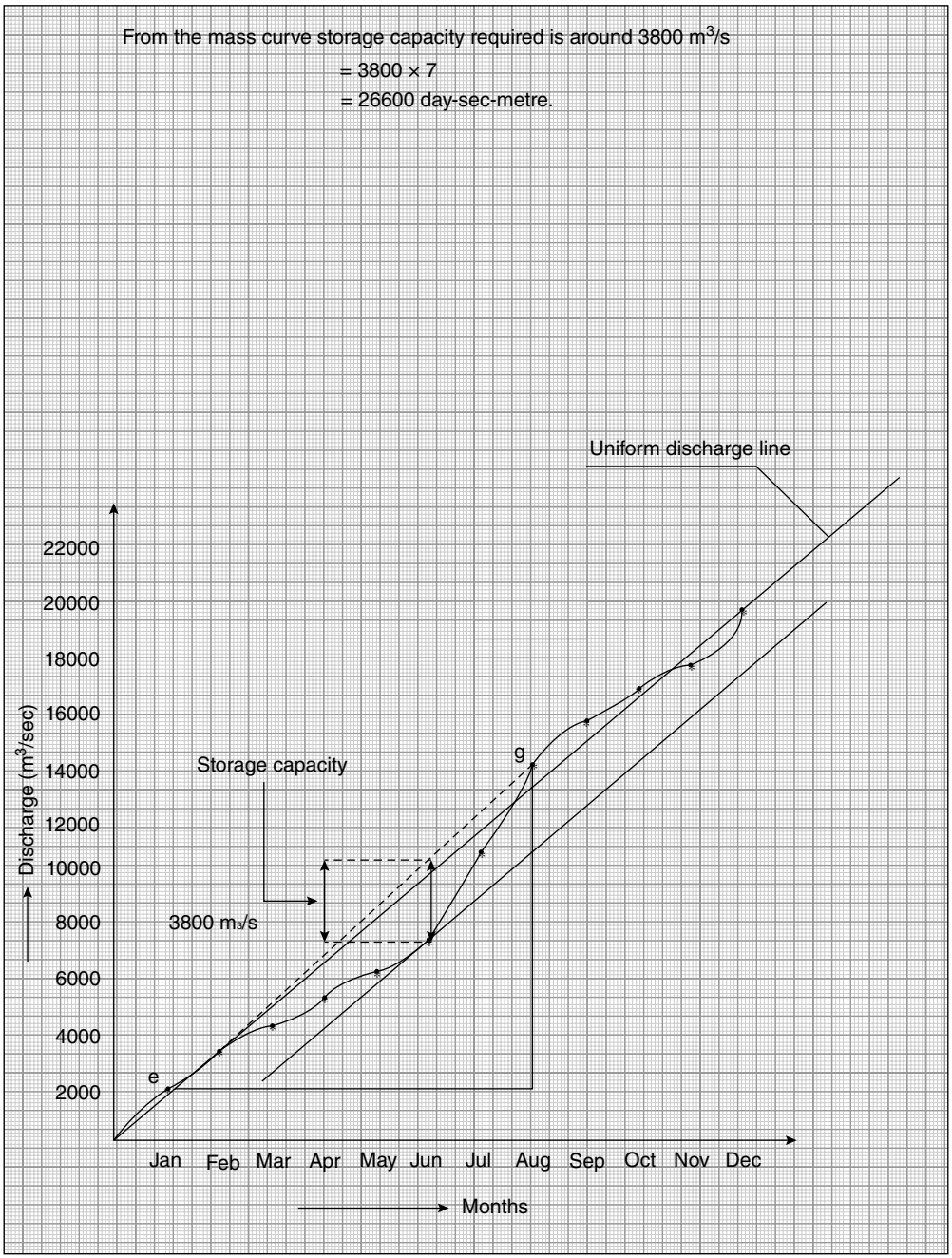
Month	Mean monthly run-off (m^3/sec)	Cumulative flow (m^3/sec)
Jan	2,000	2,000
Feb	1,500	3,500
Mar	1,000	4,500
Apr	1,050	5,550
May	1,050	6,600
Jun	1,300	7,900
Jul	3,200	11,100
Aug	3,500	14,600
Sep	1,700	16,300
Oct	1,000	17,300
Nov	1,100	18,400
Dec	2,100	20,500

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From the mass curve storage capacity required is around 3800 m^3/s

= 3800 × 7

= 26600 day-sec-metre.



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Example 15.8

The run-off data of a river at a particular site is tabulated below.

Month	Discharge (millions m^3/mo)
Jan	80
Feb	50
Mar	40
Apr	20
Mar	0
Jun	100
Jul	150
Aug	200
Sep	220
Oct	120
Nov	100
Dec	80

(i) Draw the hydrograph and find out the mean flow.

(ii) Draw flow duration curve.

(iii) Find out the power in MW available at mean flow if the head available is 100 m and overall efficiency of generation is 85 per cent.

Take each month of 30 day.

Solution: Mean discharge for the given data:

$= (80 + 50 + 40 + 20 + 0 + 100 + 150 + 200 + 220 + 120 + 100 + 80)/12$

$= 96.67$ million m^3/mo

Discharge (millions m^3/mo)	Total number of months during which flow is available	Time (%)
0	12	100
20	11	91.67
40	10	83.4
50	9	75

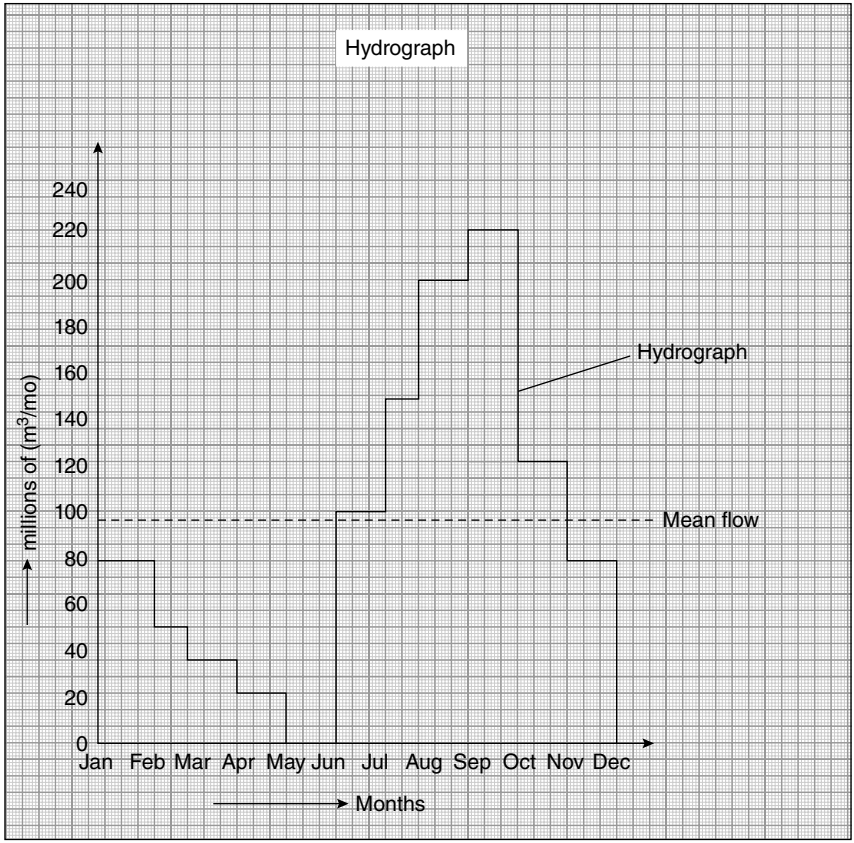
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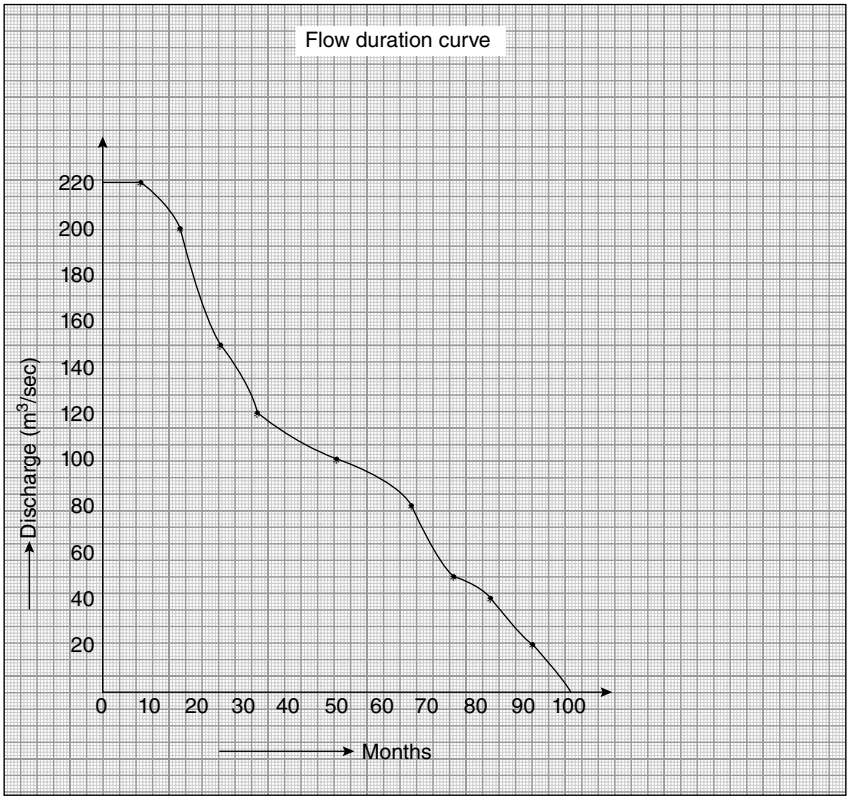
Discharge (millions m^3/mo)	Total number of months during which flow is available	Time (%)
80	8	66.6
100	7	58.4
120	6	50
150	4	33.4
200	3	25
220	2	16.66
1	0	8.33

(i) The following hydrograph is drawn based on the data tabulated above:



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(ii) The flow duration curve is drawn based on the data tabulated above:



(iii) Average MW energy available is

$$= \text{WQH} \times \eta_g$$

$$= \frac{9810 \times 96.67 \times 10^6 \times 100}{30 \times 24 \times 3600} \times 0.8$$

$$= 29.3 \text{ MW}$$

Example 15.9

The run-off data of a river at a particular site are tabulated below:

Month	Discharge
Jan	45
Feb	30
Mar	20

(*Continued*)

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(*Continued*)

Month	Discharge
Apr	15
May	0
Jun	50
Jul	75
Aug	100
Sep	110
Oct	70
Nov	60
Dec	40

Given that discharge in millions of cubic meter and month = 30 day,

(i) Draw a hydrograph and flow duration curve.

(ii) If the head available is 80 m and overall efficiency of generation is 85 per cent. Find the power in mega-watt available at mean flow of water (VTU January 2004).

Solution: Mean discharge for the given data is

$$= \frac{45+30+20+15+0+50+75+100+110+70+60+40}{12}$$

$$= 51.25 \text{ million m}^3\text{/month}$$

(i) Flow duration curve

To draw the flow duration curve, the following table is constructed.

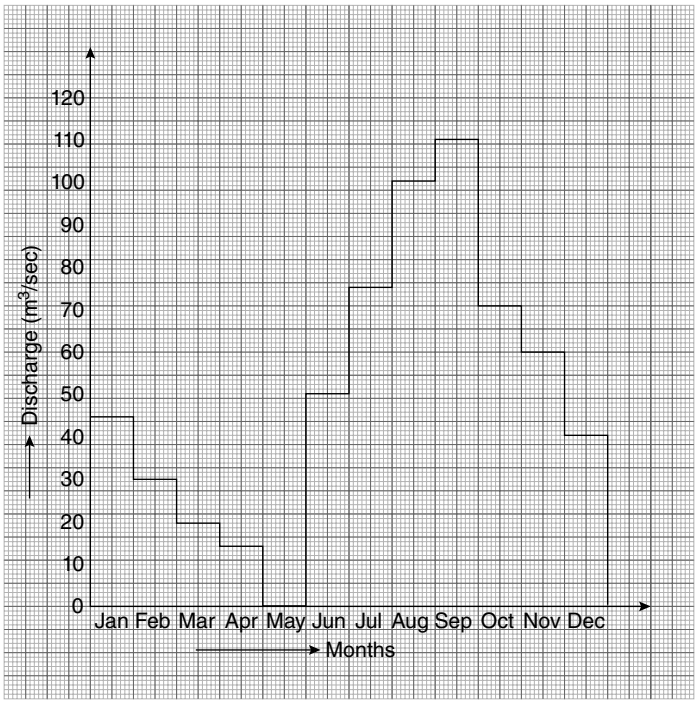
Discharge	Total number of months which flow is available	Time available (%)
(a)	(b)	$c = (\frac{b}{12} \times 100)$
0	12	100
15	11	91.7
20	10	83.4
30	9	75.0
40	8	66.7
45	7	58.4

(*Continued*)

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Discharge	Total number of months which flow is available	Time available (%)
50	6	50.0
60	5	41.7
70	4	33.4
75	3	25
100	2	16.7
110	1	8.4



(ii) Average mega-watt energy available is

$$= \eta_o \times \text{WQH} \times \frac{1}{1000}$$

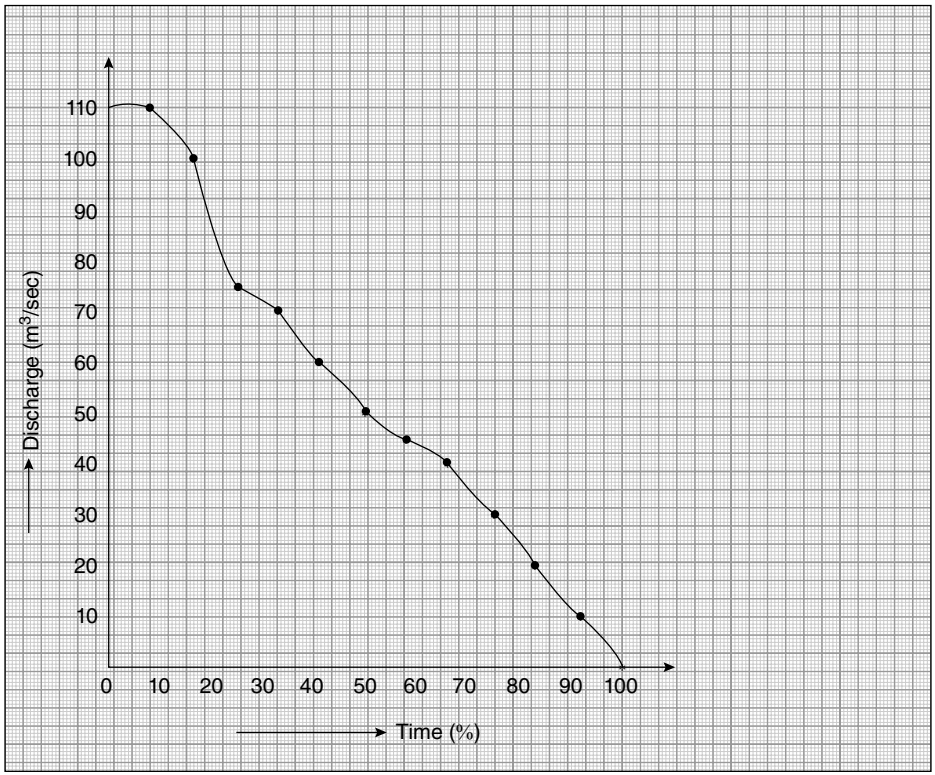
$$\text{Discharge, } Q = \frac{51.25 \times 10^6}{30 \times 24 \times 3600} = 19.773 \text{ m}^3/\text{sec}$$

$$\therefore P = 0.85 \times 9.81 \times 19.773 \times \frac{80}{1000}$$

$$= 13.19 \text{ MW}$$

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The flow duration curve is shown in the following graph:



Example 15.10

What is the volume of rainfall in day–second–metre if 6.5 cm rainfall occurs over an area of 2,400 km^2? Also find out in terms of km^2-m.

Solution: Total rainfall

$$= 2{,}400 \times 10^6 \times \frac{6.5}{100}\ m^3$$

$$= 156 \times 10^6\ m^3$$

Therefore, rainfall in day–second–meters $= \dfrac{156 \times 10^6}{86400} = 1805.56$ day–sec–m

Rain fall in square kilometre-metre $= \dfrac{156 \times 10^6}{10^6} = 156\ km^2$-m

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Example 15.11

It is proposed to utilize monsoon stream by constructing a dam across it. The stream discharge during the monsoon season of 4 month is 20 m^3/sec and for the remaining year, it is taken as 2.5 m^3/sec.

(i) Find the minimum capacity required for the reservoir on the upstream side of the dam.

(ii) If the head loss in the pipe is 2 per cent of the actual head and overall efficiency of the generation is 92, find the output of the generating station.

Take the mean level of the water in the reservoir above the tail race level 90 m. Take monsoon period from June 1 to September 30 and take the year of 365 day.

Solution: The number of days during which the discharge of 20 m^3/sec is available

$$= 30 + 31 + 31 + 30 = 122 \text{ day}$$

and the number of days during which the discharge of 2.5 m^3/sec is available

$$= 365 - 122 = 243 \text{ day}$$

Total flow during the year $= 20 \times 3{,}600 \times 24 \times 122 + 2.5 \times 3{,}600 \times 24 \times 243$

$$= 263.304 \times 106 \text{ m}^3$$

$$\text{Average discharge} = \frac{263.304 \times 10^6}{3600 \times 24 \times 365} = 8.35 \text{ m}^3/\text{sec}$$

The difference between the maximum and average discharge

$$= 20 - 8.35 = 11.65 \text{ m}^3/\text{sec}$$

Reservoir capacity to store the excess water

$$= 11.65 \times 3{,}600 \times 24 \times 122 \text{ m}^3$$

$$= \frac{11.65 \times 3600 \times 24 \times 122}{86400} = 1421.3 \text{ day–sec–m}$$

$$\text{Average kilo-watt generated} = \frac{mgh_n\eta}{1000}$$

$$= \frac{8.35 \times 1000 \times 9.81 \times 90 \times 0.98}{1000} \times 0.92$$

$$= 6646.8 \text{ kW} \cong 6.65 \text{ MW}$$

Example 15.12

A lake behind a dam has a capacity of 30,000 km^2-m approximately. For how many days would this water supply be sufficient to a city of 10^6 population if daily requirement per person is 500 L.

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Solution: Per day requirement = 500×10^6 l = 500×10^3 m^3

Available water in the dam = $30{,}000 \times 10^6$ m^3

Therefore, the number of days water supplied = $\dfrac{30{,}000 \times 10^6}{500 \times 10^3} = 60{,}000$ day

Example 15.13

A storage-type hydroelectric power plant having a catchment area of 200 km^2 and annual average rainfall of 100 cm is used to generate power. A total of 80 per cent run-off is available for power generation. The mean head available is 80 m. Assuming an overall efficiency of generation to 75 per cent, find the capacity of the plant. If the average period of working the power plant is 18 hr/day, find the energy generated in kilo-watt hour/year.

Solution: Total water available for power generation per year

$$= 200 \times 10^6 \times 0.8 \times 1 \text{ m}^3/\text{yr}$$

$$= 160 \times 10^6 \text{ m}^3/\text{yr}$$

$$= \frac{160 \times 10^6}{365 \times 24 \times 3600} = 5.07 \text{ m}^3/\text{sec}$$

Capacity of the plant in kilo-watt $= \dfrac{mgH}{1000} \times \eta_g$

$$= \frac{1000 \times 5.07 \times 9.81 \times 80}{1000} \times 0.75 = 2984.20 \text{ kW}$$

Energy generated per year = $2984.20 \times 18 \times 365$ kWh

$= 19.61 \times 10^6$ kWh

Example 15.14

The catchment area of a hydroelectric power plant is 2,500 km^2 and the annual average rainfall is 150 cm. The head drop available at a power house site is 120 m. Assuming the turbine efficiency to be 85 per cent and the generation efficiency to be 90 per cent, find the kilo-watt that can be developed from the hydroelectric plant. Take percolation and evaporation losses as 20 per cent and load factor as unity.

If the speed of the runner is to be maintained below 240 rpm, also suggest the type of prime mover to be used.

Solution: The quantity of water available for power generation per year

$$= 2{,}500 \times 10^6 \times (150/100)\,(1 - 0.2)\text{m}^3 \; (1 \text{ km}^2 = 10^6 \text{ m}^2)$$

$$= 3 \times 10^9 \text{ m}^3$$

Q (quantity of water available per second)

$$= \frac{3 \times 10^9}{365 \times 24 \times 3600} = 95.13 \text{ m}^2/\text{sec}$$

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Power developed in kilo-watt $= \frac{\text{mgH}}{1000} \times \eta_m \times \eta_g$

where η_m and η_g are mechanical and generator efficiencies, respectively.

Therefore, the power developed in kilo-watt $= \frac{1000 \times 95.13 \times 9.81 \times 120}{1000} \times 0.85 \times 0.9$

$$= 85.67 \times 10^3 \text{ kW}$$

Specific speed, $N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{240\sqrt{85.67 \times 10^3}}{120^{5/4}}$

$$= 176.87$$

Francis turbine can be used as a prime mover.

Example 15.15

A hydroelectric power plant having 100 km^2 reservoir area and 100 m head is used to generate power. The energy utilized by the consumers whose load is connected to power plant during a 5-hr period is 15×10^6 kWh. The overall generation efficiency is 75 per cent. Find the fall in the height of water in the reservoir after 5-hr period.

Solution: Assume the fall in the height in the reservoir level is hectometre.

Therefore, water used during 5-hr $= 100 \times 10^6 \times \text{H}$

$$Q \text{ (discharge/second)} = \frac{100 \times 10^6 \times H}{5 \times 3600}$$

But, $P = \frac{\text{mgH}}{1000} \times \eta$

$$= 1{,}000 \times \frac{100 \times 10^6 \times H}{5 \times 3600} \times 9.81 \times \frac{100}{1000} \times 0.75$$

$$= 4.0875 \times 10^6 \text{ H}$$

The energy produced during 5-hr period

$$= 5 \times 4.0875 \times 10^6 H = 20.44 \times 10^6 \text{ H}$$

$$20.44 \times 10^6 H = 15 \times 10^6$$

$$\therefore H = \frac{15}{20.44} = 0.734 \text{ m}$$

Example 15.16

A medium capacity storage-type hydroelectric power plant covers 1,200 km^2 area. The annual rainfall in catchment area is 160 cm. The head available at the power plant site is 380 m. Assuming 25 per cent of the rainfall is lost in evaporation and percolation, find the average

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power developed by the power plant and maximum demand. Take overall efficiency of the plant as 75 per cent and load factor 0.5.

Solution: Water available per year for power generation

$$= 1{,}200 \times 10^6 \times \frac{160}{100} \times (1 - 0.25) = 1.44 \times 10^9 \text{ m}^3/\text{yr}$$

Average flow per second,

$$= \frac{1.44 \times 10^9}{365 \times 24 \times 3600} = 45.66$$

Average power developed in kilo-watt,

$$= \frac{\text{mgH}}{1000} \times \eta$$

$$= \frac{1000 \times 45.66 \times 9.81 \times 380}{1000} \times 0.75$$

$$= 127658.51 \text{ kW} = 127.66 \text{ MW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}}$$

$$\therefore \text{ Maximum demand} = \frac{127.66}{0.5} = 255.32 \text{ MW}$$

Example 15.17

The average daily stream flows for 7 day resulting from a heavy storm on a basin of 3,500 km^2 is tabulated below. Find the total flow volume in day–sec–m km^2-m, cm and millions of m^3.

Days	1st	2nd	3rd	4th	5th	6th	7th
Mean daily flow (m^3/sec)	100	320	210	120	50	30	25

Solution: Total flow volume for 7 day

$$= 24 \times 3{,}600 \ (100 + 320 + 210 + 120 + 50 + 30 + 25)$$

$$= 24 \times 3{,}600 \times 855 \text{ m}^3$$

$$= \frac{24 \times 3600 \times 855}{10^6} \text{ million m}^3 = 73.87 \text{ million m}^3$$

$$= 73.87 \times 10^6/86400 = 854.98 \text{ day-s-m}$$

$$= \frac{24 \times 3600 \times 855}{3600 \times 10^4} \text{ cm} = 2.5 \text{ cm}$$

$$= 73.87 \text{ km}^2\text{-m} \ (1 \text{ km}^2\text{-m} = 1 \text{ million m}^3)$$

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Example 15.18

The catchment area of the dam used for hydroelectric station is 250 km^2. The annual rainfall is 150 cm. If 70 per cent of the water in the dam is used for power generation at a head of 60 m, find the capacity of the power plant in MW. Assume turbine efficiency of 90 per cent and generator efficiency of 95 per cent. Neglect all other losses.

Solution: Total water used for power generation

$$= (250 \times 10^6) \times 1.5 \times 0.7 \text{ m}^3 = 262.5 \times 10^6 \text{ m}^3$$

Water flow rate $q = \dfrac{262.5 \times 10^6}{365 \times 24 \times 3600} = 8.32 \text{ m}^3/\text{sec}$

Power $P = \dfrac{\text{mgH}}{1000} \times \eta_t \times \eta_g \times \text{MW}$

where η_t and η_g are turbine and generator efficiencies.

$$\therefore P = \frac{8.32 \times 1000 \times 9.81 \times 60}{1000} \times 0.9 \times 0.95 \times \frac{1}{1000} = 4.19 \text{ MW}$$

Example 15.19

A hydel plant is supplied from a reservoir of 5×10^6 m^3 capacity at a head of 75 m. Determine the number of electrical units produced (kWh) during the year if the load factor is 0.6 and overall efficiency of generation is 72 per cent.

Solution: The power capacity of the plant in kW is given as follows:

$$P = \frac{\text{mgH}}{1000} \times \eta_{\text{overall}}$$

$$= \frac{5 \times 10^6}{365 \times 24 \times 3600} \times \frac{1000 \times 9.81 \times 75}{1000} \times 0.72$$

$$= 83.99 \text{ kW}$$

Energy produced in kilo-watt hour $= P \times \text{load factor} \times (365 \times 24)$

$$= 83.99 \times 0.6 \times 365 \times 24 = 441450 \text{ kW.} = 441.45 \text{ MW}$$

Example 15.20

A hydel power station is supplied from a reservoir at a head of 40 m. If the area of the reservoir is 1.8 km^2 and generating 25 MW power, determine the rate at which the water level will fall in the reservoir. Take overall efficiency as 80 per cent.

Solution: The power generating capacity is given by

$$P = \frac{\text{mgH}}{1000} \times \eta_{\text{overall}} \times \frac{1}{1000} \text{ MW}$$

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$$\therefore\ 25 = \frac{q \times 1000 \times 9.81 \times 40}{1000} \times 0.8 \times \frac{1}{1000}$$

$$\therefore\ q = \frac{25 \times 1000}{9.81 \times 40 \times 0.8} = 79.64\ \text{m}^3/\text{sec.}$$

If x metre is the rate of fall of height of reservoir in metre per second, then

$$1.8 \times 10^6 \times x = 79.64 \times 3{,}600$$

$$x = \frac{79.64 \times 3600}{10^6 \times 1.8} = 0.159\ \text{m/hr}$$

Example 15.21

A hydel power station has to operate with a mean head of 30 m and supplied from a reservoir whose catchment area is 300 km^2 and average rainfall is 125 m/yr. If 70 per cent of the total rainfall can be used at a expected load factor of 50 per cent, calculate the power in kW for which the station can be designed. Assume head loss in the system is 8 per cent, mechanical η of the turbine is 90 per cent and generator efficiency is 95 per cent.

Solution: Water available during the year

$$= 300 \times 10^6 \times 1.25 \times (0.7) = 262.5 \times 10^6\ \text{m}^3$$

Water flow per second $Q = \dfrac{262.5 \times 10^6}{8760 \times 3600} = 8.32\ \text{m}^3/\text{sec} = 8.32 \times 1{,}000\ \text{kg/sec}$

Available head, $H = 30$ m
The power P in kilo-watt is given by

$$P = \frac{\text{Q}\rho\text{gH}}{1000} \times \eta_\text{o}$$

where η_o (overall efficiency) = $\eta_\text{h} \cdot \eta_\text{m} \cdot \eta_\text{g}$

$$\therefore \eta_\text{o} = 0.92 \times 0.9 \times 0.95 = 0.7865$$

$$\therefore\ P = \frac{8.32 \times 1000 \times 9.81 \times 30 \times 0.7865}{1000}$$

$$= 1925.80\ \text{kW}$$

With 80 per cent load factors,

$$\text{Generator capacity} = \frac{1925.80}{0.8} = 2407.25\ \text{kW}$$

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Example 15.22

The run-off data of a river at a particular site are tabulated below:

Month	Mean discharge (millions m^3/mo)	Month	Mean discharge (millions m^3/mo)
Jan	90	J	150
Feb	50	A	200
Mar	40	S	250
Apr	20	O	120
May	0	N	100
Jun	100	D	90

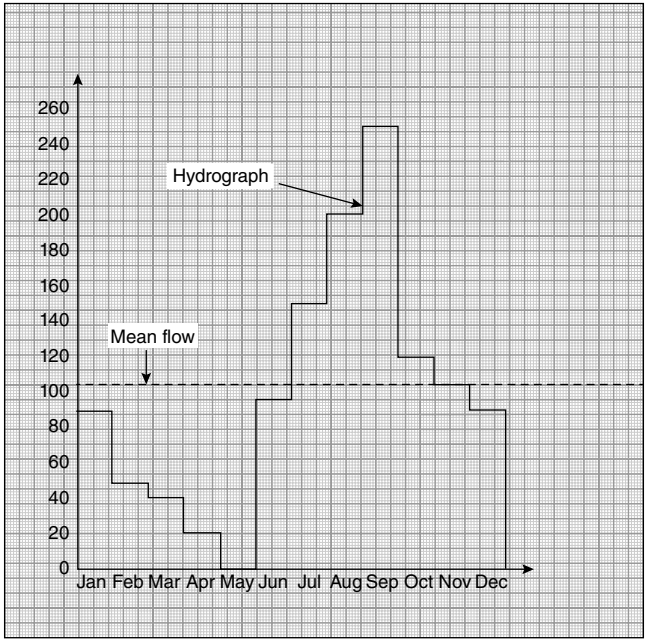
(i) Draw a hydrograph and find the mean flow.

(ii) Also draw the flow duration curve.

(iii) Find the power in megawatt available at mean flow if the head available is 100 m and overall efficiency of generation is 80 per cent.
Take each month of 30 day.

Solution:

(i) The hydrograph for the given data is drawn as shown in the figure below.



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The mean discharge for the given data:

$$= \frac{90+50+40+20+0+100+150+200+220+120+100+90}{12}$$

$$= \frac{1180}{12} = 98.33 \text{ million of } m^3/\text{month}$$

(ii) It is necessary to find the lengths of time during which certain flows are available to obtain the flow duration curve. This information is tabulated using the hydrograph in the following table:

Discharge (millions m^3/mo)	Total number of months during which flow is available	Percentage (%)
0	12	100
20	11	91.8
40	10	83.4
50	9	75
90	8	66.6
100	6	50
120	4	33.3
150	3	25
200	2	16.65
250	1	8.325

The flow duration curve can be drawn using the data tabulated as shown in figure above.

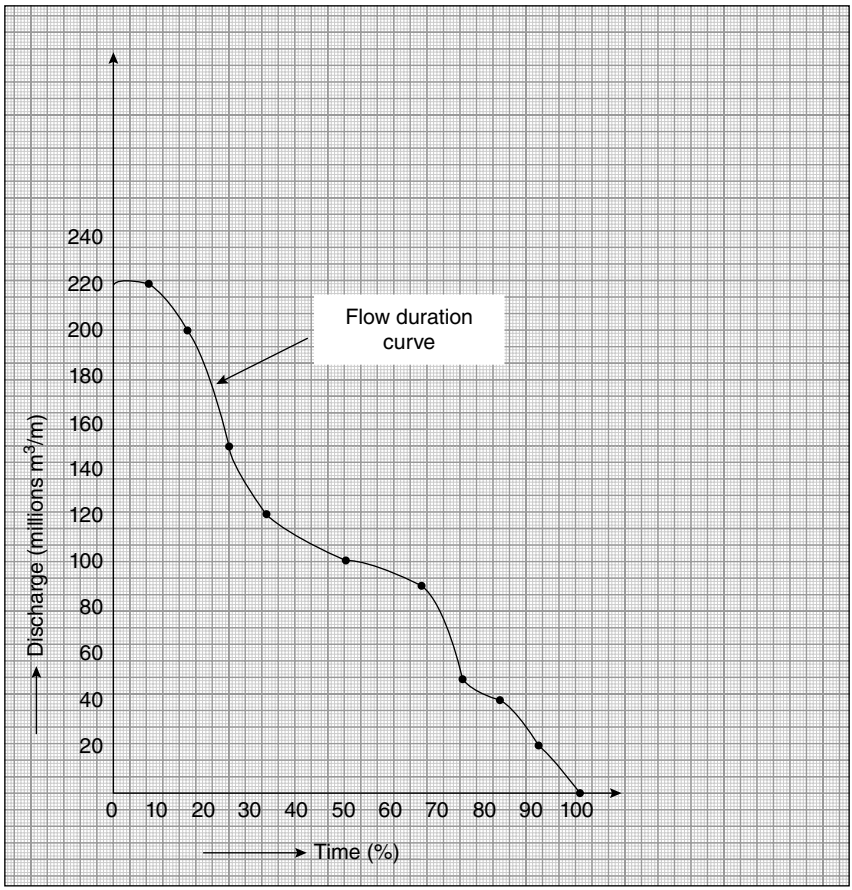
(iii) Average Mega-watt energy available:

$$= \frac{mgH}{1000} \times \eta_a \times \frac{1}{1000}$$

$$= \frac{98.33 \times 10^6 \times 9.81}{30 \times 24 \times 3600} \times \frac{1000 \times 100}{1000} \times 0.8 \times \frac{1}{1000}$$

$$= 29.76 \text{ MW}$$

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Example 15.23

The mean monthly discharge for 12 mo at a particular site of a river is tabulated below:

Month	Discharge (millions m^3/mo)	Month	Discharge (millions m^3/mo)
Apr	600	O	2,000
May	300	N	1,500
Jun	1,500	D	1,500
Jul	2,500	J	1,000
Aug	3,000	F	800
Sep	2,400	M	600

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Draw:

(i) Hydrograph for the given discharge and find the average monthly flow.

(ii) The power available at mean flow of water if the available head it 80 m at the site and overall efficiency of the generation is 80 per cent.

Take 30 day in a month.

Solution: The hydrograph is shown in the figure

The average monthly flow is given by (from the figure)

$$= \frac{(600+300+1500+2500+3000+2400+2000+1500+1500+1000+800+600)}{12}$$

$$= 1{,}475 \text{ million of } m^3/\text{mo}$$

It is necessary to find the lengths of time during which certain flows are available to obtain the flow duration curve. This information is tabulated using the hydrograph in the following table:

Discharge (millions m^3/mo)	Total number of months during which flow is available	Time during which flow is available (%)
300	12	100
600	11	91.7
800	9	76.00
1,000	8	66.60
1,500	7	58.40
2,000	4	33.30
2,400	3	25.00
2,500	2	16.65
3,000	1	8.325

By using the above tabulated data, we can draw the flow duration curve as shown below.

The average flow available per second,

$$= \frac{1475 \times 10^6}{30 \times 24 \times 3600}$$

$$= 569.06 \text{ } m^3/\text{sec}$$

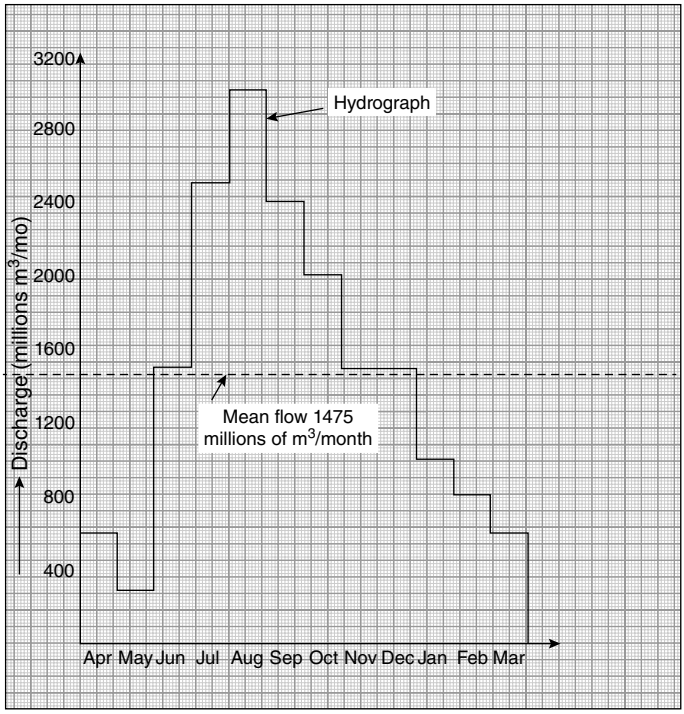
Average kilo-watt available at the side $= \frac{mgH}{1000} \times \eta_g$

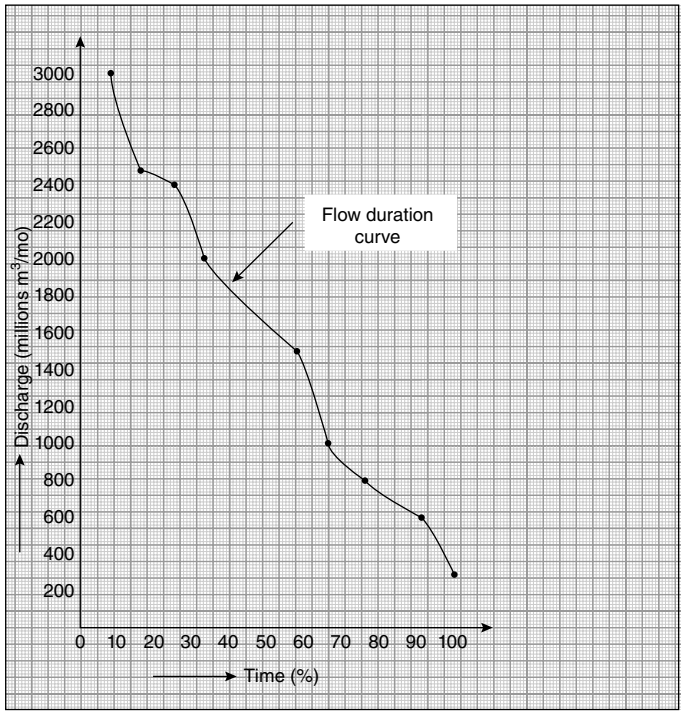
$$= \frac{569.06 \times 1000 \times 9.81 \times 80}{1000} \times 0.8$$

$$= 357278.63 \text{ kW}$$

$$= 357.28 \text{ MW}$$

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Example 15.24

The data for 12 mo flow at a particular site are given below:

Month	Flow (millions m^3/mo)	Month	Flow (millions m^3/mo)
1	100	7	190
2	50	8	40
3	20	9	30
4	80	10	200
5	10	11	170
6	10	12	100

Find the following:

(i) The required reservoir capacity for the uniform flow of 50 million m^3/mo throughout the year.

(ii) Spill-way capacity.

(iii) Average flow capacity if whole water is used and required capacity of the reservoir for this condition.

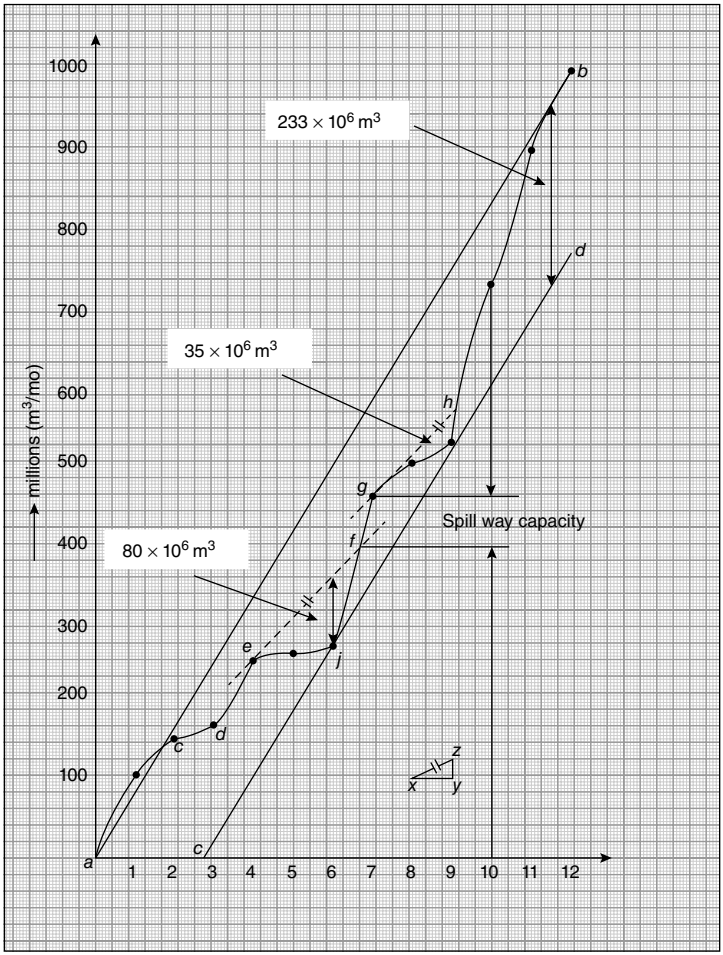
Solution: For drawing the mass curve, we have to find the cumulative volume of water that can be stored month after month. This is shown in the following table:

Month	Flow (millions m^3/mo)	Cumulative volume (millions m^3)
1	100	100
2	50	150
3	20	170
4	80	250
5	10	260
6	10	270
7	190	460
8	40	500
9	30	530
10	200	730
11	170	900
12	100	1,000

By using the values given in the table, the mass curve can be drawn.

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For finding the capacity of the reservoir for uniform flow of 50 million m^3/mo, construct the triangle (*xyz* as shown in figure below, where *xy* represents 1wqaaaaaa mo and *yz* represents 50 million m^3.



Draw parallel lines to the line *x* and *z* through the points *e* and *g* which are the apex of mass curve. The greatest departure of the mass curve from these lines is drawn parallel to *x* and *y*, which represents the storage capacity.

$$\text{Storage capacity} = 80 \times 10^6 \text{ m}^3$$

Spillway capacity required (from the figure)

$$= 85 \times 10^6 \text{ m}^3$$

Join the points *a* and *b*, then the slope of the line *a* and *b* represents the uniform discharge throughout the year.

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$$= \frac{1000}{12} \times 10^6 = 83.33 \times 106 \text{ m}^3/\text{mo}$$

Draw the line 'cd' parallel to *a* and *b* that touches the mass curve to its lowest point '*j*'. The maximum departure of the line cd from curve represents the required storage capacity for the uniform supply of 83.33×10^6 m^3/mo. In this case, storage capacity required.

$$= 233 \times 10^6 \text{ m}^3$$

Example 15.25

The data for a weekly flow at a particular site is given below for 12 weeks.

Week	Weekly flow (m^3/sec)	Week	Weekly flow (m^3/sec)
1	6,000	7	1,200
2	4,000	8	4,500
3	5,400	9	8,000
4	2,000	10	4,000
5	1,500	11	3,000
6	1,000	12	2,000

Find the size of the reservoir and the possible rate of available flow after the reservoir had built in with the help of mass curve.

Solution: For drawing the mass curve, it is necessary to find the cumulative volume of water that can be stored week after week. This is done as tabulated below:

Week	Weekly flow (m^3/sec)	Week flow (day–sec–m)	Cummulative volume (day–sec–m)
(a)	(b)	c = b × 7	(d)
1	6,000	42,000	42,000
2	4,000	28,000	70,000
3	5,400	37,800	1,07,800
4	2,000	14,000	1,21,800
5	1,500	10,500	1,32,300
6	1,000	7,000	1,39,300
7	1,200	8,400	1,47,700
8	4,500	31,500	1,79,200
9	8,000	56,000	2,35,200
10	4,000	28,000	2,63,200
11	3,000	21,000	2,84,200
12	2,000	14,000	2,98,200

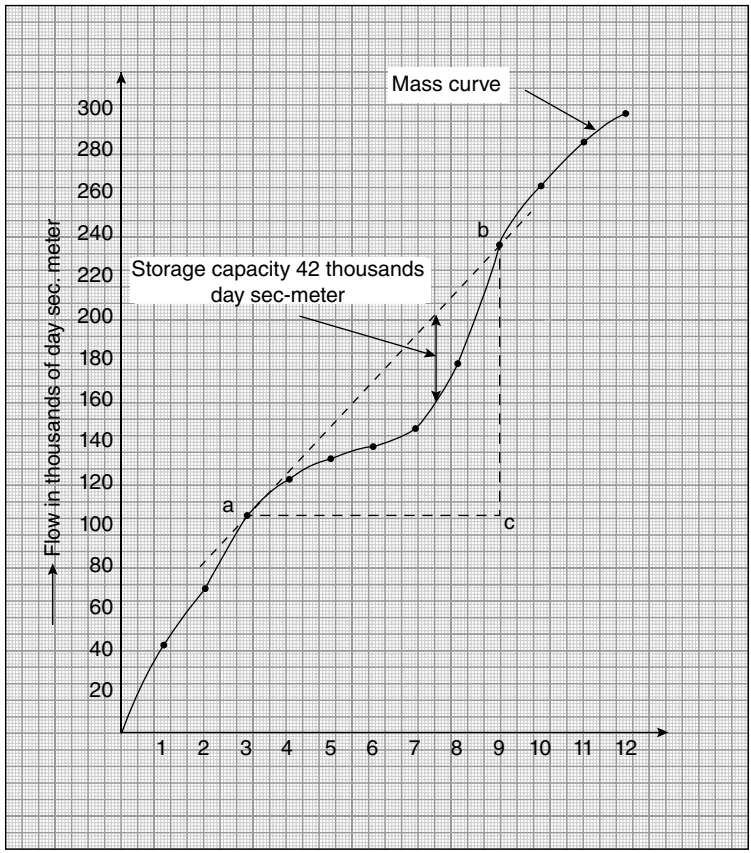
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If the mean flow is available in the week at the given rate, then the total flow in the week

$$= 7 \times \text{day} \times \text{m}^3/\text{sec}$$

$$= 7 \times \text{day–sec–m}$$

By using the above data as tabulated, the mass curve can be drawn as shown in the figure below.



To find the reservoir capacity draw the tangent at the highest point on the mass curve from 'a' and measure the highest distance between the tangent drawn and the mass curve. Hence, the capacity of the reservoir, from the figure, is

$$= 42 \times 10^3 \text{ day-sec-m}$$

The slope of the line '*ab*' gives the flow rate available for the given capacity reservoir.

$$\text{Flow rate available} = \frac{bc}{ac}$$

$$= \frac{6.4 \times 20 \times 10^3 \text{ day-sec-m}}{6 \times 7 \text{ day}}$$

$$= 2961 \text{ m}^3/\text{sec}$$

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Example 15.26

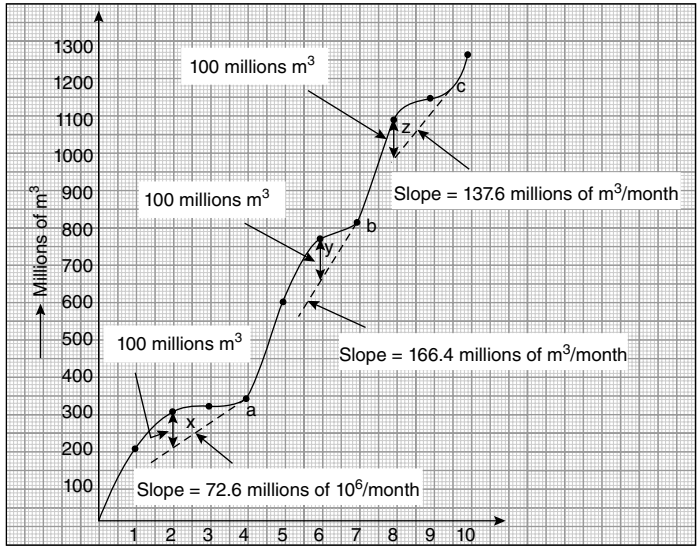
The following run-off data are collected for 10 mo for 10 mo at a particular site of the river.

Month	Discharge (millions m^3/mo)	Month	Mean discharge (millions m^3/mo)
1	200	6	180
2	100	7	40
3	20	8	280
4	20	9	60
5	260	10	120

Find the maximum flow available throughout the year if the storage capacity at the site is 100 million m^3.

Solution: From the given data, find out the cumulative volume of water that can be stored month after month as tabulated in the following table:

Month	Cumulative volume (millions m^3)	Month	Cumulative volume (millions m^3)
1	200	6	780
2	300	7	820
3	320	8	1,100
4	340	9	1,160
5	600	10	1,280



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Draw the mass curve with the above tabulated data.

Draw the tangents from the points *a, b* and *c* in such a way that the distance *x*, *y* and *z*, each will be equal to 100 million m^3 (or 1 cm as per the scale).

Now measure the slopes at the points *a*, *b* and *c*, which will directly give flow rates.

Flow rate at point *a* = 72.6 million m^3/mo

Flow rate at point *b* = 166.4 million m^3/mo

Flow rate at point *c* = 137.6 million m^3/mo

The lowest rate (72.6) among the three is the minimum available flow rate throughout the period considered.

Example 15.27

The average monthly run-off data of two rivers A and B for 12 mo is tabulated as given below. The water source of river B is from a snowfall region. The run-off is given in millions of cubic metre/month.

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
River A	40	30	30	20	20	160	180	180	100	80	50	50
River B	50	50	60	80	100	100	90	90	70	60	60	60

The head available for river A is 90 m and for river B is 95 m.

Using the above data,

(i) Find which river is more suitable for storage type hydroelectric power plant? Assume the overall efficiency of generation is same for both sites.

(ii) If a run-off-river power plant is to be established, then select the proper river for the same. The minimum quantity of water must be available for 85 per cent of the total year for run-off-river plant. Also find the ratio of power generation if both plants are used as run-off-river plants for 80 per cent period of the year at constant run-off.

(iii) The ratio of run-off of river A and river B as well as ratio of power if the constant run-off from both rivers is required for 60 per cent of the year.

(iv) At what percentage of time, the run-off rate of both sites is same?

Solution: First draw the hydrographs and then the flow duration curves for both rivers based on the data that is tabulated as given in the following table:

River A			River B		
Discharge (millions m^3/mo)	Total number of months during which flow is available	Total number of months during which flow is available	Time during which flow is available (%)	Time during which flow is available (%)	Discharge (millions m^3/mo)
20	12	100	50	12	100
30	10	83.30	60	10	83.3

(*Continued*)

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(*Continued*)

River A				River B	
Discharge (millions m^3/mo)	Total number of months during which flow is available	Total number of months during which flow is available	Time during which flow is available (%)	Time during which flow is available (%)	Discharge (millions m^3/mo)
40	8	66.60	70	6	50
50	7	58.3	80	5	41.55
80	5	41.65	90	4	33.30
100	4	33.3	100	2	16.66
160	3	25			
180	2	16.66			

(i) The average flow per month for river A

$$= \frac{40+30+20+20+150+180+180+100+80+50+50}{12}$$

$$= 77.5 \text{ million } m^3/\text{mo}$$

The average flow per month for river B

$$= \frac{50+50+60+80+100+100+90+90+70+60+60+60}{12}$$

$$= 72.5 \text{ million } m^3/\text{mo}$$

The power developed HQ

$$\therefore \frac{P_a}{P_a} = \frac{Q_a H_a}{Q_b H_b} = \frac{77.5}{72.5} = \frac{90}{95} = 1.013$$

As $P_a > P_b$, the river A is more suitable for storage-type power plant.

The flow available for 85 per cent of the time in year B is 59.5 million m^3/mo and from river A is 29 million m^3/mo.

In this case

$$P_b(Q_b H_c) = 59.5 \times 95 = 5652.5 > P_a(Q_a H_a) = 29 \times 90 = 2{,}610$$

Therefore, the site of river B is more suitable than the site of river A for run-off-river power plant.

(ii) When 60 per cent time of the year, the run-off is required from both the rivers, then from Figure (a),

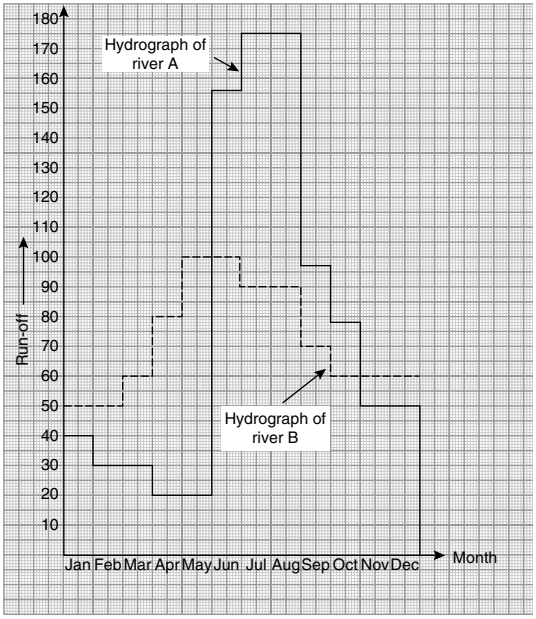
$$\frac{Q_a}{Q_b} = \frac{47}{66} = 0.712$$

and

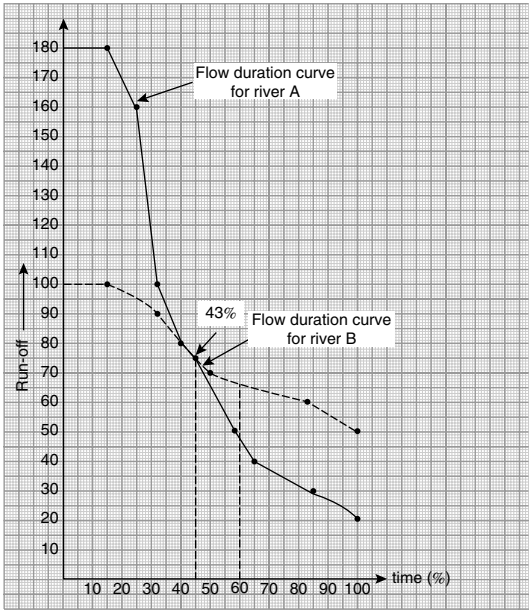
$$\frac{Q_a H_a}{Q_b H_b} = \frac{47 \times 90}{66 \times 95} = 0.675$$

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(iii) From Figure (b), the percentage of time at which the run-off for both rivers is same is the intersection point of two flow duration curves and that is 80 million m^3/mo and time available is 43 per cent.



(a)



(b)

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Example 15.28

The run-off data of two rivers for 12 months is tabulated below. The run-off is given in millions of cubic metre per month.

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
River A	50	30	20	15	10	80	140	120	100	60	50	40
River B	50	50	40	40	40	90	100	100	80	70	60	70

Using the above data,

(i) Find the ratio of run-off of river A and river B if the constant run-off is required for 40 per cent time of the year.

(ii) If the constant run-off for 80 per cent time of the total year is required, which river site is more preferable for run-off plant and why?

(iii) Which site of the river is more preferable for storage type plant and why?

(iv) At what percentage of time, the run-off rate of both rivers is same.

Solution: Draw the hydrographs and the flow duration curves for river A and B using the required data that are tabulated in the following table:

River A			River B		
Discharge (millions m^3/mo)	Total no. Of months during which flow is available	Percentage time during which flow is available	Discharge (millions m^3/mo)	Total no. Of months during which flow is available	Percentage time during which flow is available
10	12	100	40	12	100
15	11	91.6	50	9	75
20	10	83.30	60	6	50
30	9	75	70	5	41.65
40	7	58.33	80	4	33.3
50	7	58.33	90	3	25
60	5	41.65	100	2	11.66
70	4	33.3	—	—	—
80	4	33.3	—	—	—
90	2	25	—	—	—
100	3	25	—	—	—
10	2	16.66	—	—	—
120	2	16.66	—	—	—
140	1	8.30	—	—	—

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(i) From Figure (a),
At 40 per cent of the time, the flow of river B = 72 million m^3/mo.
At 40 per cent of the time, the flow of river A = 61 million m^3/mo.

$$\text{Ratio} = \frac{72}{61} = 1.18$$

Therefore, 18 per cent flow is more from river B for 40 per cent time of the year.

(ii) From Figure (b),
The flow from river A for 80 per cent of the year = 23 million m^3/mo.
The flow from river B for 80 per cent of the year = 48 million cm^3/mo.
More quantity of water is available from river B for 80 per cent of the year (48 > 23); therefore, river B is preferable for run-off-type power plant.

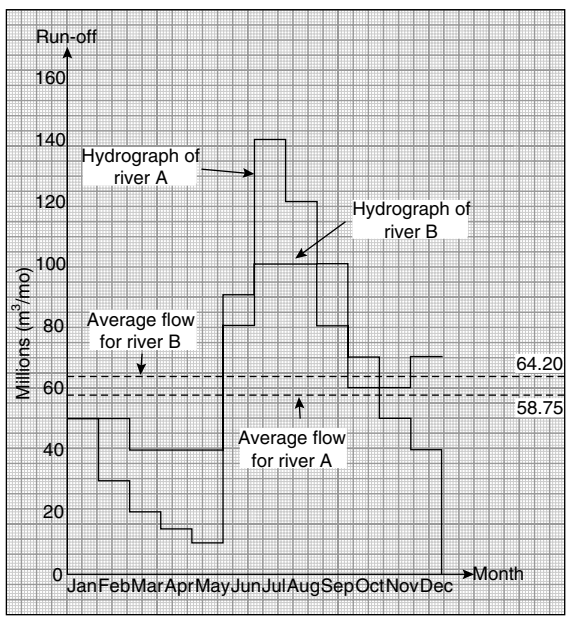
(iii) Total flow from the river A in the whole year,

$$= 50 + 30 + 20 + 15 + 10 + 180 + 100 + 120 + 100 + 60 + 50 + 40$$
$$= 775 \text{ millions } cm^3/yr$$

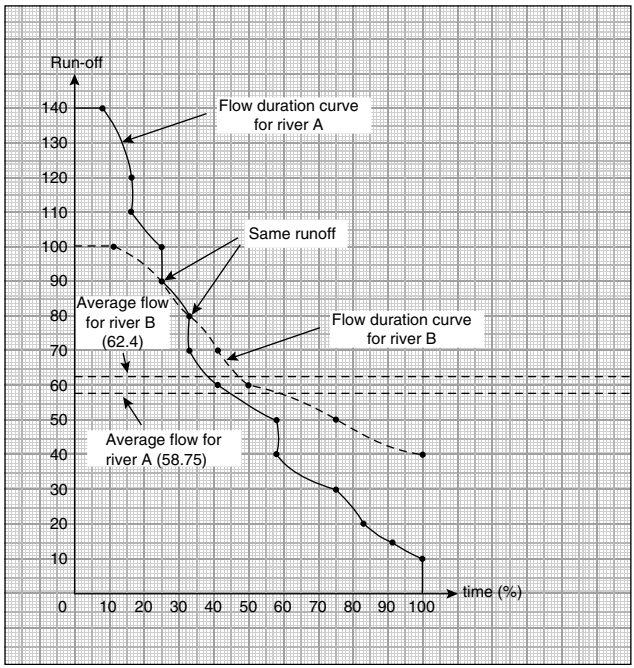
Total flow from river B in the whole year,

$$= 50 + 50 + 40 + 40 + 40 + 90 + 100 + 100 + 80 + 70 + 60 + 50$$
$$= 770 \text{ million } cm^3/yr$$

$$\text{flow river } A \text{ Average} = \frac{775}{12} = 64.58 \text{ } cm^3/mo$$



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Average flow for river $B = \dfrac{770}{12} = 64.17$ cm^3/mo

As the average flow from river B is greater than average flow from river A, therefore river B is more preferable for storage type plant also.

(iv) From Figure (b),
The run-off rate of both rivers is same at 25 per cent (90 cm^3/mo) and 33.3 per cent (80 cm^3/mo) of the year.

Example 15.29

The nature of load required for 24 hr is tabulated below.

Time period	10 AM to 6 PM	6 PM to 8 PM	8 PM to 12 AM	12 AM to 6 AM	6 AM to 10 AM
Load in MW	150	80	40	20	100

(i) Find the total input to the thermal plant if the load is supplied by the single thermal plant only. Take the thermal efficiencies of the plant at 150, 80, 40, 20, 100 MW as 35 per cent, 25 per cent, 15 per cent, 10 per cent, and 30 per cent, respectively.

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(ii) If the above load is taken by combined thermal and pump storage plant, then find the percentage saving in the input to the plant. Thermal plant efficiency at full load is 35 per cent.

(iii) Also find the overall efficiencies in both cases.
In pump storage plant, the pump and turbine are separate. The efficiency of pump is 85 per cent and water turbine is 95 per cent.

Solution:

(i) The load curve is shown in Figure (a):

Total output per day = $100 \times 4 + 150 \times 8 + 80 \times 2 + 40 \times 4 + 20 \times 6$

$= 400 + 1{,}200 + 160 + 160 + 120 = 2{,}040$ MW-hr

The input to the thermal plant $= \frac{100 \times 4}{0.3} + \frac{150 \times 8}{0.35} + \frac{80 \times 2}{0.25} + \frac{40 \times 4}{0.15} + \frac{20 \times 6}{0.1}$

$= 7668.6$ MW-hr

Therefore, the overall efficiency $= \frac{2040}{7668.6} = 0.266 = 26.6\%$

(ii) The overall efficiency of the pump storage plant

$= 0.85 \times 0.95$

$= 0.8075 = 80.75\%$

Let us assume the capacity of the thermal plant is x mega-watt when it is working combined with pumped storage plant.

The energy used from the thermal plant to pump the water of pump storage plant during off-peak period must be equal to the energy supplied by the pump storage plant during peak period.

From the Figure (a),

$[(x - 80) \times 2 + (x - 40) \times 4 + (x - 20) \times 6] \times 0.8075$

$= (100 - x) \times 4 + (150 - x) \times 8$

$\therefore\ 9.69x - 355.3 = 1{,}600 - 12x$

$x = 90.15$ MW

The energy supplied in the second case

$= \frac{90.15 \times 24}{0.35} = 6181.71$ MW-hr

The overall efficiency of the combined plant

$= \frac{2040}{6181.71} = 0.33 = 33\%$

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The percentage saving in input is the load taken by combined plant instead of single thermal plant.

$$= \frac{7668.6 - 6181.71}{7668.6} \times 100$$

$$= 19.39$$

Example 15.30

A load required by the consumers from the power plant for 24 hr in tabulated as given below:

Time period	6 AM to 10 AM	10 AM to 6 PM	6 PM to 12 PM	12 PM to 6 AM
Load (MW)	60	120	40	10

(i) Find the net revenue earned if the load is taken by a single thermal power plant. The energy rate is ₹ 1.5 per kW-hr and the cost of input is ₹ 2.2 per 20,000 kJ. The thermal efficiency of plant may be taken as 40 per cent at 120 MW load, 35 per cent at 60 MW load, 30 per cent at 40 MW load and 20 per cent at 10 MW load.

(ii) It is proposed to take the above load by a combined thermal and pump storage plant. If the thermal plant always runs at constant load with 40 per cent thermal efficiency and the overall efficiency of pump storage plant is 80 per cent, find the capacity of the thermal plant required and percentage increase in the revenue earned. The cost of power sell and cost of energy input is same as given above.

Solution: Referring to Figure (a),

(i) Total energy generated by the thermal plant during 24-hr period

$$= 60 \times 4 + 120 \times 8 + 40 \times 6 + 10 \times 6$$

$$= 240 + 960 + 240 + 60$$

$$= 1{,}500 \text{ MW hr}$$

$$= 1{,}500 \times 10^3 \text{ kWh}$$

Total cost of selling the power for 24 hr,

$$= 1.5 \times 1{,}500 \times 10^3$$

$$= ₹\ 22{,}50{,}000$$

Total input to the thermal plant during 24 hr,

$$= \frac{120 \times 8}{0.4} + \frac{60 \times 4}{0.35} + \frac{40 \times 6}{0.3} + \frac{10 \times 6}{0.2}$$

$$= 2{,}400 + 686 + 800 + 300$$

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$$= 4185.71 \text{ MW-hr}$$

$$= 4185.71 \times 10^3 \text{ kWh}$$

Therefore, the total cost of input energy during 24 hr,

$$= 2.2 \times \frac{1}{20000} \times \frac{4185.71 \times 10^3 \times 3600}{1}$$

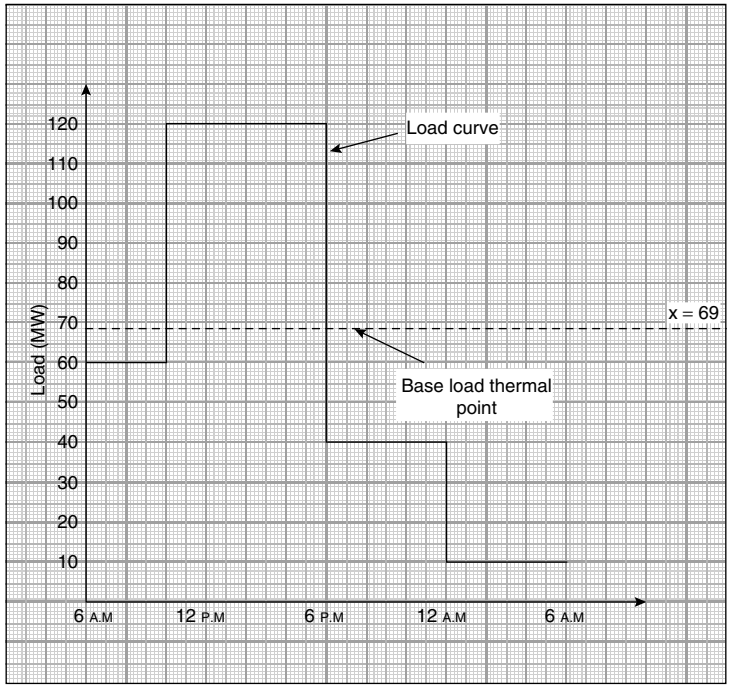
$$= ₹\ 1657542.86$$

Therefore, the net revenue earned from the thermal plant is

$$= 2{,}250 \times 10^3 - 1657541.16$$

$$= ₹\ 592457.14 \text{ per day.}$$

(ii) Assume the capacity of the thermal plant is x MW when it is working in combination with pump storage plant.



From Figure (a), the energy used from thermal plant to pump the later of pump storage plant during off-peak period must be equal to the energy supplied by the pump storage plant during peak period.

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$$\therefore [(x-60)\,4 + (x-40)\,6 + (x-10)\,6] \times 0.8 = (120-x)8$$

$$1.8x - 432 = 960 - 8x$$

$$\therefore 20.88x = 1{,}392$$

$$x = \frac{1392}{20.8}$$

$$= 66.92 \text{ MW}$$

Therefore, the base load of 66.92 MW must be taken by thermal plant.

As the total energy generated remains the same; therefore, total revenue earned also remains same.

Now total input to the thermal plant in 24 hr

$$= \frac{66.92 \times 24}{0.4}$$

$$= 4015.2 \text{ MW-hr}$$

$$= 4015.2 \times 10^3 \times 3{,}600 \text{ MJ}$$

$$= 14.455 \times 10^9 \text{ MJ}$$

Therefore, the total cost of input energy during 24 hr

$$= 1.5 \times \frac{1}{20000} \times \frac{14.455 \times 10^9}{1}$$

$$= ₹\ 1084.1 \times 10^3$$

Net revenue earned from the combined plant per day

$$= 2{,}250 \times 10^3 - 1084.1 \times 10^3$$

$$= ₹\ 1165.9 \times 10^3$$

Percentage increase in the profit,

$$= \frac{1165.9 - 592.459}{592.459} \times 100$$

$$= 96.78\%$$

15.6 HYDRO ELECTRIC PLANT – INTRODUCTION

Figure 15.23 shows a hydel power plant equipped with a Pelton wheel. Water from a high head source or reservoir like dam enters the turbine runner through large diameter pipes known as *penstocks*. Each penstock pipe is branched in such a way that it can accommodate a nozzle at the end. Water flows through these nozzles as a high-speed jet striking the vanes or buckets attached to the periphery of the runner. The runner rotates and supplies mechanical work to the shaft. Water is discharged at the *tail race* after doing work on the runner.

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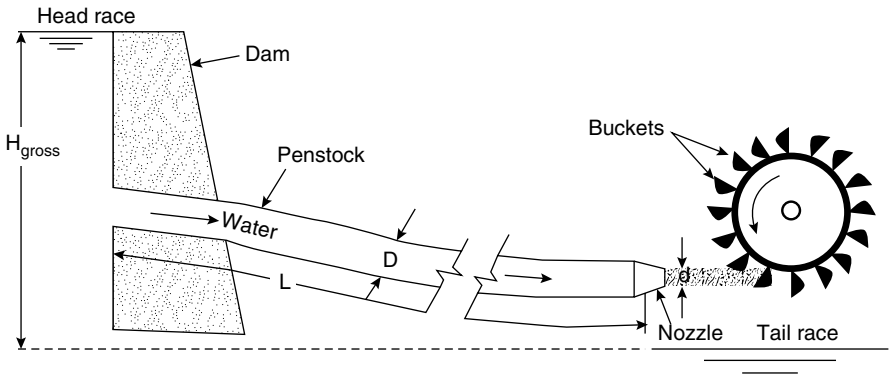


Fig. 15.23 *Hydraulic Power Plant*

Definitions

(i) *Head race:*

It is the difference between the reservoir water level at any given instance.

(ii) *Tail race:*

It is the level of water at the turbine exit.

(iii) *Penstock:*

It is a large-diameter pipe carrying water to the turbine runner. Diameter of the penstock is given by D and length by L. The plant may have one or more penstocks.

(iv) *Water reservoir:*

Large quantity of water is stored usually by building a dam across a flowing river. The water that gets collected behind this obstruction (dam) is known as *water reservoir* or *catchment area*.

(v) *Gross head:*

It is the head of water available for doing useful work. It is the difference between the head race and tail race. It is denoted by H_{gross}. Generally, the tail race level is taken as a reference level for finding the gross head.

(vi) *Net head:*

It is the head available at the turbine inlet. Due to pipe bends and fittings a fraction of kinetic energy of water is lost. Some amount of head is also lost due to friction in the pipes that carry water from head race to the turbine runner. Compared to head loss due to fittings, bends, entry, etc., the loss due to friction is more which considerably reduces the gross head available for doing useful mechanical work.

If h_f = head loss due to friction then net head,

$$H_{net} = H_{gross} - h_f = H$$

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Using Darcy's equation for h_f,

$$H = H_{net} = H_{gross} - \frac{4fLV^2}{2gD} \quad (1)$$

where f = friction coefficient
V = velocity of water flowing in penstock, m/s
L = length of penstock of diameter D, m

15.6.1 Site Selection for Hydel Power Plants

In a hydel power plant several structures such as dam, conduits, intakes, surge tank and power house are involved. To select a site, several alternative layouts are to be studied, and the most economical one is adopted. The scope for the site investigation depends partly on the availability of existing published data, partly on the nature and size of the proposed plants and partly on the difficulties pre-set by the existing topographical and geological features which in turn affects the number of alternatives which have to be studied. Site investigations are carried out in two main stages:

1. Preliminary investigation – to provide sufficient information to find out the practicability of the proposed scheme and to choose between alternatives.
2. Final investigation – detailed exploration of recommended site to establish the suitability and to enable the final design. The investigations include the following: (i) hydrological, (ii) topographical, (iii) geological, (iv) consideration of water pollution effects.

(i) *Hydrological investigations*

(a) *Water availability* Run-off data at the proposed site must be available beforehand. If not available calculation are based on the rainfall data.

(b) *Water storage* Necessity to store the water for continuous power generation as there is a wide variation in rainfall during the year. Storage capacity and minimum quantity of water available are calculated with the help of mass curve. Two types of storage in use are as follows:

1. Storage for 1 year only with no carryover for the next season full at the beginning empty at the end of the year
2. Enough storage sufficient even during worst dry periods

(c) *Water head* This depends on topography. Increase in head reduces water to be stored and therefore capital cost.

(d) *Ground water data* Stability of ground slopes, amount and type of grounding required to prevent leakage.

(ii) *Topographical investigation*

Topographical features of the whole catchment area should be mapped either by ground surveying or aerial surveying methods (for larger area and difficult terrain).

(iii) *Geological investigation*

To provide most accurate picture of the ground character of the site selection depends on the following factors:

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(a) Light basin of ample size
(b) Narrow outlet requiring a dam of less volume
(c) Strong foundation to support dam structure
(d) Safe and ample spill way to dispose surplus water
(e) Water should not submerge valuable mineral and agricultural land
(f) Availability of material for constructing a dam
(g) Distance from load centre to reduce the cost of creation of transmission lines and their maintenance
(h) Access to site by rail and road

(iv) *Consideration of water pollution effects*

Poor-quality water results in foul-smelling gases, corrosion of metallic structure and electrical equipment and overheating of machine parts. Water gets polluted by submerged vegetation mineral deposits and presence of algae. Leaves, twigs, logs under-water decay generate bad-smelling gases. Thermal stratification and depletion of oxygen in lower strata generate harmful gases, such as CO_2, H_2S, CH_4, which pollute the water.

Effects of water pollution

(i) *Damage to dam structure and fitting*

Water containing H_2S, CO_2 and CH_2 is highly corrosive to dam structure. H_2SO_4 formed by oxidation of H_2S attack of the cement and disintegrate the concrete. These gases increase the solubility of soil and rocks. Structures, metallic piping and penstock are also affected by the gases.

(ii) *Damage of electrical machinery*

Presence of corrosive gases such as H_2S and SO_2 affects the voltage drop at brush contacts in generator leading to poor commutation. Silver contacts of relays get blackened and lead to high contact resistance and failure of relay occurs. Copper bus bars corrode.

(iii) *Health hazard to operating personnel*

Headache, dizziness, dryness, nose, throat and chest pain some of the effects of H_2S. High concentration of H_2S even causes paralysis.

(iv) *Sedimentation effect*

It reduces the capacity of reservoir and causes rapid erosion of turbine blades.

(v) *Environment aspects of site selection*

It is compatible with the local environment.

15.6.2 General Arrangement of Hydroelectric Power Plant

The basic requirement of a hydroelectric power stationer is a reservoir where large quantity of water is stored during flood season and used during dry season. The reservoir is generally built of constructing a dam across a river. The water from reservoir is drawn by the forebay through an open canal or tunnel. The water from the forebay is supplied to water turbine through penstock that is located at much lower level than the height of water in reservoir (Figure 15.24)

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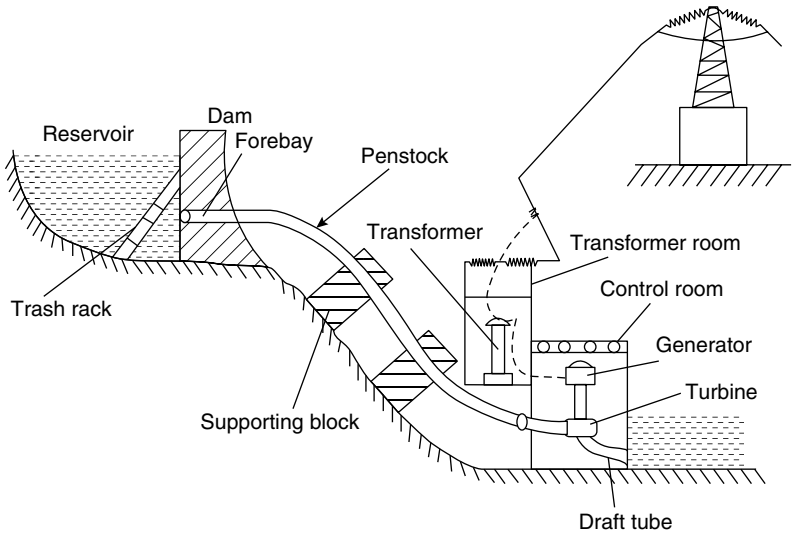


Fig. 15.24 *Arrangement of Hydroelectric Power Plant*

The different components used in this power plant are listed below:

(i) *Reservoir*

The purpose of reservoir is to store water during rainy season and supply the same during dry season.

(ii) *Dam*

The purpose of dam is to increase the height of water level, which increases the storage capacity.

(iii) *Trash rack*

The purpose of trash rack is to prevent the entry of debris, which might damage the wicket gate and turbine.

(iv) *Forebay*

This is a naturally provided storage, which is able to absorb the flow variation.

(v) *Surge tank*

Surge tanks are introduced between the dam and power houses nearest to the power house and preferably on the high ground to reduce the height of tower to provide better regulation of water pressure in the systems during variable load conditions. It helps in stabilizing the velocity and pressure in penstock and reduces the water hammer effect.

(vi) *Penstock*

A pipe between the surge tanks and prime mover is known as penstock. It is made of steel through reinforced concrete.

(vii) *Spill way*

It is a safety valve for a dam to discharge major floods without damage to dam. It keeps the reservoir level below some pre-determined maximum level.

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(viii) *Power house*

It has two parts, a substructure to support the hydraulic and electrical equipment and a superstructure to house and protect this equipment.

(ix) *Prime mover*

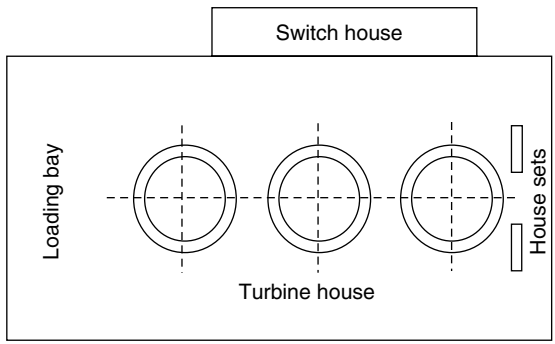
Pelton turbine, Frances turbine, Kaplan turbine and propeller turbine are used.

(x) *Draft tube*

It is a diverging discharge passage connecting the runner with tail race to regain KE of water coming out from reaction turbine.

15.6.3 Plant Layout

The general layout of the plant is decided by the type of the hydraulic plant. In the case of vertical turbines, the most convenient and economical layout will be placing turbines in a line parallel to the length of the turbine house. The spacing between the machines is determined by the size of the scroll case and the width of the fume or by the overall diameter of the alternators. If the machines are horizontal, the most suitable layout will be arranging the turbines at right angles to the length of the turbine house or even can replaced parallel to the longitudinal axis of the turbine house. A repair bay should be provided at one end of the turbine house close to the workshop and sufficient space should also be provided for the dismantling and re-erection of machines. A typical layout of a water plant with vertical machines is shown below.



Plant Layout

15.6.4 Penstock and Water Hammer

It is a short length of pipe connecting turbine and main water way. Penstocks are made of steel, RCC or wood. Cast iron is used for small plants. The intake of penstock at the dam or forebay of canal should be positioned such that it always provides adequate water even at low level. The penstock should be inclined towards the power house and its grade may be so as to match with the topography. Sharp bends in the penstock are avoided to reduce the head loss. Penstocks are not covered because exposed pipes are cheaper and easily accessible for repair and maintenance. Covered penstocks are used whenever the topography is such that there may be slides of snow, rock, earth, etc. The water flow rate is 2 m/sec, 4 m/sec and 7 m/sec in low-, medium- and high-head power plants, respectively.

Penstocks are classified as follows:

(i) Exposed penstocks

(ii) Burried penstocks

Exposed penstocks are less expensive having longer life. Their inspection and maintenance is easier.

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Burried penstocks are used in cold climates where freezing is expected. However, they have shorter life period.

15.6.5 Specific Speed and Capacity Calculations

Hydraulic machines designed for different specifications may not give the desired results. As the components of the machines are very costly, it is necessary to predict the performance of a prototype of hydraulic machines.

The discharge, speed, power, efficiency, etc., of all hydraulic machines are the functions of working head H. In order to compare machines of same type having different parameters and hence to use the experimental data, all the quantities are reduced to unit head and expressed in terms of unit head. These reduced quantities are known as unit quantities, e.g. unit speed, unit discharge, unit power, etc.

To facilitate the comparison between hydraulic machines of different types, some specific quantities obtained by reducing any quantity to a value corresponding to unit head and unit size are used. These quantities are known as *specific quantities*, e.g. specific speed, specific power, specific discharge.

Unit quantities refer to the turbine parameters that are determined when the turbine is operating under unit head. For estimating unit quantities, the overall efficiency is presumed. For this presumption, the velocity triangles under the actual working head and any other assumed head should be similar.

Consider the inlet and outlet velocity triangles of a turbine as shown in Figure 15.25. Let u be the peripheral velocity, V be the absolute velocity, V_r be the relative velocity and V_m be the flow velocity when the turbine is working under normal head. Let u', V', V_r' and V_m' be the corresponding values when the turbine is working under assumed head H'.

Since absolute velocity $V \alpha \sqrt{H}$ and for similar velocity triangles,

$$\frac{U}{U'} = \frac{V}{V'} = \frac{V_m}{V_m'} = \frac{V_r}{V_r'} = \frac{\sqrt{H}}{\sqrt{H'}} \tag{1}$$

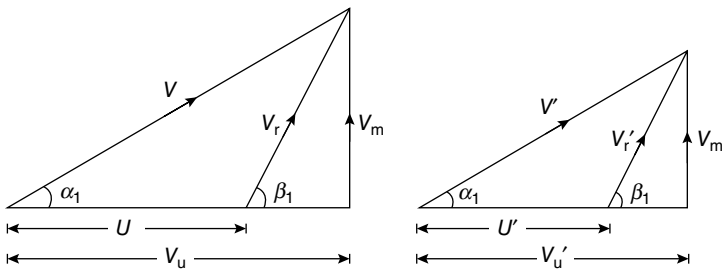


Fig. 15.25 *Inlet Velocity Triangles*

Specific speed of any turbine is defined as the speed of turbine geometrically similar to an actual turbine working under unit head and developing unit power. It is denoted by the letter N_s.

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The overall efficiency of the turbine is given by

$$\eta_0 = \frac{P}{\text{Water Power}} = \frac{P}{W \times Q \times H}$$

Hence power developed at the shaft

$$P = \eta_0 \times W \times Q \times H$$

$$P \, \alpha \, (Q \times H)$$

$$P \, \alpha \, \frac{H^{3/2}}{N^2} \times H$$

Hence, the specific speed of turbine is

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} \tag{10}$$

Significance of specific speed is that it provides a basis on which different types of turbines can be compared irrespective of their sizes. From the above Eqn. (10), we can obtain the following information:

(i) As the specific speed is proportional to the speed of rotation high-speed propeller turbines (Kaplan type) have high specific speeds compared with low speed units (Pelton wheels).

(ii) As speed is inversely proportional to head, high head machines (Pelton wheels) have low specific speed than the low head machines.

(iii) The units of specific speed is revolutions per minute ($kW^{1/2}m^{-5/4}$). But it is a common practice to express it in rpm ignoring the unit in the brackets.

(iv) All turbines of the same shape have the same specific speed.
In dimensionless form, specific speed is expressed as follows:

$$\Omega = \frac{N\sqrt{P/\rho}}{gH^{5/4}}$$

Example 15.31

A turbine is to be operated under a head of 25 m at 200 rpm. The discharge is 9 m^3/s. If the efficiency is 90 per cent determine, (i) specific speed of the turbine, (ii) power generated and (iii) type of the turbine.

Solution:
Given data:
$H = 25$ m, $N = 200$ rpm, $Q = 9$ m^3/s and $\eta_0 = 90\%$

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(i) Specific speed of turbine

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

$$\eta_0 = \frac{P}{WQH}$$

$$P = \text{WQH}\eta_0 = \frac{9810 \times 9 \times 25 \times 0.9}{1000}$$

$$= 1986.53 \text{ kW}$$

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{200\sqrt{1986.53}}{25^{5/4}}$$

$$= 159.46$$

(ii) Power generated $P = 1986.53$ kW

(iii) As the specific speed lies between 60 and 300, Francis turbine is the choice.

Example 15.32

A Kaplan turbine with a vertical shaft works under a head of 9.8 m. The diameter of the runner is 930 cm. At the optimum conditions, the turbine runs at 51.7 rpm and develops 45 *MW* power, discharging 3.21×10^4 m^3/min water. Determine specific speed of the turbine. If the critical value of the cavitation coefficient for the runner is 0.95, determine the location of the runner with respect to the tail race water level. The atmospheric and saturated vapour pressure may be taken as 1.02 bar and 0.05 bar, respectively.

Solution:

Given data:

$H = 9.8$ m, $D = 930$ cm, $N = 51.7$ rpm, $P = 45$ *MW* and $Q = 3.21 \times 10^4$ m^3/min

$$\text{Specific speed, } n_s = \frac{N\sqrt{P}}{H^{5/4}}$$

$$= \frac{51.7\sqrt{45 \times 10^3}}{9.8^{5/4}}$$

$$= 632.50$$

The critical value of the cavitation coefficient

$$\sigma_c = \frac{p_b - p_s}{1/2\,\rho V^2} = 0.95$$

where $p_b = p_{atm} - p_{vap}$

$$= (1.02 - 0.05)\ 10^5 = 0.97 \times 10^5 \text{ N/m}^2$$

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$$V = \frac{Q}{A} = \frac{535}{\frac{\pi}{4}(9.3)^2} = 7.876 \text{ m/s}$$

$$\therefore \quad 0.95 = \frac{0.97 \times 10^5 - p_s}{\frac{1}{2} \times 1000 \times (7.876)^2}$$

i.e. $2.946 \times 10^4 = 0.97 \times 10^5 - P_s$

$$\therefore \quad p_s = 6.754 \times 10^4 \text{ N/m}^2 = W \times H_s$$

Location of runner with respect to tail race water level

$$H_s = \frac{6.754 \times 10^4}{9810} = 6.88 \text{ m}$$

15.7 CLASSIFICATION OF HYDRO-PLANT

The hydroelectric power plants can be classified as follows:

1. *According to the availability of water head,*

(a) High-head plant > 100 m.

(b) Low-head plant < 30 m.

(c) Medium-head plant 30 – 100 m.

2. *According to the nature of load,*

(a) Base load plant

(b) Peak load plant peak

(c) Storage-type plants

3. *Run-off river power plants*

(a) With pondage

(b) Without pondage

4. *Pumped storage power plants*
5. *Mini- and micro-hydel plants*

15.7.1 Low-, Medium- and High-Head Plants

(i) *Low-head plant*

If the head available is only up to 50 m, the plant is known as low head power or canal power plant (Figure 15.26).

A dam is built on the river and the water is diverted into a canal that conveys water into the forebay from where water is allowed to run through turbines. After doing work, water is discharged to the turbine tailrace. Head gates fitted at the mouth of the canal controls flow in the canal. Before water

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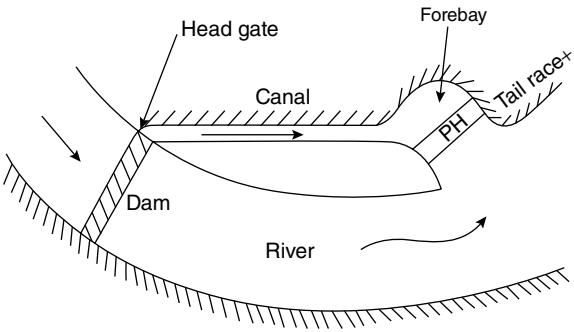


Fig. 15.26 *Low Head Plant*

enters the turbine, it is made to pass through screens or trash racks to eliminate suspended matter. If there is excess water due to increased flow in the river or due to decrease of load in the plant, it will flow over the top of the dam. A waste weir can also be constructed along the forebay so that excess water flows over it into the river. A drain gate provided on the side of the waste weir is used for periodic cleaning and repair of the waste weir and the forebay. The head gate is closed while the drain gate is open.

(ii) *Medium-head plant*

It the head of water available is more than 50 m, but below 100 m, the plant is known as medium head plant (Figure 15.27).

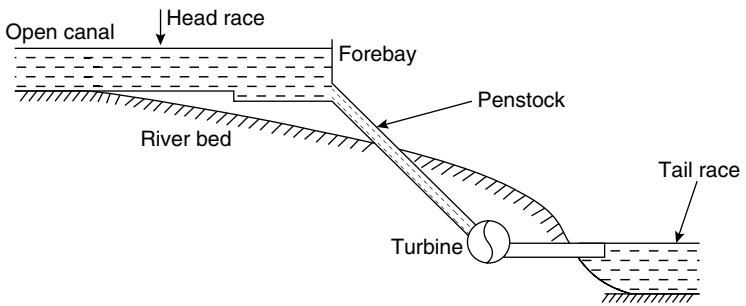


Fig. 15.27 *Medium Head Plant*

Water stored in the forebay is conveyed to the turbine through penstocks. In these plants, the river water is tapped off to a forebay on one bank of the river.

(iii) *High head plants*

If the head of water available is above 100 m, the plant is known as a high-head plant Figure 15.28. The water is taken from the reservoir through tunnels to the penstock from where it is supplied to the power house. The water can also be taken from the reservoir to a small reservoir known as forebay by means of tunnels, and from there it can be distributed to the penstocks. The function of the forebay is to distribute the water to the penstocks leading to turbines. The inflow to the forebay is regulated in such a way that the level of water in it always remains constant, so that the turbines will be fed always with a constant head. The forebay is helpful in regulating the water flow as per the load on the turbine. Trash racks fitted at the inlet of the tunnels prevent the foreign matter from going into the tunnels. When it is not possible to construct forebays, vertical construction known as surge tanks are built before the valve house and after the tunnel

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from where head works. The function of the surge tank is to prevent a sudden pressure rise in the penstock when load on the turbine decreases and the turbine inlet valves are suddenly closed. The valve house contains butterfly valves or sluice valves driven by electric power to regulate the water flow into the penstock. The gate valves regulate the flow of water through the turbines.

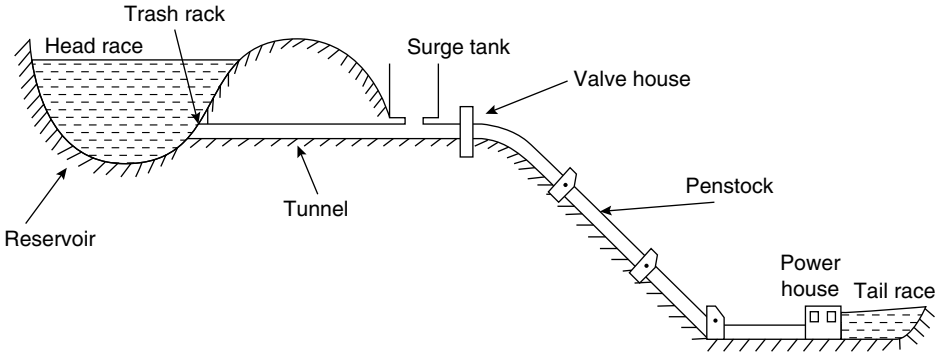


Fig. 15.28 *High Head Plant*

15.7.2 Pumped Storage Plant

These plants supply the peak load for the base power plants and pump all or a portion of their own water supply. The plant contains a tail water pond and a head water pond connected by a penstock. The generating pumping unit is at the lower end. During off-peak hours, some of the surplus electric energy generated by the base load plant is utilized to pump water from tail water pond into the head water pond. This energy will be stored there. During peak load, this energy will be released by allowing the water to flow from the head water pond through the water turbine of the pumped storage plant. These plants can be used with hydro, steam and IC engine plants (Figure 15.29).

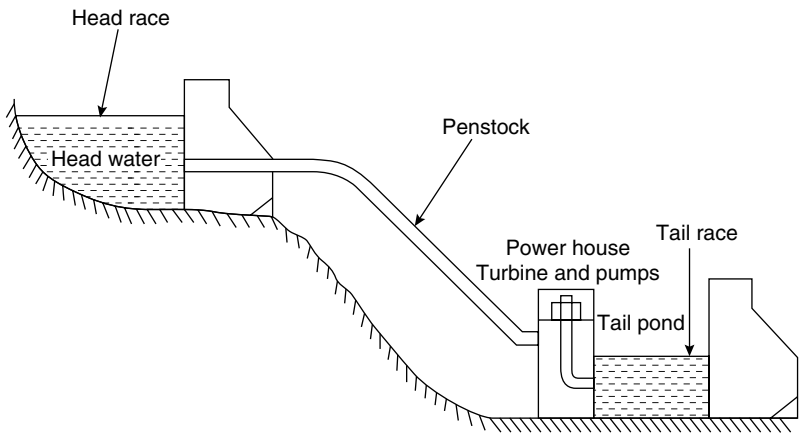


Fig. 15.29 *Pumped Storage Plant*

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The plants have either horizontal or vertical shaft arrangement. Modern plants use Francis turbine. When water flows from the head race through the runner, it acts as turbine and rotates the generator. When it is rotated in the reverse direction by means of an electric motor, it will act as a pump to shunt the water from the tail water pond to the head water pond.

15.7.3 Run-Off River Power Plant

This power plant can be with pondage or without pondage. A run-off river plant without pondage has no control over river flow and uses the water as it comes. These plants usually supply peak load. During floods, the tail water level may become excessive rendering the plant inoperative. The run-off river with pondage may supply base load or peak load power. At the time of high water flow, it may be base loaded and during dry seasons it may be peak loaded.

15.8 SURGE TANKS

It is a storage reservoir fitted to the penstock at a point near the turbine. When the load on the turbine reduces, the gate valves of the turbine are closed partly by the governor to maintain constant speed. Due to this sudden closure of the valve, the pressure inside the penstock increases rapidly. Due to this, high-pressure waves are set up, which move to and fro, hitting the valves. This may result in heavy vibration and ultimately may break the pipe joints. This phenomenon is known as water hammer.

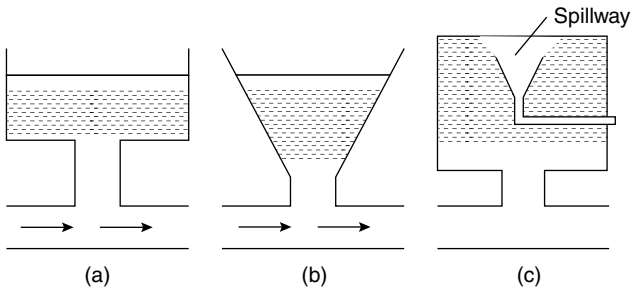
The surge tank controls the water when the load on the turbine decreases. The excess water is stored in the surge tank, and hence the pressure variations in the penstock are taken care of preventing water hammer. The surge tanks may be opened or closed at the top. Various types of surge tanks are shown in Figure 15.30. It may be either conical or cylindrical in shape with a vertical pipe connecting the penstock. It may be having a bell-mouth spillway to discharge excess water. Figures 15.30(c) and (d) show a differential surge tank, with a central riser having a small hole at its lower end. Water enters the surge tank through this hole and the flow rate depends on the c/s area of the hole. Figure 15.30(e) is a surge tank having separate galleries. The upper gallery stores water when load on the turbine is less. The lower gallery supplies water to the turbine when the load on the turbine increases.

Functions

(i) Reduces the water hammer effect

(ii) Acts as a relief valve

(iii) Acts as a temporary reservoir



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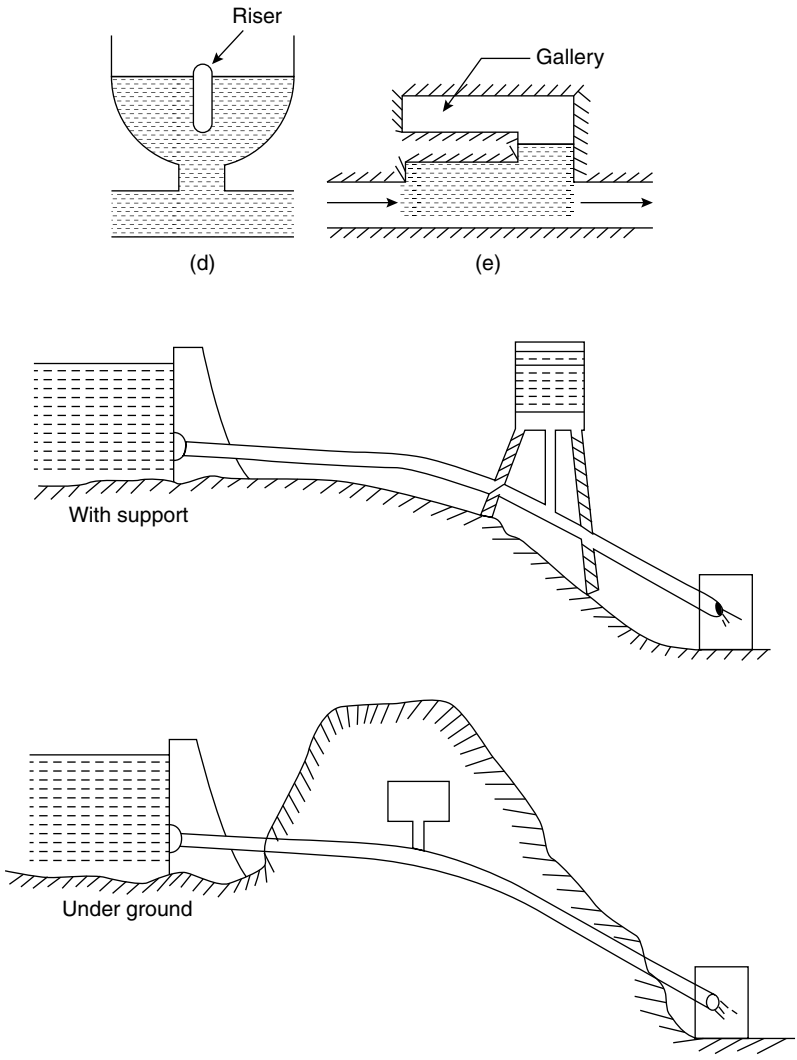


Fig.15.30 *Different Types of Surge Tanks*

15.9 CONTROL GATES AND VALVES

The storage capacity above spillway crest can be increased by installing movable gate.

15.9.1 Control Gates

(i) *Vertical lift gate*

Figure 15.31 shows a vertical lift gate.

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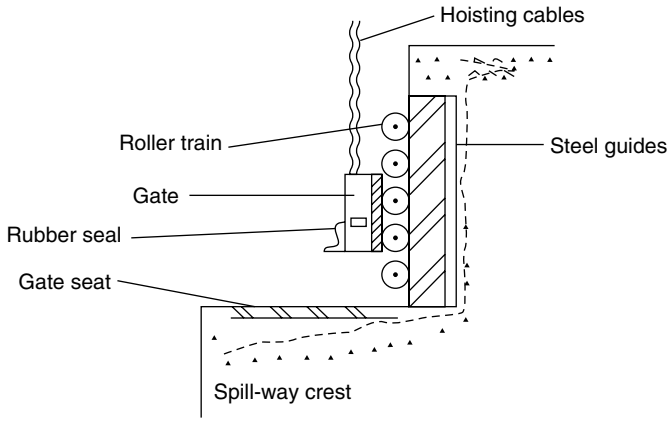


Fig. 15.31 *Vertical Lift Gate*

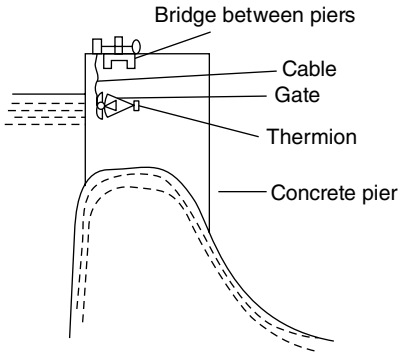


Fig. 15.32 *Radial Gate*

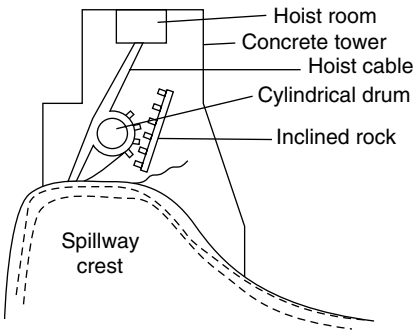


Fig. 15.33 *Rolling Gate*

Characteristics

(a) Steel gates slide in vertical guides on piers on the crest of the dam

(b) Used for small power plants

(c) Complicated structure

(d) Creates mechanical problem

(ii) *Radial gate*

Figure 15.32 shows a radial gate.

Characteristics

(a) Segment of cylinder is supported on a steel frame work which is pivoted on trunnion.

(b) Hoisting cables are attached to the gate.

(c) It involves less friction and less hoisting load.

(iii) *Rolling gate*

Figure 15.33 shows a rolling gate.

Characteristics

(i) Moving cylinder mounted between two piers

(ii) Rolls on rack with the help of hoist cable

(iv) *Drum gate*

Figure 15.34 shows a rolling gate.

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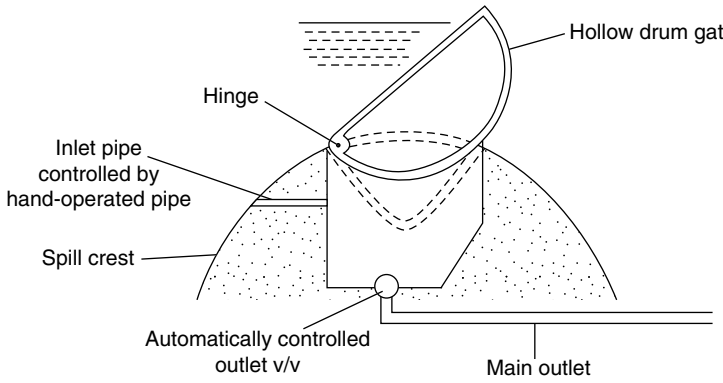


Fig. 15.34 *Drum Gate*

Features

(a) Gate consists of segment of cylinder.
(b) It fits in a recess in the top of spillway in open position.
(c) It is hinged at upstream edge.
(d) Large recess is required and is not used in small dams.

(v) *Tilting flap gate*

Figure 15.35 shows a fitting flap gate.

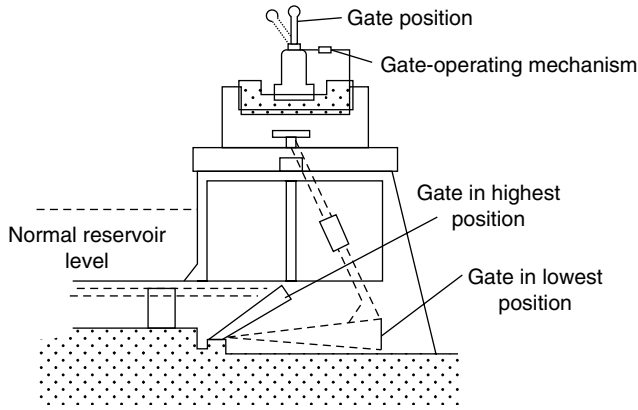


Fig. 15.35 *Tilting Flap Gate*

Features:

(a) The lower edge of flap is hinged and the upper edge is held in position by chains or screwed rods supported by an overhead bridge.
(b) It is suitable for small-size openings.

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15.9.2 Interior Gate Valves

Gate valves are used to regulate the flow. For moderate heads, butterfly valves are preferred. For high-head installations, needle valves and tube valves are used. These valves are fitted at the downstream end of the sluices.

(i) *Needle valve*

Figure 15.36 shows a typical needle valve used for flow regulation.

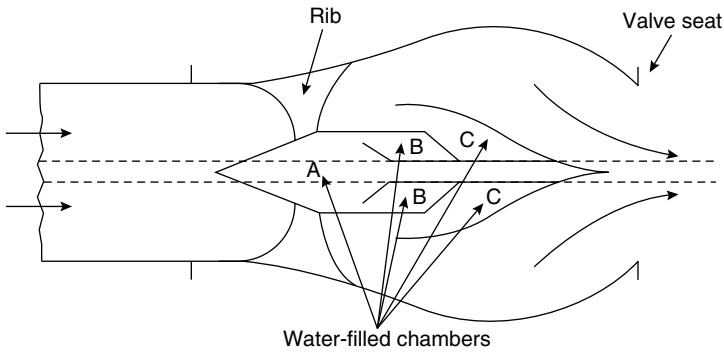


Fig. 15.36 *Needle Valve*

The needle valve consists of three water-filled chambers *A*, *B* and *C* in which hydraulic pressures can be varied. The valves are opened and closed by varying the pressures in these chambers. The valve is opened by increasing the pressure in the chambers *B* and releasing the chambers *A* and *C* that are inter-connected and vice versa.

(ii) *Tube valve*

Tube valve shown in figure 15.37, can be opened or closed by mechanical means. The valve cylinder (tube) is moved towards or away from the valve seat with the help of screw stem activated by the bevel gear to open or close the valve. The valve is light in weight and short in length as compared with the needle valve Figure 15.37.

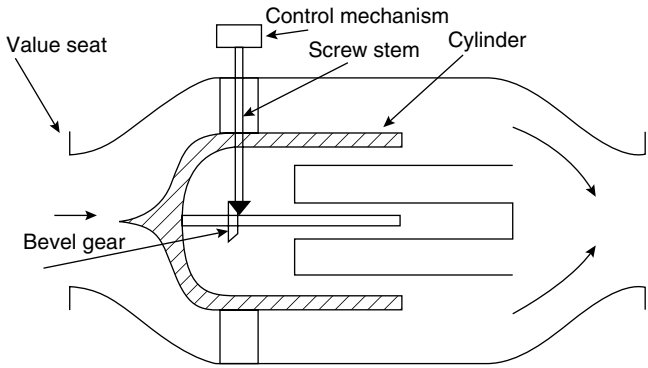


Fig. 15.37 *Tube Valve*

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(iii) *Butterfly valve*

This valve is used both as intake gate and as turbine valve. It helps inspection without dewatering the penstock line. It consists of a lens-shaped disc that can be moved in the frame of the valve. Figure 15.38 shows a Butterfly valve.

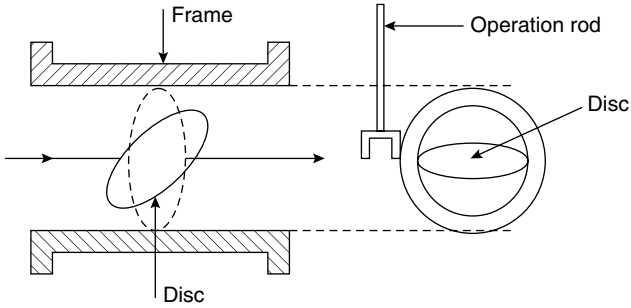


Fig. 15.38 *Butterfly Valve*

15.10 CLASSIFICATION OF DAMS AND SPILLWAYS

Dams are structures built across a river to store water in the form a reservoir before water flows down the river. This water is used to run turbines hydro power stations to generate power. Basic function of a dam is to collect water during the rainy season and store it, build a necessary head and thus facilitate steady flow of water through the turbines, year round. Apart from this, dams are built to control floods and for irrigation. Dams are classified as based on hydraulic structure design, function and material of construction and shape as follows:

(i) Based on hydraulic structure design: Based on structure design, dams can be classified as arch dams, gravity dams and buttress dams.

Gravity dams A gravity dam is made either from concrete or from stone masonry, or sometimes both. As the dam is held by gravity to the ground stopping the water in the reservoir pushing it over, it is called as a gravity dam. The dam is roughly triangular in cross section and best suited to sites with either wide or narrow valleys. For stability, resultant force, R, acting should pass through at a point 1/3 of the height from the dam base. The buttress dam is similar to a gravity dam, except that it uses less material due to hollow spaces between the buttresses. Like gravity dams, they are suited for both narrow and wide valleys. Buttress dams are made from concrete or masonry with a watertight upstream side supported by triangular shaped walls, known as buttresses. The buttresses are spaced at intervals on the downstream side for resisting the force exerted by the reservoir water, pushing the dam over (Figure 15.39).

Arch dams These are made from concrete structured in the form of a curve resembling to an arch, with the top of the arch pointing back into the water. Because of this shape, the arches offer a resisting force to the pushing force of the water behind the dam. In an arch dam, all the forces act perpendicular to the surface thereby compressing the material in the arch. Arch dams are constructed in narrow, steep sided valleys, with good rock for foundations (Figure 15.40).

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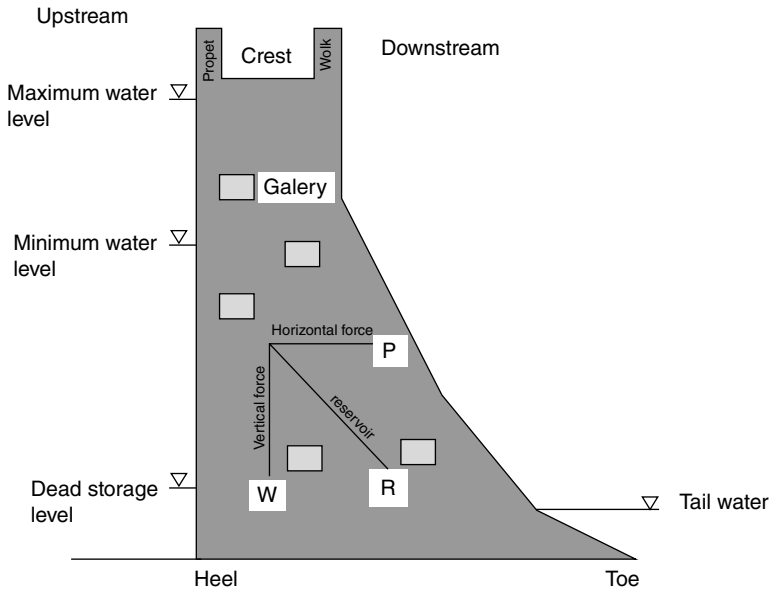


Fig. 15.39 *Gravity Dam*

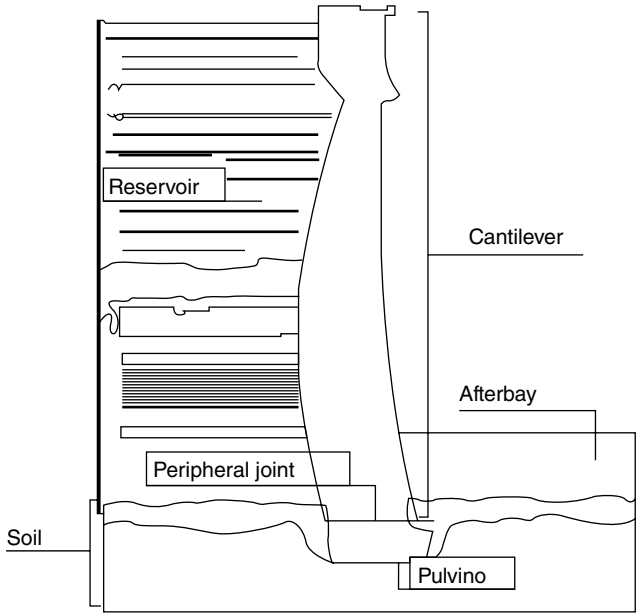


Fig. 15.40 *Arch Dams*

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Embankment dams These are also known as earth-filled dams that are made mainly from natural materials, available in the vicinity. These can be of earth-filled-type dams and rock filled-type dams. Earth-filled dams are constructed mostly from compacted earth, whereas rock-filled (loose rocks) dams are made from dumped and compacted rockfill. The materials are usually excavated or quarried from nearby sites, preferably within the reservoir basin. Figure 15.41 shows a cross section of an embankment dam shaped like a bank or a hill. A typical dam structure consists of a central section (core), made from an impermeable material to stop water leaking through the dam. The core is made of clay/soil, concrete or asphaltic concrete. Rock-filled dams are permeable in nature, possessing a core or an impermeable cover made of reinforced concrete and asphaltic concrete on the upstream face. Embankment dams are preferred for sites with wide valleys. Since they exert not much pressure on their foundations, they can be built on hard rock or softer soils.

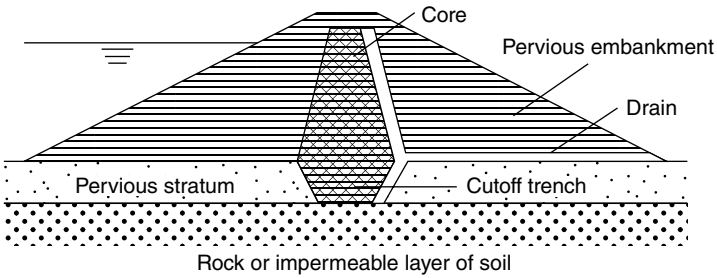


Fig. 15.41 *Embankment Dam*

(ii) Based on function: Dams can be classified as storage dams, diversion dams and detention dams. In a storage dam, water stored is utilized for various purposes such as power generation, irrigation, or water supply. A diversion dam is used to raise water storage and divert it for some other purpose. Detentions dams are constructed to store flood waters.

(iii) Based on material: Dams may be constructed using RCC, cement, earth, rock pieces, etc.

(iv) Based on shape: Dams may be of trapezoidal- or arch-shaped to site the specific structural condition.

Spillways

When the water level in the reservoir area increases dam structure becomes unstable. A spillway is a structure used to spill away the excess water from a dam viz. to release flood water from a dam. It prevents the over toping of the dam structure due to excessive pressure exerted by water. Spillways are classified as controlled type or uncontrolled type. In uncontrolled-type spillway, water is released when a particular level of water is reached in the dam. In controlled type, regulation of water flowing from the spillway is possible.

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Different types of spillways are as follows:

(i) *Overall or gravity spillway*

The gravity spillway shown in Figure 15.42 is used in concrete or masonry dams. In this spillway, water spills and virtually rolls over. In order to reduce the excess energy in water, a bucket is provided at the bottom end of the spillway, which changes the flow direction of water. When the valley is too narrow to accommodate the solid gravity spillway, a chute or trough spillway is used.

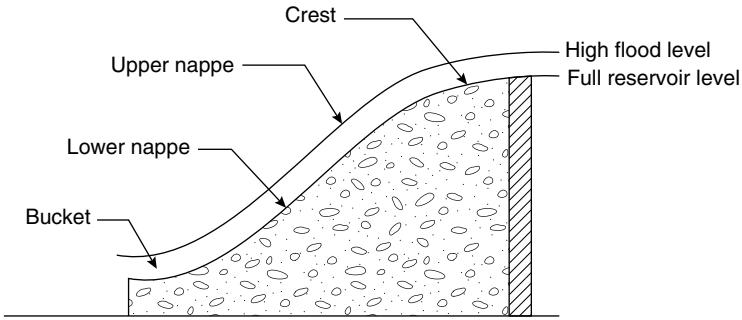


Fig. 15.42 *Gravity Spillway*

(ii) *Side-channel spillway*

When the valley is too narrow or when the dam is non-rigid where the flood water is not expected to flow over the dam, a side-channel spillway is used. In a side-channel spillway (Figure 15.43), water flows parallel to the crest after passing over it.

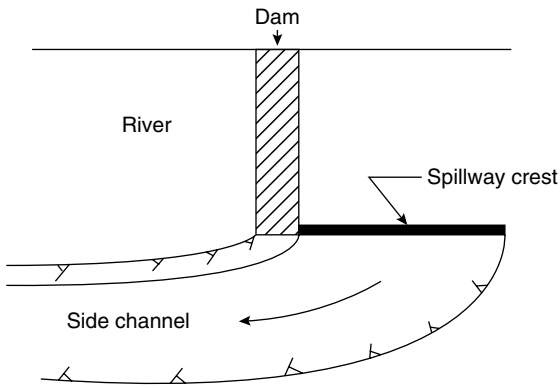


Fig. 15.43 *Side Channel Spillway*

(iii) *Siphon spillway*

In a siphon spillway, the crest is constructed at the full level of the reservoir. Upon reaching the full level, water spills over the crest. A step provided on the downward slope of the dam deflects

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sheet of water and seals the lower end as shown in Figure 15.44. The air entrapped is driven out by the incoming water (priming process).

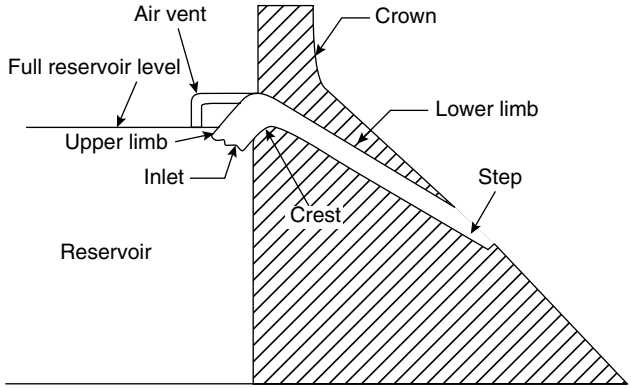


Fig. 15.44 *Siphon Spillway*

(iv) *Saddle spillway*

A saddle spillway is preferred when other types of spillways cannot be used owing to certain reasons. Saddle spillway is constructed at a site where natural depression or a saddle, generally away from the dam, is available at the bottom of the depression at the same level of the free or full reservoir level as shown in Figure 15.45.

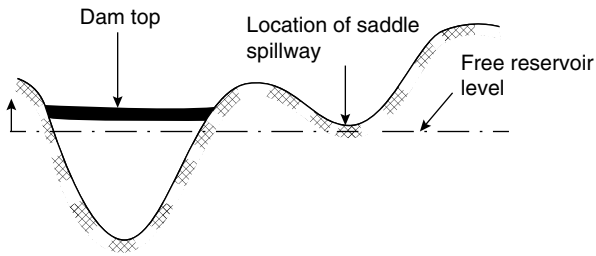


Fig. 15.45 *Saddle Spillway*

15.11 BRIEF DESCRIPTION OF SOME OF THE IMPORTANT HYDEL INSTALLATIONS IN INDIA

India was one of the pioneering countries in establishing hydroelectric power plants. The power plant at Darjeeling and Shimsha (Shivanasamudra) was established in 1898 and 1902, respectively, and it is one of the first in Asia. The installed capacity as on March 31, 2011 was approximately 37567.40 MW, i.e. 21.64 per cent (source: CEA). Out of total hydro-generation, the state sector contribute the highest 27257.00 MW, followed by PSUs with capacity of 8885.40 MW, whereas the private sector accounts for only 1425.00 MW (source: CEA as on March 31, 2011).

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Status of Hydroelectric Potential Development as on 30.11.2013 (In Terms of Installed Capacity – Above 25 MW)

	Identified capacity as per reassessment study		Capacity under operation		Capacity under construction		Capacity under operation + under construction		Capacity yet to be taken up under construction	
Region/state	Total (MW)	Above 25 MW (MW)	(MW)	%	(MW)	(%)	(MW)	(f%)	(MW)	%
NORTHERN										
Jammu & Kashmir	14146	13543	2609.0	19.26	1690.0	12.48	4299.0	31.74	9244.0	68.26
Himachal Pradesh	18820	18540	7594.0	40.96	3948.0	21.29	11542.0	62.25	6998.0	37.75
Punjab	971	971	1206.3	100	206.0	21.22	1412.3	100.00	0.0	0.00
Haryana	64	64	0.0	0	0.0	0.00	0.0	0.00	64.0	100.00
Rajasthan	496	483	411.0	85.09	0.0	0.00	411.0	85.09	72.0	14.91
Uttarakhand	18175	17998	3426.4	19.04	1196.0	6.65	4622.4	25.68	13375.7	74.32
Uttar Pradesh	723	664	501.6	75.54	0.0	0.00	501.6	75.54	162.4	24.46
Sub Total (NR)	53395	52263	15748.3	30.13	7040.0	13.47	22788.3	43.60	29474.8	56.40
WESTERN										
Madhya Pradesh	2243	1970	2395.0	100	400.0	20.30	2795.0	100.00	0.0	0.00
Chhattisgarh	2242	2202	120.0	5.45	0.0	0.00	120.0	5.45	2082.0	94.55
Gujarat	619	590	550.0	93.22	0.0	0.00	550.0	93.22	40.0	6.78
Maharashtra	3769	3314	2487.0	75.05	0.0	0.00	2487.0	75.05	827.0	24.95
Goa	55	55	0.0	0.00	0.0	0.00	0.0	0.00	55.0	100.00

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(*Continued*)

Region/state	Identified capacity as per reassessment study		Capacity under operation		Capacity under construction		Capacity under operation + under construction		Capacity yet to be taken up under construction	
	Total (MW)	Above 25 MW (MW)	(MW)	%	(MW)	(%)	(MW)	(f%)	(MW)	%
Sub-total (WR)	8928	8131	5552.0	68.28	400.0	4.92	5952.0	73.20	2179.0	26.80
SOUTHERN										
Andhra Pradesh	4424	4360	2177.8	49.95	410.0	9.40	2587.8	59.35	1772.3	40.65
Karnataka	6602	6459	3585.4	55.51	0.0	0.00	3585.4	55.51	2873.6	44.49
Kerala	3514	3378	1881.5	55.70	100.0	2.96	1981.5	58.66	1396.5	41.34
Tamilnadu	1918	1693	1782.2	100	0.0	0.00	1782.2	100.00	0.0	0.00
Sub Total (SR)	16458	15890	9426 9	59.33	510.0	3.21	9936.9	62.54	5953.2	37.46
EASTERN										
Jharkhand	753	582	170.0	29.21	0.0	0.00	170.0	29.21	412.0	70.79
Bihar	70	40	0.0		0.0	0.00	0.0		40.0	100.00
Orissa	2999	2981	2027.5	68.01	0.0	0.00	2027.5	68.01	953.5	31.99
West Bengal	2841	2829	272.2	9.62	160.0	5.66	432.2	15.28	2396.8	84.72
Sikkim	4286	4248	669.0	15.75	2322.0	54.66	2991.0	70.41	1257.0	29.59
Sub Total (ER)	10949	10680	3138.7	29.39	2482.0	23.24	5620.7	52.63	5059.3	47.37

(*Continued*)

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(*Continued*)

Region/state	Identified capacity as per reassessment study		Capacity under operation		Capacity under construction		Capacity under operation + under construction		Capacity yet to be taken up under construction	
	Total (MW)	Above 25 MW (MW)	(MW)	%	(MW)	(%)	(MW)	(f%)	(MW)	%
NORTH EASTERN										
Meghalaya	2394	2298	282.0	12.27	40.0	1.74	322.0	14.01	1976.0	85.99
Tripura	15	0	0.0		0.0		0.0		0.0	
Manipur	1784	1761	105.0	5.96	0.0	0.00	105.0	5.96	1656.0	94.04
Assam	680	650	375.0	57.69	0.0	0.00	375.0	57.69	275.0	42.31
Nagaland	1574	1452	75.0	5.17	0.0	0.00	75.0	5.17	1377.0	94.83
Arunachal Pradesh	50328	50064	405.0	0.81	2710.0	5.41	3115.0	6.22	46949.0	93.78
Mizoram	2196	2131	0.0	0.00	60.0	2.82	60.0	2.82	2071.0	97.18
Sub Total (NER)	58971	58356	1242.0	2.13	2810.0	4.82	4052.0	6.94	54304.0	93.06
ALL INDIA	**148701**	**145320**	**35107.8**	**24.16**	**13242.0**	**9.11**	**48349.8**	**33.27**	**96970.2**	**66.73**

Notes:

[1]Does not include pumped storage schemes.

[2]In some states, the total of the capacity developed and balance capacity is different from the potential assessed. This is due to under-assessment of the potential initially.

[3]In addition to this, above 9 PSS (4785.6 MW) are under operation and 2 PSS (1080 MW) are under construction.

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15.12 WATER TURBINES

Turbine is a prime mover wherein a gradual change in momentum of fluid is utilized to set a mobile unit into motion. In other words, turbine is a machine wherein rotary motion is obtained by centrifugal forces resulting in change in the direction of high velocity fluid jet issuing from the nozzle.

A machine that converts hydraulic energy into mechanical energy or vice versa is known as a hydraulic machine. The former is known as turbine, whereas the latter is known as pump. Hydraulic turbine is a device that uses the potential and kinetic energy of water and converts it into mechanical energy. This energy is utilized to run an electric generator coupled with the turbine shaft for the generation of power. Hydraulic turbines are basically selected depending on the head available at the inlet.

15.12.1 Selection of Turbines

The power developed by a hydraulic turbine P is dependent on mass density (of liquid, rotational speed N, diameter of the runner D, working head H and the gravitational acceleration g. It can be expressed mathematically as follows:

$$P = f(D, N, \rho, H, g)$$

From the above expression, it is evident that numerous parameters are available to select the best turbine. In general, based on yearly power output of the plant and the cost of plant installation, selection is made apart from considering the following points:

(i) *Speed*

Turbines with high specific speeds are used where the head is low and output is large. When the rotational speed is low, cost of turbo-generator and powerhouse will be high. In a medium capacity plant, there is no need of choosing a high value of specific speed for high installations, because even with low specific speed, it is possible to achieve high rotational speed.

Rotational speed depends on specific speed. As the rotational speed of an electrical generator coupled directly with the turbine is dependent on the frequency and number of pair of poles, it is necessary to choose a turbine whose rotational speed synchronizes with the generator speed.

(ii) *Efficiency and part load operation*

A turbine that gives the highest overall efficiency at all operating conditions is selected. This means the efficiency at part loads and overloads should not vary too much from the normal operating efficiency. For part or overload conditions, a Deriaz turbine may be used and for low heads, Kaplan turbine will be more useful.

(iii) *Cavitations*

The reaction-type water turbines are prone to cavitations. Before making a selection, it is essential to calculate critical values of cavitation indices to ensure that the turbine works in safe zone.

(iv) *Position of turbine shaft*

Turbines with vertical shaft arrangement are better for large-sized reaction turbines, and horizontal shaft arrangement is preferable for large-sized impulse turbines.

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(v) *Available head and its fluctuation*

As discussed in turbine classification,

(a) Very high heads (350 m and above): Pelton turbine is generally used

(b) High heads (150–350 m): Either Pelton or Francis turbine may be used. For the same working conditions, Francis turbine is more compact and economical than the Pelton turbine which is bigger and heavier.

(c) Medium heads (60–150 m): Francis turbine is opted.

(d) Low heads (below 60 m): Both Kaplan and Francis turbines may be used between 30 and 60 m. For variable loads, Francis turbine is better, whereas Kaplan turbine is generally used for heads less than 30 m. For lower heads up to 15 m, a propeller turbines could be used.

15.12.2 Classification and Types of Water Turbines

15.12.2.1 Classification of Turbines

Hydraulic machines viz, turbines are classified based on the following important factors.

(i) *Classification based on action of water on blades*

Based on the action of water striking the blades or the energy available at the turbine inlet, hydraulic turbines are classified as *impulse* or *reaction* turbines.

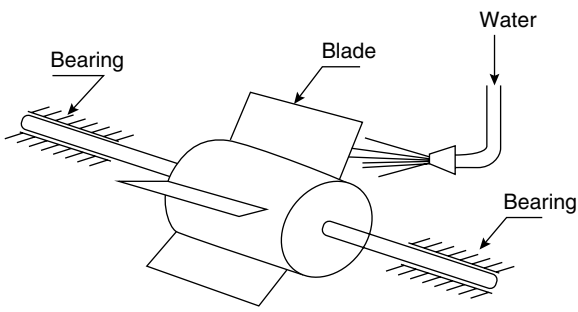


Fig. 15.46 *Impulse Turbine Working Principle*

(a) *Impulse turbine*

If the energy of the fluid entering the rotor is in the form of kinetic energy of jets, the turbine is known as impulse turbine. In an impulse turbine, the available hydraulic energy is converted into kinetic energy by means of nozzles. The high-velocity jet strikes a series of vanes fixed around the rim of the wheel as shown in Figure 15.46. The buckets or vanes change the direction of the jet without changing the pressure. The resulting change in momentum due to change in direction of the jet sets the buckets, and hence the wheel into rotary motion. Thus in an impulse turbine, the pressure of the fluid as it flows over the rotating wheel is constant and is atmospheric. The potential energy of water is completely converted into kinetic energy that is absorbed by transformation into mechanical work by pure impulse action. In an impulse turbine, turbine casing is not filled with water. Examples include Pelton wheel, Girad turbine, Banki turbine, Jonval turbine, etc.

(b) *Reaction turbine*

If the energy of the fluid entering the rotor is in the form of kinetic energy of jets and pressure energy, the turbine is known as reaction turbine.

In a reaction turbine, a portion of hydraulic energy available is transformed into kinetic energy before water enters the turbine runner. However, a substantial portion remains in the form of

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pressure energy. Thus, both velocity and pressure change simultaneously as water glides over the vanes. Figure 15.47 shows the working principle of a reaction turbine. Water from the reservoir is passed through a hollow shaft and hence finally to a hollow disc. The disc has radial openings shaped in the form of nozzles. When water leaves these nozzles, the kinetic energy of water increases, decreasing its pressure energy relative to the disc. The resulting reactive force sets the disc rotation. The disc and the shaft rotate in a direction opposite to the direction of the jet.

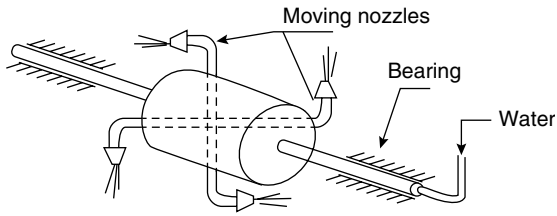


Fig. 15.47 *Reaction Turbine Working Principle*

In reaction turbine, the casing is filled with water and hence the pressure of the fluid changes as water flows over the vanes.

(ii) *Classification based on direction of fluid flow through the runner*

Based on the direction of fluid flow through the runner, turbines are classified as tangential flow turbine, radial flow turbine, axial flow turbine and mixed flow turbine.

(a) *Tangential flow turbine*

If water strikes the runner along the tangential direction, it is known as a tangential flow turbine (Figure 15.48(a)). This type of turbine is also known as peripheral flow turbine, e.g. Pelton turbine.

(b) *Radial flow turbine*

If water strikes the runner along the radial directional, then the turbine is known as a radial flow turbine. Radial flow turbines (Figure 15 48(b)) can be further divided into inward radial flow and outward radial flow turbines.

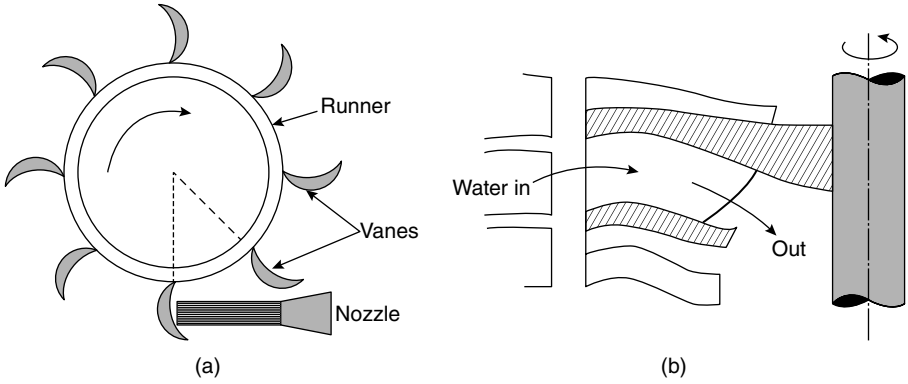


Fig. 15.48 *Classification of Turbines Based on Direction of Flow through Runner: (a) Tangential Flow Turbine and (b) Radial Flow Turbine*

If water flow from the periphery towards the centre of the wheel radially, it is known as an inward radial flow turbine; and if water flows from the centre of the wheel towards outside, it is an outward flow turbine, e.g. *Francis turbine*.

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(c) *Axial flow turbine*

If water flows through the runner parallel to the axis of rotation of the runner, then the turbine is known as axial flow turbine, e.g. *Kaplan turbine*.

(d) *Mixed flow turbine*

If water enters radially and does work on the runner and leaves axially, then the turbine is known as mixed flow turbine, e.g. *Deriaz turbine*. (Figure 15.49)

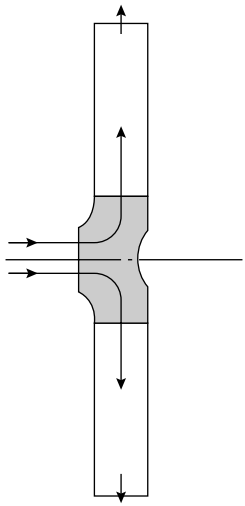


Fig. 15.49 *Classification of Turbines Based on Direction of Flow through Runner–Mixed flow turbine*

(iii) *Classification based on specific speed*

Generally all the turbo machines used in practice are classified based on the specific speeds. The specific speed of a turbine is the speed of a geometrically similar turbine that would develop unit power working under unit head at maximum efficiency.

(a) *Low specific speed turbine*

Such turbines have usually small discharges and use either single or multiple jets. With single jet the specific speed varies between 4 and 30 and with multiple jets, the specific speed may be about 70.

E.g. Pelton Wheel

(b) *Medium specific speed turbine*

Such turbines work under *medium head* and have specific speeds comparatively higher than the low specific speed turbines. The specific speed ranges between 70 and 400

E.g. Francis Turbine

(c) *High specific speed turbine*

Such turbines have usually large discharges and work under low head. They have high specific speeds ranging from 500 to 1,000. These turbines are very sensitive to fluctuation in available

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head. Unless they posses adjustable blades, they have low efficiency at variable heads and inferior part load efficiency

E.g. Kaplan turbine, Deriaz turbine

(iv) *Classification based on head available*

Based on the head of water available, hydraulic turbines are classified as low head, medium head and high head turbines.

(a) *Low head turbines*

These turbines work under *low heads* ranging between 3 and 50 m of water. They are designed for higher discharge rates

E.g. Kaplan Turbine

(b) *Medium head turbines*

These turbines work under *medium heads* ranging between 55 and 400 m. They are designed for medium discharge rates

E.g. Francis Turbine

(c) *High head turbines*

These turbines are designed only for low discharges and work under *high heads* ranging above 300–400 m

E.g. Pelton turbine

15.12.2.2 Pelton Turbines

Pelton turbine, named after Mr L.A. Pelton from the United States, is a high-head *impulse turbine*. In this turbine, water carried from the penstock which enters the nozzles, impugning out in the form of a fluid jet. The potential energy of water is converted into kinetic energy which is used to run the turbine runner (Figure 15.50).

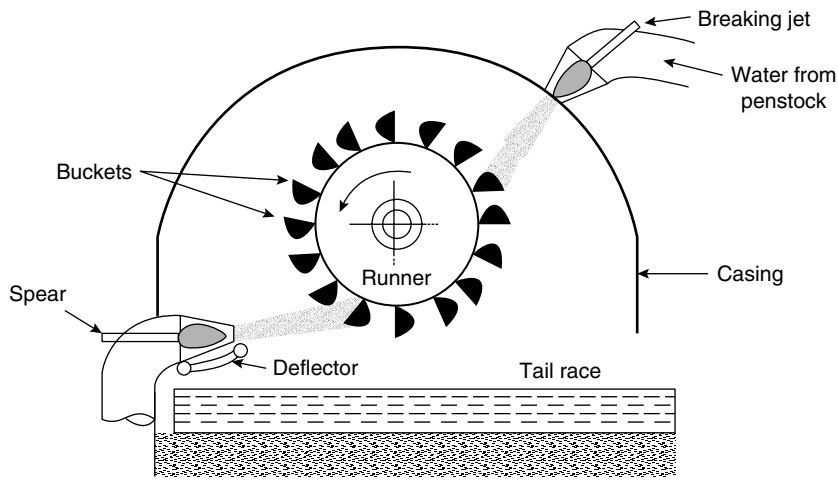


Fig. 15.50 *Components of a Pelton Wheel*

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(i) *Components of a pelton wheel*

The major components of Pelton wheel are as follows: (a) guide mechanism, (b) runner and buckets, (c) casing and (d) breaking jet or nozzle.

(a) *Guide mechanism*

Guide mechanism controls the amount of water striking the vanes or buckets on the periphery of the runner by using a spear operated by a governor. The quantity of water flowing through the nozzle and striking the buckets is reduced by pushing the spear into the nozzle, whereas the quantity of water striking the buckets can be increased by pulling the spear from the nozzle. The spear maintains a constant speed of the runner irrespective of head variation. Due to sudden drop in available head, sometimes the main nozzle has to be closed. During this period, a bypass nozzle is used, which ensures safe passage of water without striking the buckets. This arrangement eliminates the possibility of bursting of pipes due to build up pressure because of nozzle closure.

In modern turbines, a *deflector* is used to serve the above purpose. The deflector is a plate connected to spear and is placed between the nozzle tip and buckets as shown in Figure 15.51. The deflector is operated by a governor whenever there is sudden reduction in load. Due to the governor action, water gets deflected from the nozzle, and hence does not strike the blades or buckets.

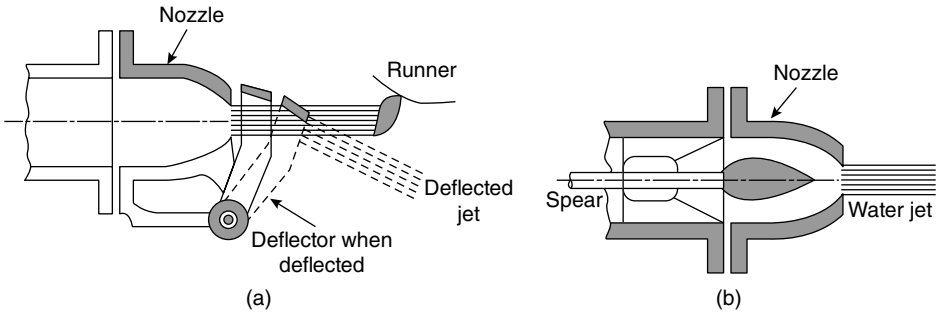


Fig. 15.51 *Guide Mechanism: (a) Deflector and (b) Spear Arrangement*

(b) *Runner and buckets*

Runner is a circular disc with a number of buckets or vanes evenly spaced on the wheel periphery. Runner is mounted on the turbine shaft and rotates along with the shaft. The *buckets* may be cast either integrally with the runner or bolted separately, the latter being more economical. Runner is made of *cast Iron, cast steel or stainless steel*.

Each bucket is divided vertically into two halves by a *splitter* in the form of a double hemispherical cup. *The function of the splitter is to divide the jet into two parts without exerting shock*. The jet gets deflected into two parts moving sideways in the opposite direction. The bucket centre cannot be made exactly hemispherical because of manufacturing difficulty and to avoid wear and tear due to abrasive action of sand and other particles carried by water.

The angle of deflection is limited to 165–170° to prevent splashing of the fluid, leaving one bucket and consequently falling back on the succeeding buckets. The rear end of the bucket is so designed as to prevent any obstruction to the passage of water to the preceding buckets (Figure 15.52).

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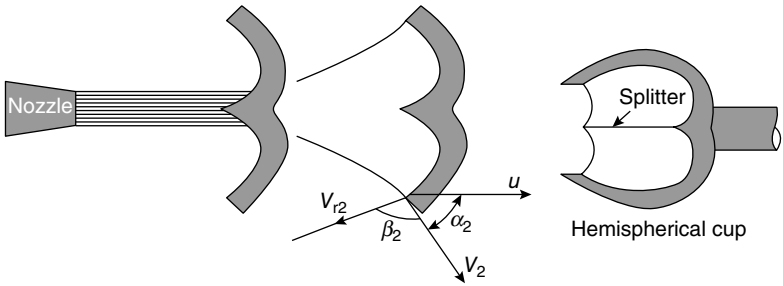


Fig. 15.52 *Bucket of an Impulse Turbine*

(c) *Casing*

Casing is a metallic cover that prevents the splashing of water and has no influence on the hydraulic work. Water after striking the buckets is guided by means of a casing towards the tailrace. The casing may be cast or fabricated and should be strong enough to take impact of jet in case of over speed. It is made of *cast iron* or *fabricated steel plates*. The turbine casing is shown in Figure 15.50.

(d) *Breaking jet of nozzle*

The heavily built runner keeps on rotating even if the turbine inlet valve is closed for a considerable time due to the effect of inertia. *In order to stop the runner within a short time interval, a breaking jet is used, which directs a jet of water on the back of the buckets*, as shown in Figure 15.50. Due to this action, a resisting force is set up, which brings down the runner speed, ultimately stopping it.

(ii) *Force, power and efficiency*

In a Pelton wheel, the jet strikes the bucket and gets deflected by the splitter into two parts. The force exerted by the jet for an ideal case can be calculated by drawing the inlet and outlet velocity triangles (Figure 15.53). As the runner diameter is same, peripheral velocity of wheel remains the same. Assuming no friction over the blades, the relative velocity of jet the remains constant.

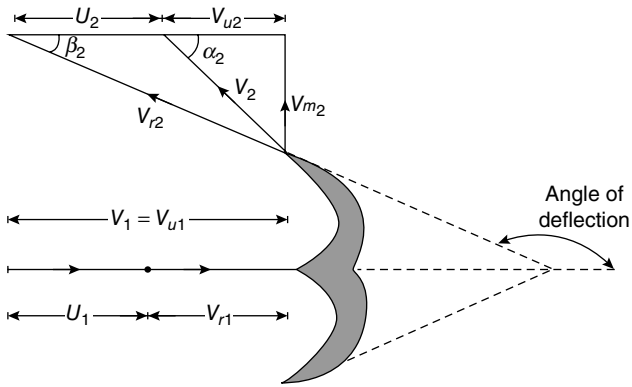


Fig. 15.53 *Inlet and Outlet Velocity Triangles*

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Thus, $V_{r1} = V_{r2} = V_r$ = relative velocity of jet, m/s
Let α_1 = guide blade angle at the inlet
= angle between V_1 and U_1
α_2 = guide blade angle at the outlet
= angle between V_2 and U_2
β_1 = vane angle or blade angle at the inlet
= angle between V_{r1} and U_1
β_2 = vane angle or blade angle at the outlet
= angle between V_{r2} and U_2
V_u = horizontal component of velocity V, m/s
= velocity of whirl, m/s
V_m = vertical component of velocity V, (meridional) m/s
= velocity of flow

Assuming $\alpha_1 = 0$ and $\beta_1 = 0$, we can construct the inlet and outlet velocity triangles.
Inlet velocity of jet

$$V_1 = C_V \sqrt{2gH}$$

Mean peripheral speed

$$U_1 = U_2 = \pi dN$$

where N = revolution per second of runner
d = diameter of runner
H = net head on the turbine
C_V = coefficient of velocity
From the inlet velocity triangle,

$$V_1 = V_{u1}$$
$$V_{r1} = V_1 - U$$

From the outlet velocity triangle, (neglecting friction)

$$V_{r1} = V_{r2} = V_1 - U$$

Velocity of whirl at the outlet

$$V_{u2} = V_{r2} \cos\beta_2 - U$$

Force exerted by the jet along the direction of motion of the jet can be calculated by assuming that mass of water striking the jet is based on absolute velocity of jet. This is because the vanes are moving at uniform speed U and are arranged in series on the periphery of the runner. Therefore,

$$F = \rho Q\ (V_{u1}(V_{u2})) \tag{1}$$

In the above equation, positive sign is used when α_2 is acute and negative sign is used when α_2 is obtuse.

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But $U = \omega r = U_1 = U_2$
Torque on the wheel is given by

$$T = F \times r$$

$$T = (Q\,(V_{u1} + V_{u2})\text{r (assuming } \alpha_2 \text{ is acute)} \tag{2}$$

Power output of the wheel

$$P = T.\omega$$

$$= (Q\,(V_{u1} + V_{u2})\,\omega r$$

$$P = (Q\,(V_{u1} + V_{u2})\,U \tag{3}$$

Wheel efficiency or hydraulic efficiency

$$\eta_h = \frac{\text{Output}}{\text{Input}}$$

$$= \frac{\rho Q V_{u1} + V_{u2} U}{\frac{1}{2}\rho Q V_1^2\,(\text{Kinetic energy of jet})}$$

$$\eta_h = \frac{2V_{u1} + V_{u2} U}{V_1^2} = \frac{2V_{w1} + V_{w2} U}{V_1^2} \tag{4}$$

(a) *Effect of friction*

In practical case, the relative velocity at the outlet is slightly less than the relative velocity at the inlet due to frictional loss over the inner surface of the bucket. Some velocity is also lost due to the jet striking over the splitter.

Hence $V_{r2} = k\,V_{r1}$
where k = a coefficient which takes into account of friction loss.
Thus, $V_{u2} = V_{r2}\cos\beta_2 - U_2$

$$= \text{kVr}_1 \cos\beta_2 - U_2$$

$$V_{u1} = V_{r1} + U_1$$

Hence Eqn. (1) can be modified as follows:

$$F = (Q\,[V_{r1} + U_1) + (kV_{r1}\cos\beta_2 - U_2)]$$

As $U_1 = U_2 = \omega r$,

$$F = (QV_{r1}\,(1 + k\cos\beta_2) \tag{5}$$

or $F = (Q\,(V_1 - U)\,(1 + k\cos\beta_2)$ (6)

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Torque on the wheel is given by

$$T = F \times r$$

$$T = \rho Q\,(V_1 - U)\,(1 + k\cos\beta_2)\,r \tag{7}$$

Power output of the wheel

$$P = T_\omega$$

$$P = \rho Q\,(V_1 - U)\,(1 + k\cos\beta_2)\,r \tag{8}$$

Wheel efficiency or hydraulic efficiency

$$\eta_h = \frac{\text{Output}}{\text{Input}}$$

$$= \frac{\rho Q\left(V_1 - U\right)\left(1 + k\cos\beta_2\right)U}{\frac{1}{2}\rho Q V_1^2}$$

$$\eta_h = \frac{2\left(V_1 - U\right)\left(1 + k\cos\beta_2\right)U}{V_1^2} \tag{9}$$

(b) *Maximum hydraulic efficiency*

Hydraulic efficiency is maximum when $\frac{d}{dU}(\eta_h) = 0$ for fixed value of absolute velocity at the inlet.

From Eqn. (9),

i.e., $\boldsymbol{U} = \frac{V_1}{2}$ (10)

Substituting Eqn. (10) in Eqn. (9), we obtain

$$(\eta_h)max = \frac{\left(1 + k\cos\beta_2\right)}{2} \tag{11}$$

(iii) *Design parameters of pelton wheel*

Some of the important design parameters of a Pelton wheel are discussed below:

(a) *Jet diameter (d)*

Discharge of jet at the throat is given by,

$$Q = d^2 V_1$$

But absolute velocity of jet is given by

$$V_1 = C_v$$

Where C_v = Coefficient velocity ranging from 0.98 to 0.99

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(b) *Nozzle diameter (d_n)*

Discharge through the nozzle at the outlet diameter of nozzle (d_n) is given by

$$Q = a_n V_n$$

But velocity of jet at nozzle outlet is

$$V_n = C_{vn}$$

where C_{vn} = Coefficient of velocity for the nozzle which ranges from 0.81 to 0.83.

(c) *Runner diameter, D*

Mean diameter of the Pelton wheel runner is given by
Where U = circumferential velocity of bucket

$$= \varphi\sqrt{2gH}$$

φ = speed ratio ranging from 0.44 to 0.48
N = runner speed in rpm.

(d) *Number of jets*

Ordinary Pelton wheel has one nozzle or jet. However, to produce more power, a number of nozzles are used. The nozzles are spaced evenly around the same runner so that one jet striking the runner does not interfere with another jet. Theoretically, six nozzles can be used with one Pelton wheel. However, *practically not more than two jets per runner are used for a vertical runner and not more than four jets per runner are used for a horizontal runner*.

(e) *Jet ratio (J_r)*

It is the ratio of mean diameter of the runner to the minimum diameter of the jet vena contracta
i.e., J_r = (ranges between 14 and 16)

(f) *Bucket dimensions*

Width of the bucket,

$$W = 2.8\text{–}4.0 \text{ times } d$$

Length of the bucket

$$L = 2.4\text{–}2.8 \text{ times } d$$

Depth of bucket,

$$T = 0.6\text{–}0.95 \text{ times } d$$

(g) *Minimum number of buckets*

According to Dr Taygun's empirical relation,

$$Z = 0.5\, J_r + 15$$

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(h) *Angle of deflection*

Angle of deflection usually ranges between 165° and 170°.

Hence, vane angle at the outlet, β_2 = (180 – angle of deflection)

(iv) *Characteristics of pelton wheel*

Turbines are always designed to operate under certain conditions so as to run at maximum possible efficiency. However, in practice, there is much deviation from these conditions. It is therefore necessary to study the variation of head, speed, flow rate, power output, efficiency, etc. The graphical representation of combination of any two variables is known as *performance characteristics* of the turbine. Among these variables, H, Q and N are independently variable and P, T and η are dependently variable. Depending on the type of variable, characteristic curves may be divided into two types (Figure 15.54).

(a) *Main characteristic curves (H = constant)*

These are the curves obtained by keeping constant head and gate opening and varying speed. The overall efficiency for each values of speed is determined. From these readings, the values of unit speed (N_u), unit power (P_u) and unit discharge (Q_u) are determined. Taking N_u along Y-axis, the variations of Q_u, P_u, P and η_o are plotted on the ordinate. For different gate openings, the procedure is repeated, and the following curves are drawn, viz. Q_u *v/s* N_u, P_u *v/s* N_u and η_0 *v/s* N_u

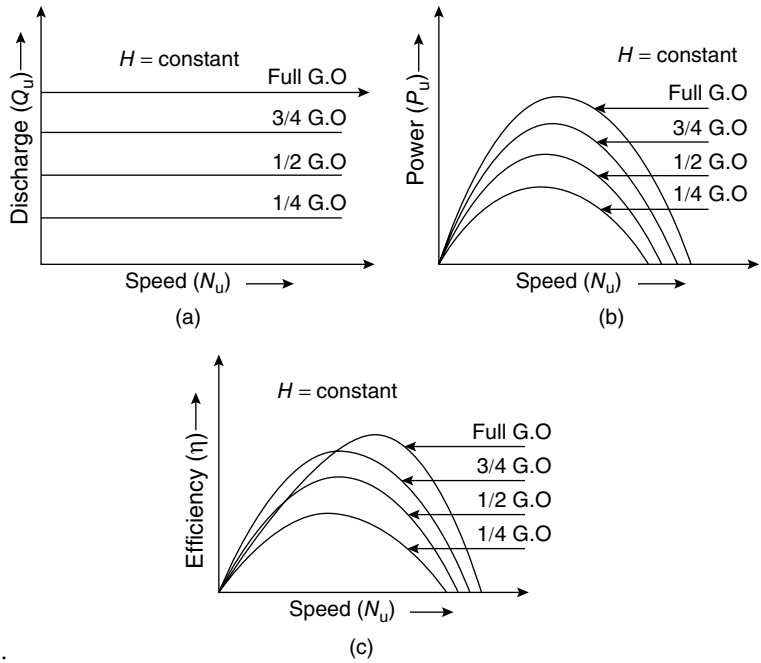


Fig. 15.54 *(a) Discharge v/s Speed, (b) Power/Torque v/s Speed and (c) Efficiency v/s Speed*

(b) *Operating characteristics curves (N = constant)*

These are the curves obtained by keeping both speed and head constant. As the quantity of discharge is varied, corresponding changes in power and efficiency are recorded. Figure 15.55

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shows operating characteristic curves. Due to frictional losses, both the curves shift a little to the right side from the origin along the *X*-axis.

(c) *Iso efficiency curves*

These are the curves obtained from *speed v/s efficiency* and *speed v/s discharge* curves. These are also known as *constant efficiency curves* or *Muschel curves*. By drawing a horizontal line for a given η_0, two values of speeds are obtained from the η_0 v/s N curve. Similarly for a given efficiency, two values of discharge can be obtained for a pre-set valve opening. For different set of gate valve openings, a series of curves are drawn as shown in Figure 15.56. The points having same efficiencies are joined. These curves are helpful in finding the constant efficiency zone and for predicting the turbine performance at various efficiencies.

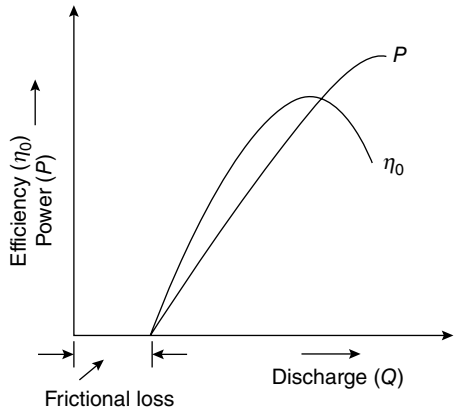


Fig. 15.55 *Operating Characteristic Curves*

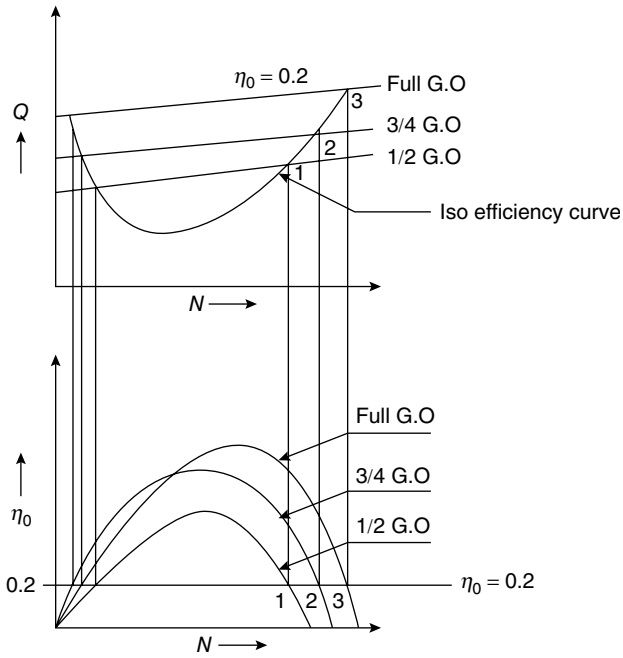


Fig. 15.56 *Iso Efficiency Curves*

15.12.2.3 Reaction Hydraulic Turbines

In reaction turbines, the energy of the fluid is in the form of kinetic energy and pressure energy. A part of head available is transformed into kinetic energy and the rest is available as pressure energy. In these turbines, the casing is always filled with water or working fluid. Therefore, pressure acting

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on both sides of the blades is not constant. Based on the direction of flow of the working fluid, the reaction turbines are classified as *inward flow reaction turbine* and *outward flow reaction turbine*.

If the water flows from the periphery towards the centre of the wheel radially, it is known as an *inward flow radial reaction turbine*. If the water flows from the centre of the wheel towards the periphery of the wheel, then the turbine is known as an *outward flow radial reaction turbine*.

Francis turbine

Francis turbine is an inward flow reaction turbine designed and developed by an American Engineer James B. Francis. In this turbine, water flows from outside of wheel periphery towards the centre, resulting in radial discharge at the outlet.

Earlier Francis turbines were purely radial flow type, but modern Francis turbines are mixed flow type in which water enters the runner radially and leaves axially at the centre. The latter type of turbine provides a large discharge area with the prescribed diameter of the runner.

Francis turbines are classified mainly into two types: open type and closed type.

1. *Open type*

In this turbine, the runner is not covered with casings but is covered by a concrete chamber. These turbines are used for low heads ranging between 6 and 10 m.

2. *Closed type*

In this type of turbine, the runner is covered by spiral casing of concrete or metal. Due to this arrangement, the surroundings are protected from splashing of water. The casing helps in directing water evenly over the guide blades.

(i) *Components of francis turbine*

Different components of a Francis turbine are as discussed below (Figure 15.57).

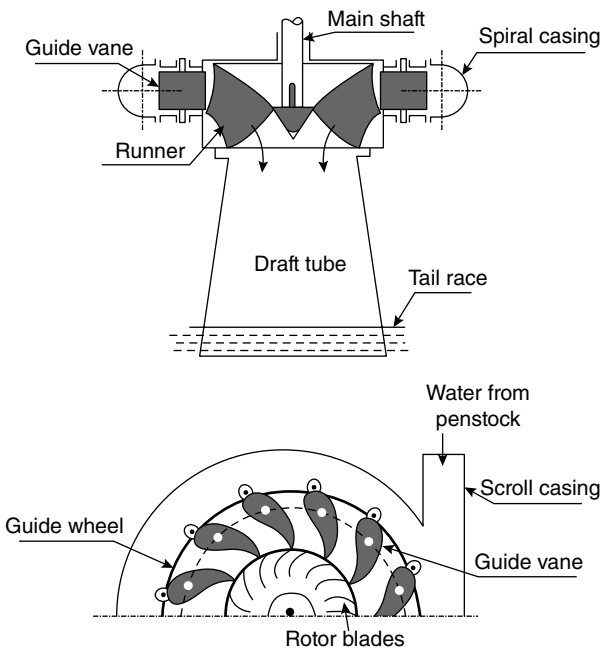


Fig. 15.57 *Components of Francis Turbine*

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(a) *Penstock*

It is a large diameter pipe that carries water to the runner from the head race. Depending on the amount of power generated, either a single penstock or multiple penstocks are used.

(b) *Scroll casing*

Scroll casing is also known as *spiral casing*. Casing directs water coming from the penstock towards the runner without the formation of eddies. Due to this action, loss due to formation of eddies reduces, and hence efficiency increases. The cross-sectional area of the casing reduces gradually from inlet to outlet ensuring uniform distribution of water around the guide vanes.

(c) *Runner*

The guide vanes impart a tangential velocity and hence angular momentum to water, before entering into the runner. The flow through the runner is a combination of radial and tangential flows and takes place from the periphery towards the centre. For high specific speed, the amount of water required is more, and hence runner is made wide and vice versa. The runner may be cast in one piece or may be separately welded. *Cast iron* is used for small power output and *cast steel* for higher output. *Stainless steel, bronze* can be used if water is chemically impure.

(d) *Guide mechanism*

The mechanism consists of guide vanes that are fixed between two concentric rings in the form of a guide wheel. The *aero-foil* cross section of guide vanes enables smooth entry of water without forming eddies ensuring minimum friction losses. The vanes can be made adjustable so that flow through them can be regulated as per the needs. The guide vanes are usually made of *Cast steel* (Figure 15.58).

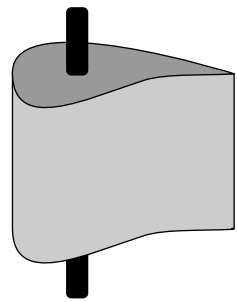


Fig. 15.58 *Guide Mechanism*

(e) *Draft tube*

As the pressure at the outlet of the turbine is lesser than atmospheric, a conduit of increasing cross section is used to discharge water to the tail race. This tube is known as *draft tube* that is immersed minimum 1 m below the lowest tail race level.

Kaplan turbine

If water enters the runner parallel to the axis of rotation of the rotor, then the turbine is known as an axial flow turbine. The axial flow turbines can be further divided into two types.

1. *Propeller turbine*

If the runner of the turbine is propeller shaped, it is known as a *propeller turbine*. The runner blades are fixed in the propeller turbine, and hence are not adjustable.

2. *Kaplan turbine*

If the runner is propeller shaped and pitch of the propeller blades are adjustable, then constant speed can be maintained under variable load conditions. This type of turbine having adjustable propeller blades is known as *Kaplan turbine*.

Figure 15.59 shows a schematic diagram of Kaplan turbine, in which water enters from penstock into the spiral casing. The guide vanes direct water towards the runner vanes without shock or formation of eddies. Between the guide vanes and the runner, the fluid gets deflected by 90° so that flow is parallel to the axis of rotation of the runner, which is known as *axial flow*.

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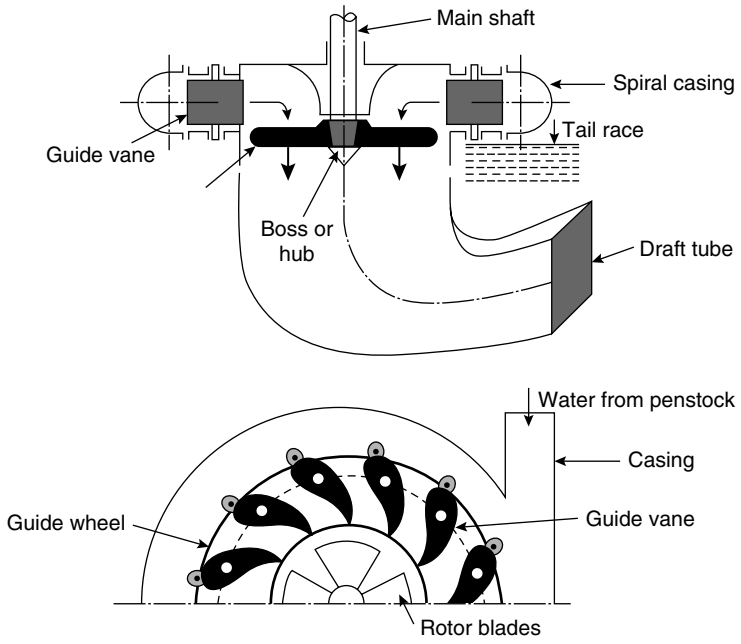


Fig. 15.59 *Kaplan Turbine*

As the flow direction changes from radial direction to axial direction, neglecting losses, the flow pattern can be approximated as *free vortex flow*, i.e.

$$V_u r = \text{constant}$$

The guide vanes impart whirl component to flow and runner vanes nullify this effect making flow purely axial.

All parts, viz. spiral casing, guide mechanism and draft tube of axial flow turbines are similar to Francis turbine. However, runner of the turbine is different from Francis turbine runner. As flow is assumed to be vortex free, the whirl velocity is inversely proportional to runner radius and blade velocity is proportional to radius. To overcome this different relationship, the blades are twisted with axis angle greater at the tip than at the root. As compared to Francis turbine runner blades (16–24 numbers), Kaplan turbine uses only three to eight blades. Due to this, the contact surface with water is less, which reduces frictional resistance and losses.

(i) *Design parameter of kaplan turbine*

(a) *Flow ratio*

Flow ratio is given by

$$C_m = \frac{V_{m1}}{\sqrt{2gH}} \tag{1}$$

where C_m = flow ratio which ranges from 0.15 to 0.30

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$V_{m1} = V_{f1}$ = flow velocity

(b) *Speed ratio*

Speed ratio is given by

$$\varphi = \frac{U_1}{\sqrt{2gH}} \tag{2}$$

where φ = Flow ratio which ranges from 0.6 to 0.9

(c) *Peripheral velocity of runner*

Peripheral velocity of the wheel

$$U = U_1 = U_2$$

$$= \pi D_{or} N \tag{3}$$

where D_{or} = runner diameter at outlet
N = speed in revolution per second

(d) *Head on the turbine without energy loss*

Neglecting energy loss, head on the turbine is given by

$$H = (V_{u1} \pm V_{u2}) \text{ U} + \frac{V_2^2}{2g} \tag{4}$$

(e) *Discharge through the runner*

Discharge through the runner is given by

$$Q = \frac{\pi}{4} (D_{or}{}^2 - D_b{}^2) V_{m1} \tag{5}$$

where D_b = diameter of hub or boss

(f) *Velocity of flow*

In a Kaplan turbine, the velocity of flow at inlet and outlet remain the same.

$$V_{m1} = V_{m2} \tag{6}$$

(g) *Power developed*

Power developed by the turbine is given by

$$P = (V_{u1} \pm V_{u2}) U \tag{7}$$

The flow or discharge at the outlet may be radial in which case $V_{u2} = 0$, $\alpha_2 = 90°$. If discharge at inlet is radial, then $V_{u1} = 0$, $\alpha_1 = 90°$.

(ii) *Draft tubes*

Water, after passing through the runner is discharged through a gradually expanding tube called draft tube. The free end of the draft tube is submerged deep into the water. Thus, the entire water passage from the head race to tail race is completely closed, and hence does not come in contact with the atmospheric air. *The draft tube is a conduit connecting runner exit to the tail race where water after doing work gets discharged*. It is a welded steel plate pipe or a concrete tunnel with gradually increasing cross sectional area at the outlet.

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Functions of a draft tube

(i) It reduces the velocity of discharged water and hence minimizes the loss of kinetic energy at the outlet.

(ii) It permits a negative (suction) head at the turbine outlet and thus enables turbine installation above the tail race without any loss of head.
If water is discharged freely from the runner, the working head is equal to gross head. If an air-tight draft tube is connected to the runner outlet, the working head is increased by an amount equal to the height of the runner outlet above the tail race.

(iii) The turbine can be installed at tail race level, above the tail race level, or below the tail race level.

(iv) Possibility of cavitation can be avoided if the turbine is installed below the tail race level due to the absence of negative head.

(v) As the net working head on the turbine increases, turbine output also increases resulting in increased efficiency.

Different types of draft tubes

1. *Simple elbow tube*

In order to offset the cost of excavation, viz. hard rocks, the vertical length of the tube is reduced. To recover the kinetic energy, the exit diameter is made as large as possible with a larger cone angle, and the draft tube is bent to keep its normal length. This type of draft tube is known as a *simple elbow tube*. Efficiency of the tube is as low as 60 per cent. A simple elbow tube is shown in Figure 15.60(a).

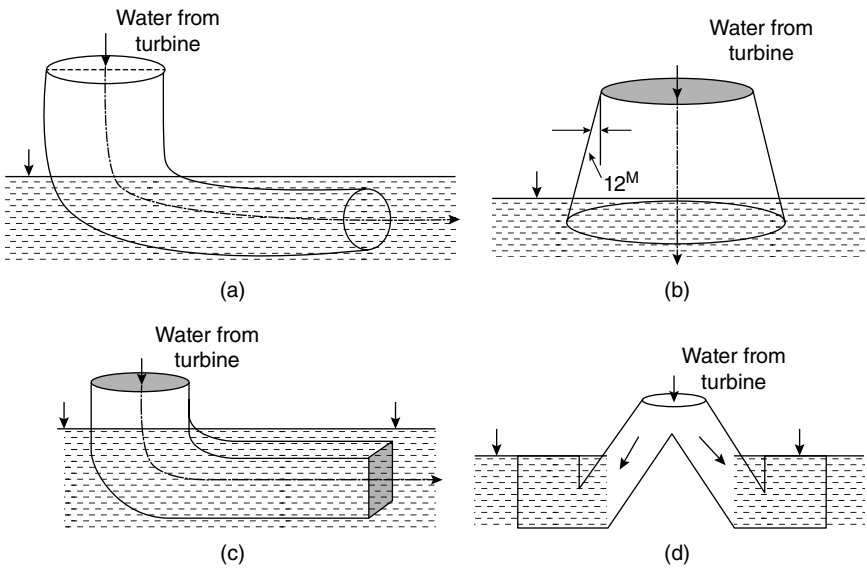


Fig. 15.60 *Different Types of Draft Tubes: (a) Simple Elbow Tube, (b) Straight Divergent tube, (C) Elbow Tube Having Circular Inlet and Rectangular Outlet and (d) Moody's Spreading Tube*

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2. *Straight divergent tube*

The tube resembles the shape of a frustum of a cone as shown in Figure 15.60(b). It is used when the turbine shaft is vertical and specific speed is low (e.g., Francis turbine). The cone angle is limited to about 8° to avoid losses due to separation and formation of vortices. The tube is designed to discharge sufficient water even if tail race level is low. The maximum efficiency is around 86 per cent.

3. *Elbow tube with a circular inlet and rectangular outlet*

This type of tube turns water from vertical to the horizontal direction with high efficiency and minimum depth of excavation. The transition from circular section to rectangular section takes place at the bend. The horizontal portion is inclined upwards to lead water gradually and to prevent the entry of air at inlet. The exit end is totally immersed in water. In order to avoid whirl of water at exit, one or two piers are constructed in the bend of the draft tube (Figure 15.60(c)).

4. *Moody's spreading tube*

It is a mouthed draft tube having a solid conical core covering the entire centre portion of the tube and allowing large exit area without excessive length. The central cone arrangement reduces the whirling component of water and eddies and hence reduces loss. The efficiency of the draft tube is about 85 per cent. A typical Moody's tube is shown in Figure 15.60(d).

(iii) *Cavitation in turbines*

According to the Bernoulli's theorem, if the velocity of flow increases, pressure should decrease. In case of liquids, this decreased pressure cannot be lesser than the vapour pressure at that place. Whenever the liquid pressure in a turbine falls below the evaporation pressure, the liquid starts boiling and results in the formation of bubbles of vapour. These bubbles are carried by the stream to a higher pressure zone where the vapours condense and bubbles suddenly collapse. This results in the formation of a void and stream of liquid rushes to the centre of the cavity from all around. This gives rise to localized pressure rise of about 7,000 atmospheres. The same cycle repeats so many times in a fraction that it causes pitting on the metallic surface of runner blades or draft tube.

This *phenomenon of formation of vapour bubbles and sudden collapsing of it at the high-pressure zone resulting in pitting of turbine blades is known as cavitation*.

(a) *Factors influencing cavitation*

1. Absolute or barometric pressure above the mean sea level at the turbine location
2. Vapour pressure that depends on the temperature of flowing water
3. The height of runner outlet above the tail race or suction head
4. Dynamic suction head and absolute velocity of water at runner exit

(b) *Methods to avoid cavitation*

Cavitation can be avoided by the following methods:

1. Turbine should not be installed below the tail race level.
2. The pressure of the fluid should not be allowed to fall below its vapour pressure.
3. Providing a cavitation free runner.
4. Materials such as *CI*, *CS*, *SS*, bronze, aluminium, zinc alloys, chrome nickel and stainless steels can be used.

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As rough surfaces interrupt smooth and streamline flow causing vortices and likelihood of cavitation, highly polished surfaces should be used.

(c) *Effects of cavitation*

1. Damaging of metallic surfaces and formation of cavities on turbine blades
2. Noise and vibration due to sudden collapsing of bubbles and vapours
3. Failure of material by fatigue and corrosion
4. Decrease in turbine efficiency due to reduced resultant torque produced by the flow of water

Professor D. Thoma has suggested a cavitation factor (where the turbine can work without cavitation.

If H_b = barometric pressure head in *m* of water
H_a = average atmospheric pressure in *m* of water
H_v = vapour pressure in *m* of water
H_s = suction pressure head in *m* of water
= height of runner outlet above tail race
H = net head on the turbine in *m* of water.
σ_c = critical cavitation factor

Mathematically,

$$\sigma_c = \frac{H_b - H_s}{H}$$

$$\sigma_c = \frac{(H_a - H_v) - H_s}{H} \quad (1)$$

(iv) *Turbine performance curves*

Turbines are usually designed for specific conditions of head, speed and output. However, the turbines may have to operate under different conditions for which they are designed. In order to determine and collect complete information about the performance of the turbine over a wide range of operating conditions, tests are conducted on an actual turbine at site or on a small-scale turbine model at the research laboratory. These experimental results are depicted in the form of curves on a graph, which are known as *performance characteristics or curves*. The performance and behaviour of the turbine is studied using two types of curves.

1. *Main characteristic curves*

To draw these curves, *head* and *opening* of the turbine are maintained *constant* and *speed* is *varied* by allowing variable amount of water to flow through fixed gate openings. The brake power *P* is measured mechanically by dynamometer. The basic data are then used to calculate the overall efficiency and unit quantities. These are plotted against unit speed as abscissa. Figure 15.61 shows typical main characteristic curves of Francis and Kaplan turbines.

The following information can be gathered from the above curves:

- The main characteristics of *Francis and Kaplan turbines* are identical except for the discharge characteristics.
- Separate curve exists for each gate opening.

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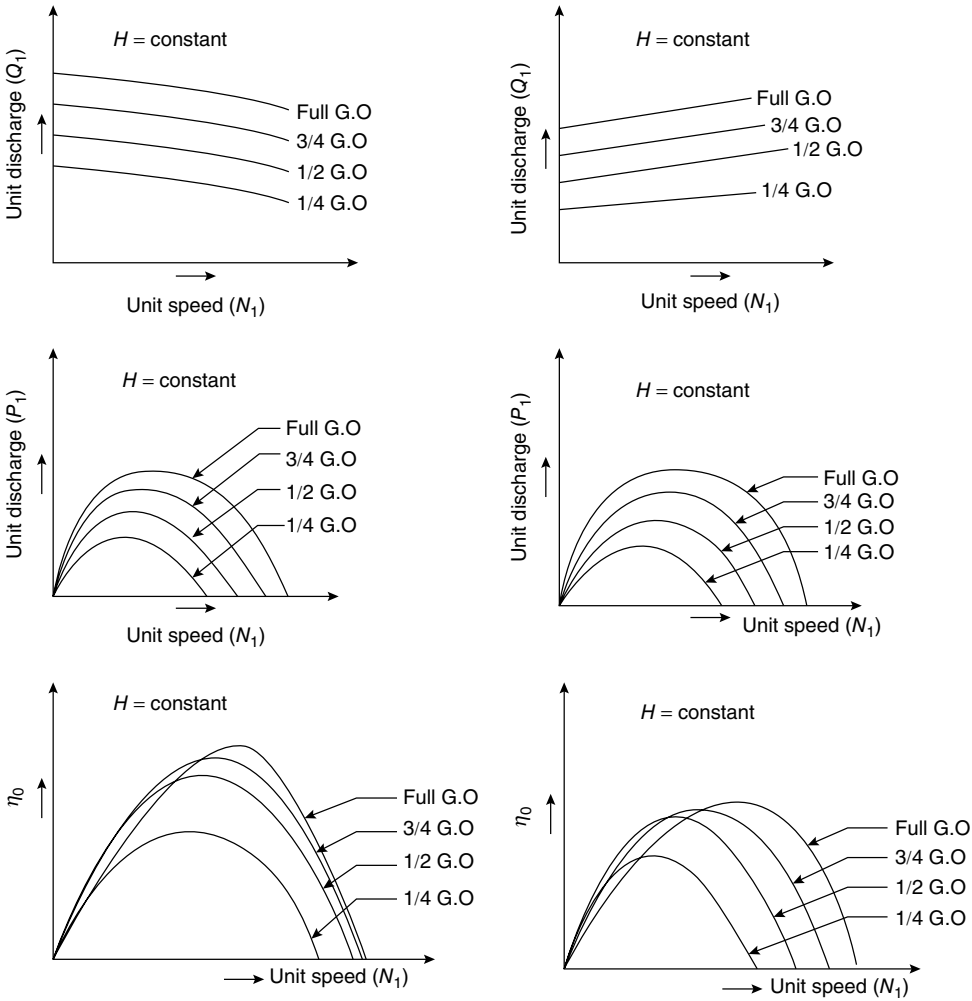


Fig. 15.61 *Main Characteristic Curves*

- For Kaplan turbine, the discharge curve rises as the speed increases. However for *Francis turbine*, the unit discharge drops with increase in speed due to increased centrifugal head which opposes the radially incoming discharge.
- Power and efficiency curves are parabolic in nature with maximum value occurring at a particular speed.

2. *Operating characteristic curves*

These curves are also known as *constant speed curves*. To draw these curves, speed is kept constant, whereas discharge Q and head H may be varied according to their availability. The brake power P is measured mechanically by a dynamometer. Overall efficiency ηo is then calculated from the measured values of discharge, head and power. Results are plotted on a graph with η_o

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and P as abscissa and Q as ordinate. Due to frictional losses, the curves do not start from the origin but are shifted slightly right side along the ordinate.

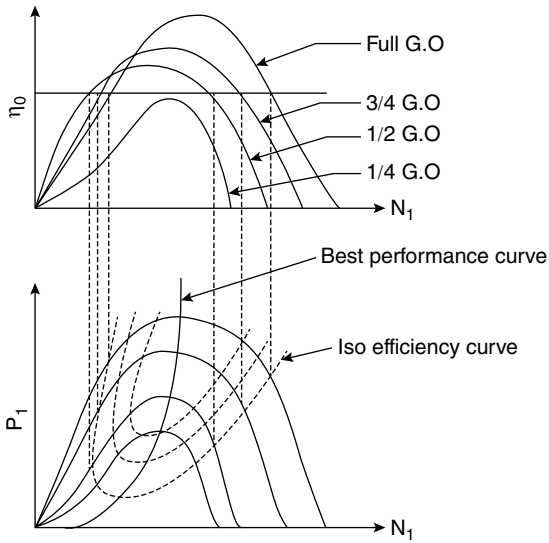


Fig. 15.62 *Muschel Curves*

3. *Constant efficiency curves*

The constant efficiency curves are also called as *iso efficiency curves* or *Muschel curves*. Data for drawing these curves are obtained from the main characteristic curves P_1 v/s N_1 and η_o v/s N_1 as shown in Figure 15.62.

For a particular efficiency say 50 per cent, a horizontal line is drawn which intersects the curves for different gate openings. From the points of intersection, the corresponding unit speed values are obtained. These values are transferred to the main curve P_1 v/s N_1 for the corresponding gate openings. Points of the same efficiency are joined by a smooth curve to get iso efficiency curves. A curve for best performance is obtained when the peak points of various iso efficiency curves are joined. These curves are useful in locating the regions of maximum efficiency where the turbine would operate.

15.12.3 Governing of Turbines

The speed of the generator always should be maintained constant irrespective of variations in the load on the turbine.

p = number of poles of the generator then, constant speed of the generator is given by

$$N = \frac{120 f}{p}$$

where N is in rpm.

As the turbine is coupled to generator, governor speed can be maintained constant only if speed of the turbine runner is maintained constant also known as synchronous speed

In order to maintain constant generator speed governing mechanisms are used to control the speed of the turbine. The governor regulates the speed by controlling flow rats of water. A centrifugal governor is usually used for this purpose.

15.12.2.1 Governing of Impulse Turbines

Figure 15.63 shows a governing system used for a Pelton turbine. The system consists of a centrifugal governer, two relay valves V_1 and V_2 a relay cylinder, an oil pump and a nozzle–spear arrangement.

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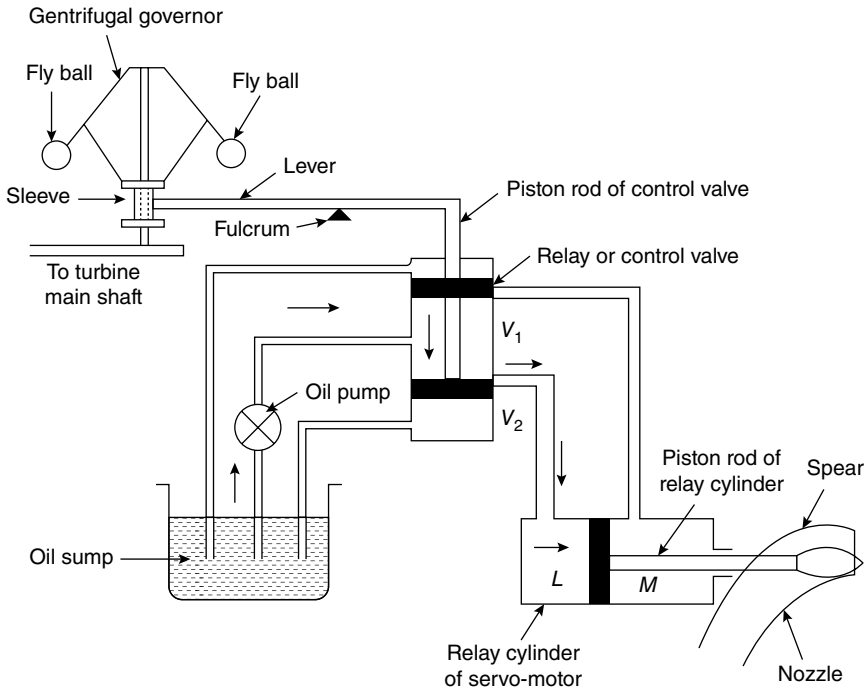


Fig. 15.63 *Governing of Impulse Turbine(Pelton wheel)*

When the turbine speed is increased, piston rod moves downwards closing value V_1 and opening value V_2. The oil pump pumps oil from the sump to the control or relay value. Oil flows through value V_2, exerting pressure on the blank end of the relay cylinder. This forces the spear of the cylinder towards the right-hand side, thereby restricting the flow through the nozzle. This action reduces the turbine speed till it reaches the normal value where fly balls, sleeve, lever also come to normal positions.

When the load on the generator increases, turbine speed decreases and the above actions will be reversed. Due to less centrifugal force, fly balls move inwards; the sleeve moves down and the piston and moves up. Valve V_1 opens and valve V_2 closes, oil under pressure flows through value V_1 and exerts a pressure on the rod end side of the piston. The spear moves towards left, making way for more water to flow through the nozzle, until normal speed is reached.

15.12.2.2 Governing of Reaction Turbines

Figure 15.64 shows the governing system of a Francis turbine. The system consists of governor, a gear pump, a footing lever, a bell crank lever PQR and a Bervo motor with a rack-and-pinion arrangement.

A compensating mechanism is used to prevent the governor from over shooting. When the turbine speed decreases due to the load on the generator, piston rod of control valve moves down opening valve V_1 and closing valve V_2. Oil pressure is exerted on the rod end of the piston,

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rotating the bell crank lever PQR upwards, about the pivot *Q*, lifting the fulcrum *B*. The relay port '*p*' partially or fully opens. Reverse operation happens when the turbine speed increases.

The governor is generally operated by a pressure relief value. When the wicket gate is closed suddenly, relief valve opens due to increased pressure and protects the water carrying conduits from inertia effect of flowing water. If the bell crank lever opens the pilot valve of the pressure chamber, reduces the fluid pressure on the spear and hence lifting it, allowing a portion of water to discharge at the tailrace, by passing the runner. It can be noted that both the deflector and the relief valve mechanism used in Pelton and Francis or Kaplan turbine are same.

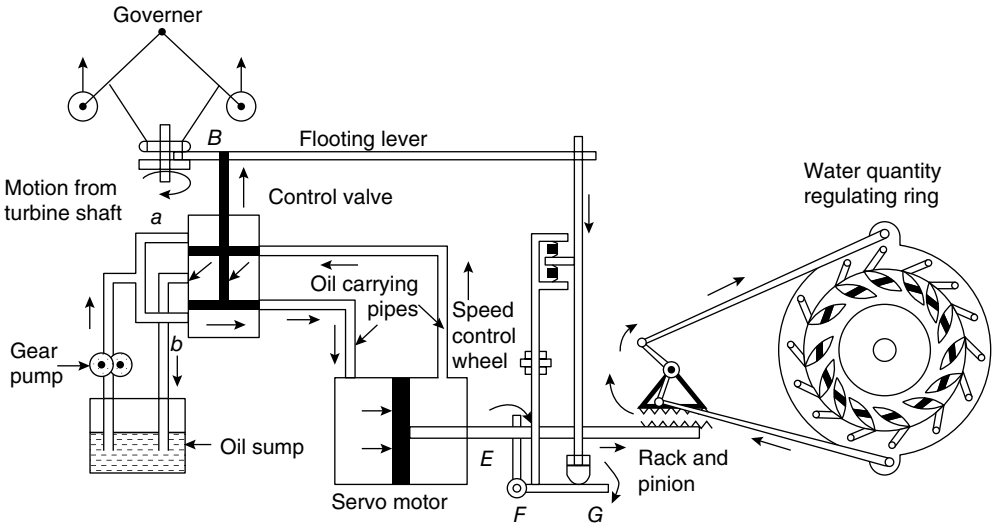


Fig. 15.64 *Governing of Reaction Turbine*

15.13 MICRO HYDEL DEVELOPMENTS

In India, hydro-projects up to 25 MW station capacities have been categorized as small hydro power (SHP) projects. While Ministry of Power, Government of India deals with large hydro-projects, small hydro power (up to 25 MW) is handled by Ministry of New and Renewable Energy. The estimated potential for power generation in the country from such plants is over 15,000 MW. Most of the SHP potential is in Himalayan States as river-based projects; and in other States, the potential is on irrigation canals. Of late, the SHP program is essentially private investment driven because of economical viability. The Ministry's aim is that at least 50 per cent of the potential in the country is harnessed in the next 10 years (source: MNRE).

Depending on station capacity small hydro power projects are further classified as micro, mini and small hydro as given in Table 15.1.

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Table 15.1 *Classification of Small Hydro-Plants*

Class	Station capacity in
Micro hydro	Up to 100 kW
Mini hydro	101 kW to 2 MW
Small hydro	2 MW to 25MW

Typical components of small hydro power projects are as follows:

(i) Diversion weir: It is a barrier built across the river to divert water through an opening on the riverside into a settling basin.

(ii) Desilting basin: Desilting basin also called as the settling basin is constructed to trap the sand or suspended silt from water before entering into penstock. It is built at the intake or at the forebay.

(iii) Water conductor system: A penstock is used to carry water to the turbine under pressure. It is a closed conduit or pressure pipe.

(iv) Power house: Power house houses turbo-generators and turbine selection is made depending on the available head. The water turbine converts the kinetic energy of water into mechanical energy.

(v) Tail race tunnel or channel: A tail race is the place where water is discharged on the downstream

(vi) Switch yard: Switch yard houses feeders from where electricity is transmitted.

(vii) Transmission line: Transmission lines supply electricity to the different load centres.

Figure 15.65 shows a typical layout of a small hydro-plant site.

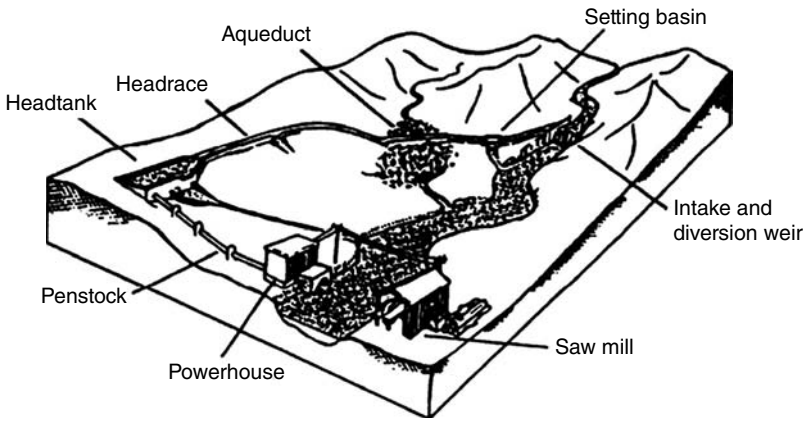


Fig. 15.65 *Layout of a Small Hydro-Plant*

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Depending on the site characteristics, the head available, the power required, part load conditions and the speed at which it is desired to run, the generator turbine selection is made. Table 15.2 shows best turbines suitable for a variety of heads available.

Table 15.2 *Best Turbines Suitable for a Variety of Head Available*

	Head on the turbine		
Turbine type	**High < 40 m**	**Medium 20–40 m**	**Low 5–20 m**
Impulse	Pelton Turgo	Cross flow(Banki) Turgo Pelton	Cross flow(Banki)
Reaction		Francis pump as turbine (PAT) Kaplan Propeller	Propeller Kaplan

15.14 PROJECT COST OF HYDROELECTRIC PLANT

Project cost estimation of a hydroelectric plant depends on direct and indirect costs. The estimates of a hydroelectric scheme shall be divided under the following heads (source: MNRE):

A. *Direct cost*

I. *Works*

A – Preliminary
B – Land
C – Works
J – Power plant civil works
K – Buildings
M – Plantation
O – Miscellaneous
P – Maintenance during construction
Q – Special Tool and Plants
R – Communications
S – Power plant and electro-mechanical system
X – Environment and ecology
Y – Losses on stock

Total I-works

1. Establishment
2. Tools and plants

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3. Suspense
4. Receipt and recoveries

Total (A) – direct cost

B. *Indirect cost*

(i) Capitalisation of abatement of land revenue
(ii) Audit and account charges

Total (B) – indirect cost
Total cost (A + B)

Same information is depicted in Table 15.3.

Table 15.3 *Estimation of Project Cost*

	Price level		
	Amount (Rs lakhs)		
Item	**Civil works**	**E & M works**	**Total**
Direct Cost	–	–	–
I- Works	–	–	–
A- Preliminary	–	–	–
B- Land	–	–	–
C-Civil Works	–	–	–
J- Power Plant Civil Works	–	–	–
K-Building	–	–	–
M-Plantation	–	–	–
O-Miscellaneous	–	–	–
P-Maintenance during Construction	–	–	–
Q-Special T & P	–	–	–
R-Communication	–	–	–
S- Power Plant **(Details as per Annex-S)**	–	–	–
X-Environment & Ecology	–	–	–
Y-Losses on stock	–	–	–
Total (I- Works)	–	–	–
II Establishment	–	–	–
III. Tools And Plants	–	–	–
IV. Suspense	–	–	–

(*Continued*)

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Table 15.3 (*Continued*)

	Price level		
	Amount (Rs lakhs)		
Item	**Civil works**	**E & M works**	**Total**
V. Receipt And Recoveries	–	–	–
Total (Direct Cost)	–	–	–
Indirect Cost	–	–	–
Capitalisation of Abatement of Land Revenue	–	–	–
Audit and Account Charges	–	–	–
Total (Indirect Cost)	–	–	–
Total Cost (Direct & Indirect Cost)	–	–	–
IDC & FC	–	–	–
Total with IDC & FC	–	–	–

**Note*: Readers are advised to visit MNRE website for further details.

*Preparation of detailed estimates of cost (I-Works) (source: MNRE)**

A – Preliminary

The provision under this head covers the works relating to various investigations, surveys, model tests, ecological studies, etc. This should be based on the actual cost likely to be incurred and should not exceed 2 per cent of the total cost of I-works.

B – Land

The provisions under this head covers acquisition of land, rehabilitation and resettlement including compensation for property, interest charges, solatium charges, demarcation and measurment charges, etc. The provision should be made as per actual.

C – Works

The provisions under this head covers the costs of the head works viz. dam, spillway, energy dissipation works, outlets (irrigation, power, water supply and scour sluices), pick up weir, barrage, head regulators, etc.

J - Power plant civil works

Items (not complete) included under this head are

(a) *Intake structures*

(i) Excavation
(ii) Foundation treatment
(iii) Cement concrete for foundation, piers and abutments
(iv) Masonry/concrete for guide walls of approach channel
(v) Concrete for trash racks including raking arrangement
(vi) Gates with auxiliary equipment

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(vii) Reinforcement steel
(viii) Instrumentation, etc.

(b) *Head race/tail race tunnels (including cut and cover section)*
(i) Excavation: Open cut tunnel including temporary supports
(ii) Rock bolts
(iii) Permanent support, ventilation
(iv) Drainage
(v) Cement concrete for lining
(vi) Steel lining
(vii) Drilling and grouting
(viii) Gates and ancillaries, where required
(ix) Reinforcement steel
(x) Instrumentation

(c) *Head race channel and tail race channel*
(i) Excavation
(ii) Embankment
(iii) Lining with cement concrete in bed and sides with drainage pipes and valves
(iv) Pucca works
– Cross Drainage(s)
– Escape(s)
– Bridge(s)
– Meter flume
– Balancing tank
(v) Instrumentation

(d) *Surge shaft*
(i) Excavation
(ii) Cement concrete lining
(iii) Drilling and grouting
(iv) Miscellaneous items such as masonry, steel lining, ladder, bolts, etc.
(v) Reinforcement steel
(vi) Instrumentation

(e) *Penstock*
(i) Excavation
(ii) Cement concrete for
– Bed
– Anchor blocks
– Intermediate supports

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(iii) Steel pipes for
- – Stiffner
- – Reducers
- – Bends
- – Wye pieces
- – Penstock valves

(iv) Instrumentation

(f) *Power house*

(i) Excavation

(ii) Concrete for foundation, sub-structure, super structure and supports for turbines and generators

(iii) Masonry/concrete for super structure and other necessary items for building work

(iv) Scroll casing/generator barrel

(v) Draft tube

(vi) Bulkhead gates, crane and hoisting equipment

(vii) Power house crane

(viii) Miscellaneous items such as anchor bolts, grouting, etc.

(ix) Instrumentation

K – Buildings

The provisions under this head covers the residential/non-residential buildings, office buildings, testing laboratories, workshops, other service buildings, community centre, etc. The provision shall be made as per the actual requirement.

M – Plantation

The provisions under this head covers the plantation program including gardens, etc., required for beautification as considered necessary downstream of dam and appurtenances around power house and other important structure. The provision should be made on lump sum basis keeping in view the experience of other projects.

O – Miscellaneous

The provisions under this head covers the capital cost and maintenance of electrification, water supply, sewage disposal and drainage works, recreation, medical, fire fighting equipments, inspection vehicles, school bus, pay van, visit of dignitaries, welfare works, etc.

The provision, however, should not exceed.

(i) @ 3 per cent of the cost of I-Works upto ₹ 1,000 crores limited to ₹ 20 crores

(ii) @ 2 per cent of the cost of I-Works up to ₹ 2,000 crores limited to ₹ 30 crores

(iii) @1.5 per cent of the cost of I-Works greater than ₹ 2,000 crores limited to ₹ 40 crores)

P – Maintenance during construction

The provisions under this head covers the cost of maintenance of all works during the construction period. The provision should be 1 per cent of the total cost under the heads of C-Works, J-Power House Civil Works and K-Buildings.

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Q – Special Tools and Plants

The provisions under this head covers the drilling and grouting equipments, transport, compaction, electrical equipments, construction plant and earth moving equipments and other miscellaneous equipments.

R – Communication

The provisions under this head covers the construction of main approach roads, quarry roads, temporary or permanent river crossing, railways, bridges, connecting roads, water ways and airstrip/helipad. The major items on this account shall be supported by sub-estimates. The provisions shall be made in consultation with the concerned authorities.

S – Power plant and electro – mechanical system

X – Environment and ecology

Y – Losses on stock

The provision under this sub-head is generally made at 0.25 per cent of the total cost of C-Works, J-Power Plant Civil Works, and K-Buildings only.

Direct cost

Establishment

The projects presently are being executed through 3 or 4 major contract packages in case of civil works and a single EPC contract for electro-mechanical works. However this should in any case not exceed the following:

For civil works estimated to cost

(i) Up to ₹ 1,000 crores @ 8 per cent with a ceiling of ₹ 60 crores

(ii) Rs. 1,000 to ₹ 2,000 crores @ 6 per cent with a ceiling of ₹ 80 crores

(iii) > ₹ 2 000 crores @ 4 per cent with a ceiling of ₹ 100 crores

For electro – mechanical works

Tools and plants

The provisions under this head covers survey instruments, camp equipments, office equipments and other small tools.

Suspense

The net provision under this minor head will be "NIL" as all the outstanding suspense accounts are expected to be cleared by adjustment to appropriate heads on completion of the project.

Receipts and recoveries on capital account

Indirect charges

(a) Indirect charges include the amount required to cover the capitalization of abatement of land revenue on the area occupied by the works and allowance for the cost of Audit and Accounts and Establishment.

15.15 ADVANTAGES OF HYDRO-POWER PLANT

Advantages

The hydroelectric power plants are the best from many viewpoints. These are simple in design, easy to maintain, pollution free with zero fuelling cost. Hydro-generation has a unique and

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significant role to play particularly in the operation of interconnected power system. In fact, thermal or nuclear power can seldom produce electrical energy economically without an adequate proportion of hydro power generation. Some of the advantages of hydroelectric power plants are mentioned below:

(i) The operating cost of the hydroelectric plant including auxiliaries is considerably low when compared with thermal plants. The annual operating and maintenance cost of a thermal plant is approximately 5–6 times that of hydro-plant of equal capacity.

(ii) Cost of power generation is less. The cost of each unit of electrical energy delivered from a thermal power plant is very adversely affected by the reduction of load factor at which the plant is operated. The cost per kilowatt generated by a thermal plant designed for 80 per cent load factor would go up by 20 per cent if operated at 60 percent load factor and by 65 per cent if operated at 40 per cent load factor.

The cost of generation by hydroelectric plant is more or less unaffected by the load factor.

(iii) Life expectancy of a hydroelectric power plant is more. The useful life of a thermal plant is 20–25 years as against 100–125 years for hydro plant.

(iv) There is no problem of handling the fuel and ash and no nuisance of smoke exhaust gases and soots and no health hazards due to air pollution.

(v) The fuel needed for the thermal plants has to be purchased, whereas in hydro plant the fuel cost is totally absent. The security of thermal power plant is mostly dependant on transport facilities and its security is less when the fuel is imported from outside countries.

(vi) Hydroelectric plants are quick to respond to the change of load compared with thermal or nuclear plant. Full output can reached in about 10–15 sec. Therefore, they are ideally suited for peaking purposes. The rapid fluctuating loads are served most economically by hydro-plants.

(vii) The machines used in hydel plants are more robust and generally run at low speeds at 300–400 RPM, whereas the machines used in thermal plants run at a speed 3,000–4,000 RPM. Therefore, there are no specialized mechanical problems or special alloys required for construction. There are no complications due to high temperature and pressures. Therefore, the outages of the machines are very few.

(viii) Flexible and economical in operation. Hydel-power plays a major role not only providing an economic source of energy but also in improving the operation of the combined power system by meeting the system load at various times and providing regulating functions.

(ix) The efficiency of hydro-plant does not change with age, but there is considerable reduction in efficiency of thermal as well as nuclear power plant with age.

(x) In a hydroelectric plant, there are no stand-by losses, whereas these are unavoidable for thermal plants and the number of operations required is considerably small compared with thermal power plants.

(xi) It does not contribute to air and water pollution to Green House Effect.

(xii) Usually, the hydro-stations are situated away from the developed areas, therefore the cost of land is not a major problem.

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Disadvantages

(i) The capital cost (cost per kilo-watt capacity installed) of the hydro-plant is considerably more than thermal plant.

(ii) It takes considerable long time for its erection compared with thermal plants.

(iii) Power generation by the hydro-plant is only dependent on the quantity of water available which in turn depends on the natural phenomenon of rain. The dry year is more serious for the hydroelectric project.

(iv) The site of hydroelectric station is selected on the criterion of water availability at economical head. Such sites are usually away from load centres. The transmission of power from power station to the load centre requires long transmission lines. Therefore, investment required for long transmission lines and loss of power during transmission, are unfavourable factors for the economical selection of hydro-plants.

15.16 QUESTIONS

15.16.1 Review Questions

1. Sketch and explain general arrangement of storage-type hydroelectric power plant.
2. Explain various elements of general layout for a hydroelectric power plant with a neat sketch.
3. Explain the factors to be considered while selecting the site for a hydroelectric power plant.
4. Write short notes on surge tanks.
5. Explain the function of (i) surge tank and (ii) draft tube.
6. What are different types of turbines used in hydroelectric plants? Give their relative advantages with reference to the conditions of head.
7. With reference to a hydroelectric power plant, explain the following:
   (i) Surge tank
   (ii) Draft tube
8. Discuss the working principle of a hydro power plant.
9. State the advantages and disadvantages of hydroelectric power plant.
10. What are the factors to be considered while selecting the site for hydroelectric power plant?
11. How are hydroelectric power plants classified?
12. What is meant by pump storage plant? What are its advantages and limitations?
13. Mention the advantages and disadvantages of hydroelectric power plants compared with thermal power plants.
14. What are the specific advantages of storage reservoir-type power plant? How do they differ from other types of hydro power plants?
15. What do you understand by run-off river power plant? How does its performance increase by introducing a pondage in the plant?
16. What is the importance of rainfall and run-off data in the design of hydro electric power plant?
17. What are the different methods used to measure the rainfall? Discuss the relative merits.

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18. What factors affect the run-off data of a particular site?
19. How is the run-off measured in practice?
20. Define mass curve and explain its use in the design of dam capacity and spillway capacity.
21. Explain the construction of flow duration curve.
22. What do you understand by unit hydrograph? Explain its use in practice.
23. The discharge through a mansion stream may be taken as follows:

Month	Discharge in (m^3/sec)
Jan	2
Feb	1.5
Mar	1
Apr	0.6
May	0
Jun	0
Jul	8
Aug	10
Sep	12
Oct	6
Nov	4
Dec	3

Draw hydrograph and find out the following:

(i) Average discharge

(ii) The storage capacity if the dam is constructed across the stream

(iii) The power supplied if the mean level of water on the upstream side is 100 m above the tail race level. Assume overall efficiency of generation to be 80 per cent.

24. The mean monthly discharge for 12 months at a particular site of a river is tabulated below.

Month	Discharge in (millions m^3/mo)
Apr	500
May	200
Jun	1,500
Jul	2,500

(*Continued*)

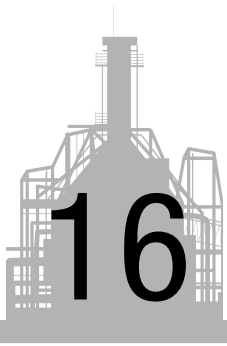
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(*Continued*)

Month	Discharge in (millions m^3/mo)
Aug	3,000
Sep	2,400
Oct	2,000
Nov	1,500
Dec	1,500
Jan	1,000
Feb	800
Mar	600

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Nuclear Power Plants



16

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16.7 Reactor control
16.8 Radiation hazards
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16.1 INTRODUCTION TO NUCLEAR ENGINEERING

With the acute shortage of other sources of energy, namely fossil-based fuels and hydel source, the use of nuclear energy has become an inevitable option for both developed and developing countries. The amount of heat generated by burning 1 kg of nuclear fuel is equivalent to the energy generated by burning 3000 tonnes of coal or 1600 tonnes of oil.

The most important advantage of nuclear energy is that it has no combustion by-products and under safe working conditions contributes no pollutant to air. Site selection for producing nuclear energy is completely independent of geographical area and it requires no fuel transportation networks and large storage facilities. However, nuclear wastes from a power plant are more hazardous and face severe disposing problems.

In this chapter, a brief study of nuclear energy, nuclear reactions, basic definitions, nuclear reactors and nuclear waste disposal is made.

Advantages of nuclear power plant

(i) Less space requirement
(ii) Consumes very small quantity of fuel
(iii) Fuel transportation cost is less
(iv) Large fuel storage facility is not required
(v) Reliable in operation
(vi) Not affected by changes in weather conditions

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Disadvantages

(i) High initial and maintenance costs
(ii) Not suitable for varying load conditions
(iii) Radioactive wastes should be disposed off carefully
(iv) Requires trained operators

16.1.1 Atomic Structure

Atom consists of a relatively heavy, positively charged nucleus and a number of much lighter negatively charged electrons. Electrons exist in various orbits around the nucleus. The nucleus consists of two sub-particles known as nucleons (Figure 16.1).

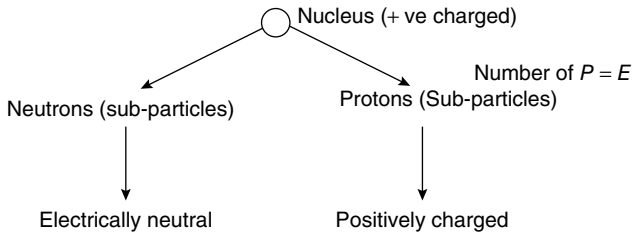


Fig. 16.1 *Atomic Structure*

The electric charge on the proton is equal in magnitude but opposite in sign to that of electron. The atom as a whole is electrically neutral. The number of protons is equal to the number of electrons in orbit.

One atom may be transformed into another by losing or gaining some of the sub-particles. This results in mass change Δm and enormous amount of energy is released (or absorbed). According to Einstein's law

$$\Delta E = \Delta m c^2 \tag{1}$$

where c = speed of light in vacuum
Energy from nuclear reactions can be estimated by using the above Equation (1).

$$\Delta E = \frac{1}{g_c} \Delta m c^2$$

If mass is in kg, $g_c = 1$ kg m/Ns^2

$$\Delta E = 9 \times 10^{16} \, \Delta m \text{ in J}$$

If mass is in amu and energy is in MeV

$$\Delta E \text{ (in J)} = 1.49 \times 10^{-10} \, \Delta m \text{ (in amu)}$$
$$\Delta E \text{ (in MeV)} = 931 \, \Delta m \text{ (in amu)}$$

where

$$1 \text{ amu} = 1.66 \times 10^{-27} \text{ kg}$$
$$1 \text{ eV} = 1.6021 \times 10^{-19} \text{ J}$$

Figure 16.2 shows the atomic structure of Hydrogen and Helium.

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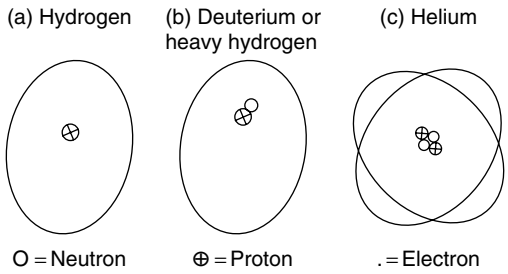


Fig. 16.2 *Atomic Structure of Hydrogen and Helium*

Most of the mass of atom is in the nucleus. The masses of three atomic subparticles are

Neutron mass, $m_n = 1.008665$ amu

Proton mass, $m_p = 1.007277$ amu

Electron mass, $m_e = 0.0005486$ amu

1 amu (atomic mass unit) $= 1.66 \times 10^{-24}$ g

16.1.2 Some Definitions

1. *Atomic number*

Number of protons in the nucleus is called atomic number Z. It is unique for each chemical element and represents both the number of positive charges on the central massive nucleus of the atom and the number of electrons in orbits around the nucleus. The maximum allowed numbers of electrons in orbits as Z increases for the first few shells are 2, 8 and 18.

2. *Mass number*

The total number of nucleons in the nucleus is called the mass number A. Nuclear symbols are written as ${}_zX^A$, where X is the chemical symbol.

The masses of atoms are compared on a scale in which an isotope has a mass of exactly 12.

1 amu

$$= \frac{1}{12} \text{ the mass of } {}_6C^{12}$$

$$= 1.66 \times 10^{-24} \text{ g}$$

Thus, $A = 1.66 \times 10^{-24}$ = mass of the atom.

3. *Binding energy*

The nuclear force acts only when the nucleons are very close to each other and binds them into a compact stable structure. The energy associated with this force (potential energy) is known as binding energy.

To disrupt a nucleus and separate it into its component nucleons, energy must be supplied from outside.

Let

M = mass of an atom (mass of nucleus + electrons)

m_n = mass of neutron

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m_H = mass of proton electron
N = neutron number $(A - Z)$

Then, binding energy

$$B = Nm_n + Zm_H - M$$

e.g. Binding energy of tritium

$$B = (3 - 1)1.00866 + 1(1.00782) - 3.01702$$
$$= 0.00812 \text{ amu}$$

4. *Valence electrons*

Electrons that orbit in the outermost shell of an atom are known as valence electrons. The outermost shell is called valence shell.

5. *Isotopes*

Atoms with nuclei having the same number of protons and similar in physical and chemical characteristics but differing in masses are known as isotopes.

e.g. Deuterium is an isotope of hydrogen

6. *Atomic number*

The number of protons in the nucleus is called atomic number Z.

16.2 RADIOACTIVE DECAY, HALF LIFE

16.2.1 Radioactive Decay

All isotopes of heavier elements starting from atomic number $Z = 84$ (polonium) posses weak binding energy/nucleon and have the tendency to emit radiation until a more stable nucleus is formed. This new nucleus is called daughter and the original nucleus is called parent. This is a spontaneous disintegration process, known as radioactive decay.

Radioactive isotopes may be either natural or artificial and are known as radioisotopes. Radioactivity is characterized by either decrease in mass or liberation of energy (in the form of kinetic energy) accompanied by electromagnetic radiation.

An example of radioactivity is

$$^{234}_{90}\text{Th} \rightarrow {}^{234}_{92}\text{Pa} + {}^{0}_{-1}\text{e}$$

Naturally occurring radioisotopes undergo the following types of decay:

1. *Alpha decay*

A nucleus undergoes alpha decay if it emits an α-particle that is identical to a helium nucleus.

1. $^{238}_{92}\text{U} \rightarrow {}^{4}_{2}\text{He} + {}^{234}_{90}\text{Th} + 2\,{}^{0}_{0}\gamma$
2. $^{239}_{94}\text{Pu} \rightarrow {}^{4}_{2}\text{He} + {}^{235}_{92}\text{U}$

2. *Beta decay*

A nucleus undergoes β decay if the energy released in the decay process creates a β-particle that is identical to a high-speed electron.

$$^{234}_{90}\text{Th} \rightarrow {}^{234}_{92}\text{Pa} + {}^{0}_{-1}\text{e}$$

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In β^- decay, a β^- particle (high-speed electron) and an antineutrino, $\bar{\nu}$, are produced. β^- particles are likely to produce nuclides having high neutron-to-proton ratios. The net effect of this is to 'convert neutron to proton' (Figure 16.3).

$$^{14}_{6}\mathrm{C} \rightarrow {}^{14}_{7}\mathrm{N} + {}^{0}_{-1}\beta^- + \bar{\nu}$$

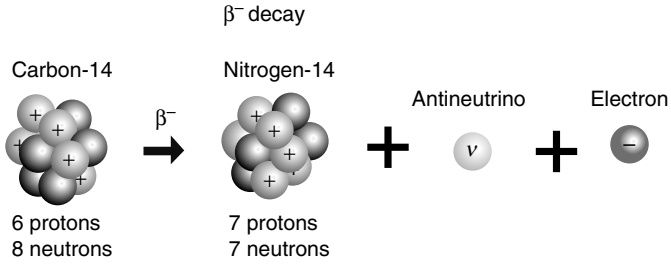


Fig. 16.3 *β^- Decay*

In β^+ decay, a high-speed electron and a neutrino, v, are produced. β^- particles are likely to produce nuclides having low neutron-to-proton ratios. The net effect of this is to 'convert proton to neutron' (Figure 16.4).

$$^{13}_{7}\mathrm{N} \rightarrow {}^{13}_{6}\mathrm{C} + {}^{0}_{1}\beta^+ + \nu$$

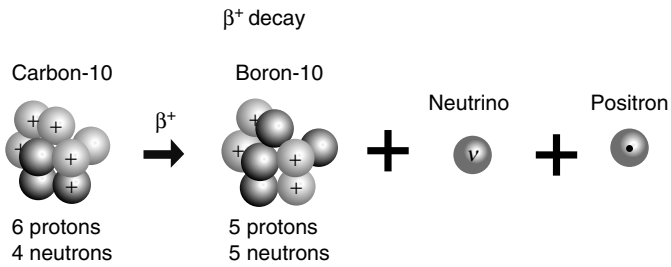


Fig. 16.4 *β^+ Decay*

3. *Spontaneous fission*

It is a slow process in which a heavy nuclide is split into lighter nuclides with similar mass numbers.

$$^{252}_{98}\mathrm{Cf} \rightarrow {}^{140}_{54}\mathrm{Xe} + {}^{108}_{44}\mathrm{Ru} + 4{}^{1}_{0}\mathrm{n}$$

4. *Electron capture*

In electron capture, a nucleus captures one of the atom's own electrons changing a proton to a neutron and emits a neutrino, v. This happens when a nucleus possesses an excess proton and does not have sufficient energy (1.024 MeV) to emit a positron and hence captures an orbital electron from the K-shell. As the captured electron usually comes from the atom's K-shell, this process is also known as K-capture. Even though the numbers of protons and neutrons in the nucleus of an atom change during electron capture, the total number of particles (protons + neutrons) remains the same (Figure 16.5).

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1. ${}^{7}_{4}\mathrm{Be} + {}^{0}_{-1}\mathrm{e} \rightarrow {}^{7}_{3}\mathrm{Li} + \mathrm{v}$
2. ${}^{195}_{80}\mathrm{Hg} + {}^{0}_{-1}\mathrm{e} \rightarrow {}^{195}_{79}\mathrm{Au}$

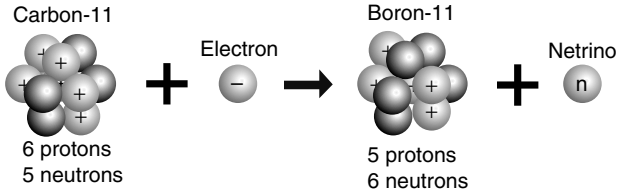


Fig. 16.5 *Electron Capture*

5. *Gamma decay*

In a gamma decay, a nucleus in an excited state emits a high-energy photon known as gamma ray while transition takes place from high-energy to low-energy state.

1. ${}^{125}_{53}\mathrm{I} \rightarrow {}^{7}_{3}\mathrm{I} + \gamma$
2. ${}^{60}_{27}\mathrm{Co} \rightarrow {}^{60}_{28}\mathrm{Ni} + {}^{0}_{-1}\mathrm{e} + 2{}^{0}_{0}\gamma$

Table 16.1 shows radioactive decay process of uranium-238 with corresponding half-lives (source: Internet).

16.2.2 Half-Life

It is the time during which one-half of a number of radioactive species decay or one-half of their activity cease.

Let N = number of radioactive nuclei of one species

θ = time

$$\therefore \text{Rate of decay} = -\frac{dN}{d\theta} = \lambda N = A$$

where λ = decay constant (depends on isotopes) in s^{-1}.

Integrating between an arbitrary time $\theta = 0$ and with radio isotopes, N_o

$$\int_{N_o}^{N} \frac{dN}{N} = l \int_{0}^{\theta} d\theta$$

$$\frac{N}{N_o} = e^{-\lambda\theta} \qquad (1)$$

Table 16.1 *Radioactive Decay Process of Uranium-238*

Uranium 238(U238) Radioactive Decay		
Type of radiation	Nuclide	Half-life
	Uranium-238	4.47 billion years
α	Thorium-234	24.1 days
β	Protactinium-234m	1.17 minutes
β	Uranium-234	245000 years
α	Thorium-230	8000 years
α	Radium-226	1600 years
α	Radon-222	3.823 days
α	Polontum-218	3.05 minutes
α	Lead-214	26.8 minutes
β	Bismuth-214	19.7 minutes
β	Polonium-214	0.000164 seconds
α	Lead-210	22.3 years
β	Bismuth-210	5.01 days
β	Polonium-210	138.4 days
α	Lead-206	Stable

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In the form of activity,

$$\frac{A}{A_o} = e^{-\lambda\theta} \tag{2}$$

Half-life time is the time period required to bring the number of active particles from N_o to $N_o/2$ (Figure 16.6),

Hence, half-life, $\theta_{1/2} = \frac{N}{N_o} = \frac{A}{A_o}$

$$= e^{-\lambda\theta\,1/2} = \frac{1}{2}$$

$$N = N_o \times e^{-\lambda\theta\,1/2}$$

$$\therefore \quad \theta_{1/2} = \frac{\ln 2}{\lambda} = \frac{0.6931}{\lambda} \tag{3}$$

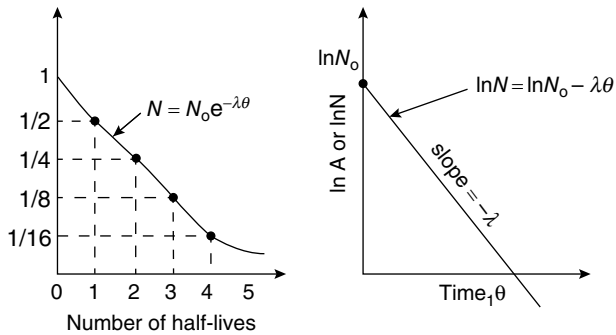


Fig. 16.6 *Half-Life*

The original unit for measuring the amount of radioactivity was the curie (Ci). It was defined corresponding to measurement of the activity of 1 g of radium-226.

1 Ci = 3.615×10^{10} radioactive decays per second (dis/s)

In the International System of Units (SI), the curie has been replaced by the becquerel (Bq) and is defined as 1 radioactive decay per second.

1 Bq = 1 dis/s = 2.703×10^{-11} Ci

Table 16.2 gives an idea of how half-lives variy for a few radioisotopes (source: http://astro.berkeley.edu).

Table 16.2 *Half-Lives of Selected Radioisotopes*

Isotope	Half-life (year)
Radium-226	1600
Molybdenum-93	4000
Plutonium-240	6563

(*Continued*)

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Table 16.2 (*Continued*)

Isotope	Half-life (year)
Thorium-229	7340
Plutonium-239	24,110
Proactinium-231	32,760
Thorium-230	75,380
Thorium-230	77,000
Uranium-233	159,200
Krypton-81	229,000
Uranium-234	245,500
Plutonium-242	373,300
Beryllium-10	1,510,000
Neptunium-237	2,144,000
Cesium-135	2,300,000
Palladium-107	6,500,000
Hafnium-182	9,000,000
Lead-205	15,300,000
Curium-247	15,600,000
Uranium-236	23,420,000
Niobium-92	34,700,000
Plutonium-244	80,800,000
Uranium-236	234,200,000
Uranium-235	703,800,000
Uranium-238	4,468,000,000
Rubidium-87	4,750,000,000
Thorium-232	14,100,000,000
Platinum-190	650,000,000,000

16.3 PRINCIPLES OF RELEASE OF NUCLEAR ENERGY

16.3.1 Fusion and Fission Reactions

16.3.1.1 Nuclear Fission

It is the process in which a heavy nucleus is split into two or more lighter nuclei. This results in decrease in mass and consequent exothermic energy and emission of neutrons. Two to three neutrons are emitted per nucleus, which are known as *fission neutrons*. These fission neutrons cause fission in additional nuclei, thus liberating more neutrons.

$$\text{Neutron} + \begin{matrix}\text{Heavy}\\ \text{Nucleus}\end{matrix} + \begin{matrix}\text{Fission}\\ \text{Fragments}\end{matrix} + \begin{matrix}\text{Neutrons}\\ (2\text{ to }3)\end{matrix} + \text{Energy}$$

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Requirement for self-sustaining reaction

(i) Heavy nuclei must be such that they can be fissioned by neutrons of an energy. Such substances are called fissile species.

(ii) Due to collisions with various nuclei, initial high kinetic energy of fission neutron decreases. Thus, for a sustained reaction, even neutrons with lower energy should be capable of causing fission.

(iii) All fissile substances or species should have long half-lives so that rate of decay is not fast.

(iv) They should be either available freely in moderate quantities or can be easily produced from available materials.

e.g. 1. U^{233}, U^{235}, Pu^{239} are fissionable by all energy neutrons

2. U^{238}, Th^{232} and Pu^{240}, are fissionable high-energy neutrons

Only U^{235} is available in nature.

Even though fission can be caused by other particles, only neutrons can result in sustained reaction as two to three neutrons are released for each one absorbed by fission.

When a neutron is absorbed by a nucleus of an atom of U^{235} a U^{236} isotope is formed. This isotope is highly unstable, which lasts for one millionth of a second and splits into two equal parts releasing an energy of 200 MeV (Figure 16.7).

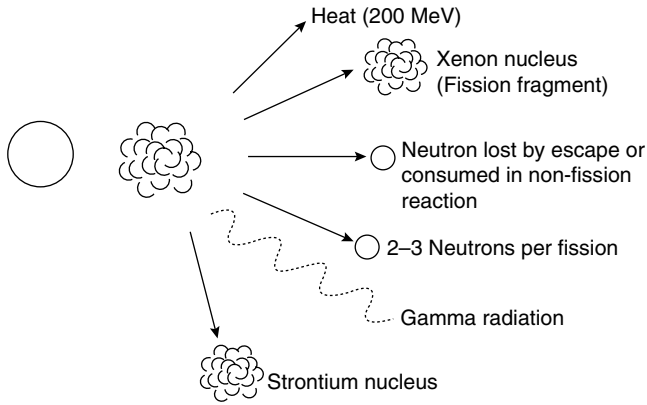


Fig. 16.7 *Nuclear Fission*

The fission products formed are fission fragments, neutrons and electromagnetic or gamma radiation. Most of the energy released is in the form of kinetic energy and is absorbed by fission products. As the fragments collide, the kinetic energy is converted into heat energy.

e. g. 1. $_{92}U^{233} + {}_0n^1 \rightarrow {}_{54}Xe^{140} + {}_{38}Sr^{94} + 2{}_0n^1 + \gamma + 196$ MeV (1)

2. $_{92}U^{235} + {}_0n^1 \rightarrow {}_{56}Ba^{137} + {}_{36}Kr^{97} + 2{}_0n^1 + \gamma$ (2)

For Equation (2) making mass balance

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$$235.0439 + 1.00867 \rightarrow 136.906 + 96.9212 + 2 \times 1.00867$$
$$236.0526 \rightarrow 235.8446$$
$$\Delta m = 235.8446$$
$$= -0.2080 \text{ amu}$$
$$\Delta E = 931 \times (-0.2080) = -193.6 \text{ MeV}$$

Complete fission of 1 g of U^{235} nuclei yields

$$= 0.513 \times 10^{24} \text{ MeV}$$
$$= 2.276 \times 10^{44} \text{ kWh (1 MeV} = 4.45056 \times 10^{-20} \text{ kWh)}$$
$$= 0.948 \text{ MW-day, which is known as fuel burn up.}$$

16.3.1.2 Nuclear Fusion

It is the process that involves fusion of two light nuclei of low mass to produce a heavy nucleus resulting in decrease of mass and release of enormous amount of energy. Energy produced in the sun and stars is by nuclear fusion.

Four nuclei of hydrogen fuse in a series of reactions involving other particles that continually appear and disappear such as He^3, nitrogen, carbon and other nuclei.

$$4_1H^1 \rightarrow {}_2He^4 + 2_{+1}e^0 \text{ (positrons)}$$

Mass decreases to about 0.0276 amu releasing 25.7 MeV. The heat liberated during this reaction results in temperature of the order of million degrees (10^8–10^9 K) and sustains the succeeding reactions.

To cause fusion, it is necessary to accelerate positively charged nuclei to high kinetic energies to overcome electrical repulsive forces. This is done by raising their temperatures to hundreds of millions of degrees resulting in plasma. The plasma should be prevented from contacting the walls of the container and confined for a period of time of the order of a second at a minimum density. Fusion reactions are called *thermonuclear* because of the higher temperature requirement to trigger and sustain the reaction.

Some fusion reactions are

$$D + D \rightarrow T + p + 4 \text{ MeV} \quad (1)$$
$$D + D \rightarrow He^3 + n + 3.2 \text{ MeV} \quad (2)$$
$$T + D \rightarrow He^4 + n + 17.6 \text{ MeV} \quad (3)$$
$$He^3 + D \rightarrow He^4 + p + 18.3 \text{ MeV} \quad (4)$$

The four hydrogen reactions observed in the sun and stars require billions of years for completion, whereas deuterium–deuterium reaction requires a fraction of a second. Of the four reactions listed above, the most readily found fusion reaction is the third (Figure 16.8).

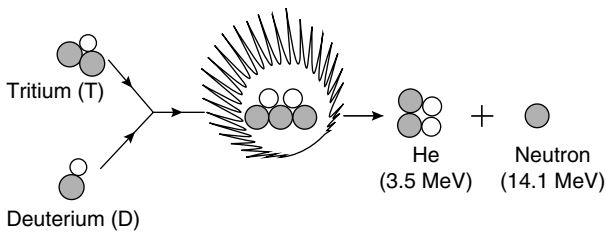


Fig. 16.8 *Fusion Process of D and T*

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In the above process, *deuterium* used is readily available in ordinary water (2 out of 6500 water molecules) and *tritium* is made by bombarding lithium available in sea water, with neutrons. The energy is supplied by accelerating the charged particles with electrostatic fields.

16.3.2 Breeding and Fertile Materials

Varieties of fuel materials have been developed for use in power reactors. These include metals and alloys, oxides, carbides, nitrides and hydrides. Different configurations used are cylindrical pellets, long extruded rods (metal fuels only), spherical elements (graphite matrix with coated particle dispersion fuel for the AVR-HTGR), dispersions in a matrix material (cermets), coated particles and fluids (molten salt reactor and aqueous homogeneous reactor). The most widely used fuel material in power reactors is uranium dioxide in the form of cylindrical, cold pressed and sintered pellets.

Plutonium, $_{92}Pu^{239}$, and Uranium, $_{92}U^{238}$, are the fissionable materials produced artificially from $_{92}U^{238}$ and Thorium, $_{90}Th^{232}$, respectively. Uranium-238 and Thorium-232 are available in nature. They are known as fertile materials. These *fertile materials* when placed close to $_{92}U^{235}$ in a reactor absorb the emitted neutrons to produce fissionable materials.

16.3.3 Nuclear Fuels Used in the Reactors

Nuclear reactions, namely *fission* and *fusion*, result in the generation of enormous amount of energy. In *fission reactions*, a heavy atom is split by neutrons into two lighter fragments. In *fusion reactions*, two lighter atomic nuclei are fused together to form a single, heavy nucleus. To carry out nuclear reactions, fissionable materials are used as fuels.

1. *Natural fuels*

Uranium is the only fissionable nuclear fuel occurring in nature. It consists of 99.3 per cent of $_{92}U^{238}$, 0.7% $_{92}U^{235}$ and small traces of $_{92}U^{234}$. Out of these isotopes, only $_{92}U^{235}$ is used in fission reaction.

2. *Artificial fuels*

Plutonium, $_{92}Pu^{239}$, and Uranium, $_{92}U^{238}$, are the fissionable materials produced artificially from $_{92}U^{238}$ and Thorium, $_{90}Th^{232}$, respectively. Uranium-238 and Thorium-232 are available in nature. They are known as fertile materials. These *fertile materials* when placed close to $_{92}U^{235}$ in a reactor absorb the emitted neutrons to produce fissionable materials as follows:

1. $_{92}U^{238} + {}_{0}n^{1} \rightarrow {}_{92}U^{239} + \gamma$ radiation

$_{92}U^{239} \rightarrow {}_{93}N_{p}{}^{239}$(Neptunium) + $_{-1}e^{0}$ (Electron)

$_{92}N_{p}{}^{239} \rightarrow {}_{94}Pu^{239} + {}_{-1}e^{0}$

2. $_{90}Th^{232} + {}_{0}n^{1} \rightarrow {}_{90}Th^{233} + \gamma$ radiation

$_{90}Th^{233} \rightarrow {}_{91}Pa^{233}$ (Protactinium) + $_{-1}e^{0}$ (Electron)

$_{91}Pa^{233} \rightarrow {}_{92}U^{233} + {}_{-1}e^{0}$

Advantages over solid fuels

(i) Under favourable conditions, the cost of generation of nuclear power is competitive with coal-based power generation.

(ii) Isotopes produced as by-products can be used in agriculture, medicine, scientific research and industrial processing.

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(iii) Used in transportation systems, especially in nuclear-powered submarines.

(iv) Complete fission of 1 kg of fissile uranium produces about 23×10^6 kWh of energy, which is equivalent to the energy generated by burning 3300 tons of bituminous coal.

Disadvantages

(i) The radiation emitted during the reaction is highly hazardous.

(ii) Nuclear waste disposal is a major problem and if not disposed off properly it is dangerous to all living things on the earth.

Different stages involved in fuel elements development are listed in Table 16.3.

Table 16.3 *Stages in Fuel Element Development**

1. Define the reactor type: purpose, coolant and performance.
2. Calculate fuel element dimensions, heat fluxes, design features, physics and critical assemblies.
3. Consider the range of available fuels and cladding materials and pick the one that is most suitable for the design.
4. Consider fuel and cladding properties, both out-of-reactor and in-reactor.
5. Examine fuel-cladding interactions (chemical and physical).
6. Develop a preliminary fuel element design.
7. Develop fabrication procedures for fuel, cladding and fuel element, if necessary.
8. Develop mathematical models of the fuel element; specify and obtain input data.
9. Test fuel elements: out-of-reactor (thermal-hydraulics), in-reactor (on increasing scale) and transient tests.
10. Analyse test: post-irradiation examination, failure mechanisms, run beyond cladding breach, feedback to models and improvements in design.
11. Optimize through 8, 9 and 10 iterations. Write detailed design and manufacturing specifications.
12. Plan production: quality assurance, NDE, SPM assay, safeguards, safety, critical control, economics and automation.
13. Establish interfaces with the rest of the fuel cycle: mining, enrichment, reprocessing and waste management.
14. Obtain license for qualified cores.

*From Frost, B.R.T. (1982). *Nuclear Fuel Elements*, Pergamon, New York.

16.3.4 Multiplication and Thermal Utilization Factors

Multiplication factor (k)

It is defined as the net number of neutrons produced per initial neutron accounting for all possible losses.

$$\text{If } k < 1 = \text{system is subcritical}$$
$$k = 1 > \text{system is critical}$$
$$k > 1 = \text{system is super critical}$$

Variation of k is made by neutron-absorbing rods or dispersed chemicals to increase or decrease neutron population. When the reactor is operated for a long period, enough fuel is consumed

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so that k goes below 1 regardless of adjustments of control materials. The reactor is shut down for refuelling.

Let η = number of neutrons/absorption in uranium
ξ = fraction not escaping by leakage
then $k = \eta\xi$

For fast neutrons, $\eta = 2.3$ and hence $\xi = 0.43$ for $k = 1$, that is, at least 43 per cent of neutrons must remain in the sphere for a reactor to go critical.

Thermal utilization factor

A moderator is used to slow down the neutrons in a reactor. These slowed down neutrons on achieving their lowest possible energy level are in thermal equilibrium with the molecules of that medium. The state of neutrons in this condition is said to be thermalized and neutrons are referred to as thermal energy neutrons.

As per the Maxwell–Boltzmann velocity distribution, the probable velocity of a thermal energy neutron is given by

$$V_m = \sqrt{\frac{2KT}{m_n}}$$

Substituting K = Boltzmann constant = 1.38×10^{-23} J/molecule – K

m_n = mass of neutron = 1.674×10^{-27} kg

T = absolute temperature, K

We get $$V_m = 128.4\sqrt{T}$$

The kinetic energy of such a neutron is given by

$$KE_m = \frac{1}{2}\times m_n \times V_m^2 = 8.613\times10^{-5} T \text{ eV}$$

Once thermalized, the neutrons continue to diffuse throughout the reactor and are subject to absorption by other materials in the reactor as well as the fuel. The thermal utilization factor is a tool that describes how effectively thermal neutrons are absorbed by the fuel. In other words, it gives an idea of how effectively neutrons are utilized within the reactor.

The *thermal utilization factor* (f) is defined as the ratio of the number of thermal neutrons absorbed in the fuel to the number of thermal neutrons absorbed in any reactor material.

$$f = \frac{\text{Number of thermal neutrons absorbed in the fuel}}{\text{Number of thermal neutrons absorbed in all reactor materials}}$$

The thermal utilization factor will always be less than 1 because some of the thermal neutrons absorbed within the reactor will be absorbed by atoms of non-fuel materials.

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16.3.5 Life Cycle of a Neutron

Fission reactions occur as an atom absorbs a nearby neutron. The atom becomes unstable and splits into two semi-stable atoms, releasing neutrons and a tremendous amount of heat. Let η be the fast neutrons created by a fission reaction. Fast neutrons contain a large amount of energy and travel at a speed of more than 14,000 km/s (30 million mph).

A fast neutron could cause a fission reaction of its own in a process called fast fission. Let the probability of this be denoted by ε known as the fast fission factor. On the contrary, this fast neutron could leak out of the reactor, never to be available for fission reaction; the probability of this not occurring is denoted by P_{NL}^{f}, known as the fast non-leakage probability. After all, once the neutron escapes, it will not return to the reactor and is worthless to sustain the reaction. The third possibility is that this fast moving neutron is slowed down using a moderator to about 2200 m/s, by a process known as thermalization. As the neutron bounces off moderator atoms, it transfers its energy to the moderator atoms, increasing their speed. This results in a rise in the temperature of the moderator, generating steam, which subsequently is used to drive a turbine. However, although this neutron is slowing down, it has to survive the resonance region as shown in Figure 16.9.

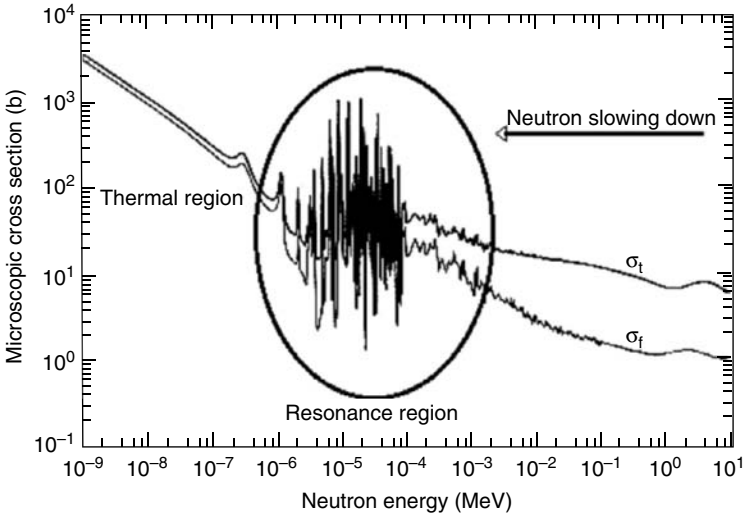


Fig. 16.9 *Microscopic Cross Section versus Neutron Energy*

The resonance region is a very chaotic region where the probability of a neutron being absorbed fluctuates from very large values to very small values. The probability that the neutron slows to thermal energies without being absorbed is known as the resonance escape probability, and is denoted by p. After the resonance region, the neutron has a much lower energy than before. Figure 16.10 shows Life cycle of a neutron.

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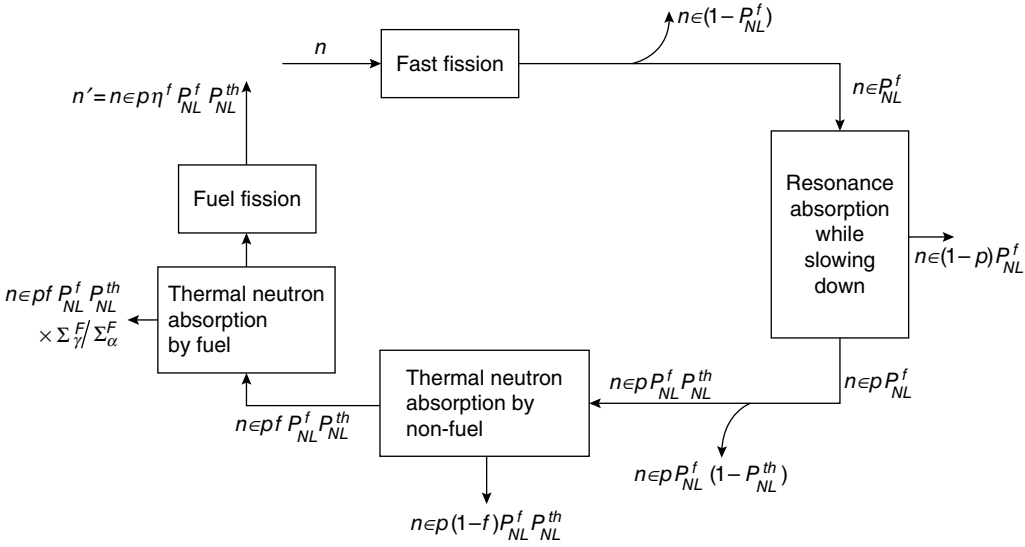


Fig. 16.10 *Life Cycle of a Neutron*

16.4 NUCLEAR REACTOR COMPONENTS

Components that are most commonly used in most nuclear reactors for power generation are shown in Figure 16.11.

1. *Fuel*

In a thermal nuclear reactor, the fission induced by neutrons using a fuel isotope U^{235} is one of the several isotopes used in nuclear power generation. Fuel isotopes that are capable of being fissioned by thermal neutrons are called fissiles, e.g. U^{235}, Pu^{239} and Pu^{241}. These are not available in nature but are produced by fission of fertile isotopes, e.g. ${}_{90}Th^{232}$, ${}_{92}Th^{238}$ and ${}_{94}Th^{240}$. A typical nuclear reaction involving fuel isotope is

$$U^{235} + {}_{1}n^{0} \rightarrow {}_{37}R_{b}{}^{94} + {}_{55}C_{s}{}^{140} + 2{}_{0}n^{1} + \text{Heat (2 MeV)}$$

Fuel isotope + Neutron → Fission fragments + Neutrons + Energy

2. *Moderator*

The function of a *moderator* is to reduce the energy of fast neutrons to thermal neutrons. Due to high energy of fission neutrons (2 MeV) relative to that required to trigger another fission event (0.0025 eV), their probability of interacting with another U^{235} is small. The probability of an interaction of a neutron and a bombarded nucleus is referred to as *neutron cross section*. The moderator slows down the neutrons to thermal energies by collision with inert atoms (scattering). Speed of the neutrons is reduced within a small number of collisions as the moderator possesses a high scattering cross section. Materials with low atomic mass number make best moderators.

e. g. Light water, heavy water, carbon, beryllium

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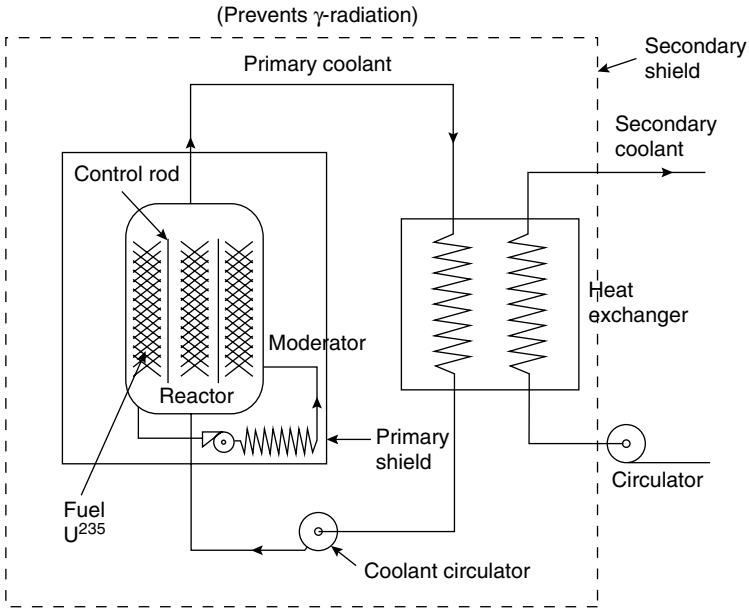


Fig. 16.11 *Nuclear Power Plant*

Desirable properties of a moderator

(i) It must be as light as possible as slowing down action is more effective in elastic collision with light elements.

(ii) It should not absorb neutrons but slow down the neutrons as early as possible.

(iii) It should have high resistance to corrosion as it has to work under high pressure and temperature.

(iv) It should have good machinability if used in solid form and should have high melting point.

(v) It should be chemically stable and should not be decomposed due to nuclear radiation.

(vi) It should have high thermal conductivity for better heat transfer.

3. *Coolant*

The function of the coolant is to remove the heat released by fission. The coolant should have high specific heat, high conductivity, good chemical stability, good pumping characteristics and low neutron-absorption cross section. Coolant can be either liquid or gaseous.
e.g. Light water, heavy water, air, CO_2, He, sodium, bismuth, potassium, organic.

4. *Control rods*

Control rods are normally made of cadmium, boron or hafnium. They have huge neutron-absorption cross sections. The control rods are lowered or raised in the reactor core. Since the reactor power is directly proportional to neutron density, lowering the control rods will remove neutron from the reactor core and will decrease the power and reaction rate. Raising the control rods will increase the power and rate.

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5. *Shielding*

Shielding prevents the passage of radiation to the outside of the reactor. The primary shield prevents the leakage of neutron and gamma radiation present in the cooling circuits due to activation of the coolant as it passes through the core. The kind and the amount of shielding required depend on the type of radiation and intensity. A low atomic weight material for neutrons and a high atomic weight material for gamma rays are used. Shield is frequently constructed in layers or heavy and light material such as concrete and water. Shields for external circuit where only gamma radiation may be present are made up of steel, lead, polyethylene and concrete. Concrete is used mostly due to its low cost.

6. *Reflector*

Function of the reflector is to minimize the neutron leakage by reflecting them back into the reactor. The material used for reflector is same as that used for moderator.

7. *Structure*

Structure provides physical support to the reactor, its components and containment of fuel elements. Aluminium, steel, zirconium and stainless steel are used for this purpose.

16.5 CLASSIFICATION OF NUCLEAR REACTORS

Nuclear reactors are classified based on the following factors:

1. *Type of fission*
   (i) Fast reactor – Fission is caused by fast neutron. The reactor has no moderator. Reactor core size is less.
   (ii) Thermal reactor – Fission is caused by slow or thermal neutrons.
   (iii) Intermediate reactor.
2. *Types of fuel*
   (i) Natural fuel – Using natural uranium as fuel and heavy water and graphite as moderator.
   (ii) Enriched uranium – Using 5–10 per cent of U^{235} and ordinary water as moderator.
3. *Fuel cycles*
   (i) Burner reactor – Produces only heat
   (ii) Converter reactor – Fertile material is converted into fissile material
   (iii) Breeder reactor – Fertile material is converted into initial fissile material

   e.g. Natural uranium is the fuel; thorium is converted into U^{233}
   Plutonium is the fuel; U^{238} in converted into plutonium
4. *Position of fissile and fertile material*
   (i) One-region reactor – In this reactor, both fissile and fertile materials are mixed.
   (ii) Two-region reactor – In this reactor, both fissile and fertile materials are separately placed (Figure 16.12).

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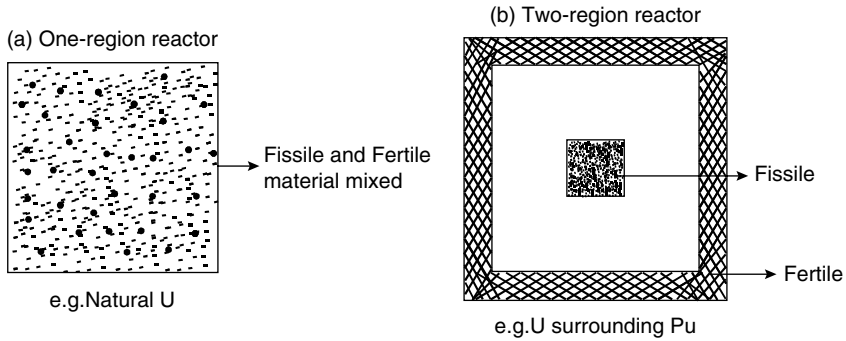


Fig. 16.12 *Position of Fissile and Fertile Material*

5. *State of fuel*
   (i) Solid state – liquid metal fuelled reactors
   (ii) Aqueous homogeneous reactor – liquid state
6. *Choice of moderator*
   (i) Graphite – higher atomic weight and bulky
   (ii) Natural water, enriched Uranium
   (iii) Natural Uranium
7. *Core composition*
   (i) Homogeneous reactor – If the fuel and moderator are mixed, it is called a homogeneous reactor.
   (ii) Heterogeneous reactor – If the fuel and moderator are separate, it is called a heterogeneous reactor. This reactor contains large number of fuel rods and the coolant circulating them carries the heat generated.

Figure 16.13 shows the difference between homogeneous and heterogeneous reactors.

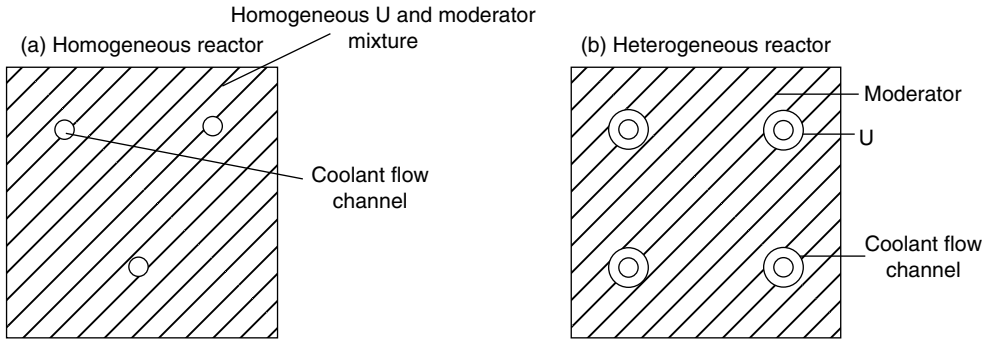


Fig. 16.13 *Classification Based on Core Position*

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8. *Method of cooling*

(i) *Direct system of cooling*

In this system, heat generated in the reactor is utilized by a liquid fuel, which exchanges heat with water circuit (Figure 16.14).

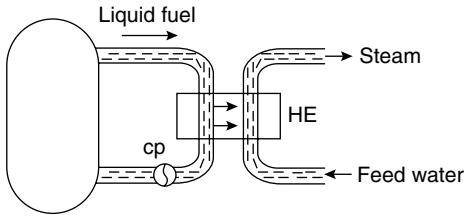


Fig. 16.14 *Direct System Liquid Fuel*

(ii) *Indirect system of cooling*

In this system, heat generated by the solid fuel is transferred to the coolant, which in turn exchanges heat with the water circuit (Figure 16.15).

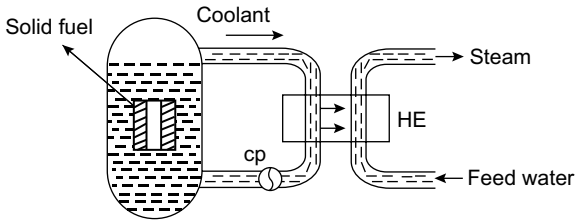


Fig. 16.15 *Indirect System*

9. *Coolant used*

(i) *Gas-cooled reactor – Air, H_2, He, CO_2*

- ➤ Uses natural uranium, graphite moderator
- ➤ Low pressure coolant, high reactor temperature
- ➤ Large size reactor
- ➤ Low power density (kW/l of core volume)
- ➤ Steam pressure and temperature low
- ➤ High pumping power

(ii) *Water-cooled reactors*

Water is used as coolant and moderator.

(a) *Light water (LW) (ordinary water) reactor*

These are the thermal reactor systems in which water serves as both the coolant and the moderator.

- ➤ High hydrogen concentration
- ➤ Good thermal properties
- ➤ Cheapest coolant and moderator

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- Cooling system is simple
- Hot water is corrosive
- Must be highly pressurised to operate at moderate temperature
- Costly reactor vessel, leak proof primary cooling circuit
- Enriched Uranium (2–3 per cent) is used

(b) *Heavy water (water) reactor (HWR)*

These are the thermal reactor systems in which ordinary water serves as the coolant and the heavy water as moderator or heavy water serves as both the coolant and the moderator.

- Low electron cross section
- Good neutron economy
- High fuel burn-up, lower fuel cost
- High initial cost and limited by critical temperature

16.6 THERMAL FISSION REACTORS AND POWER PLANT AND THEIR LOCATION

The most important nuclear power reactors used at present include the following:

1. *Light-water (LW) moderated and cooled reactors*

These may be either pressurized water (PWR) or boiling water reactor (BWR). In PWR, operating at about 14 MPa pressure and 300°C, the heat is transferred from the core to steam generators through intermediate heat exchangers. In BWR, operating at about 7 MPa and 300°C, the coolant water boils at the top of the core and supplies steam directly to the turbines. The fuel used is slightly enriched UO_2.

2. *Heavy-water moderated reactors (CNDU)*

In these reactors, the heavy-water moderator is contained in a calandria. Insulated pressure tubes containing the fuel elements circulate the pressurized light-water coolant in the calandria at 15 MPa and 300°C and transfer the heat from the fuel elements to steam generators. The fuel used is natural UO_2.

3. *Carbon-dioxide gas-cooled graphite-moderated reactors*

The first-generation reactors (Magnox) of these types are cooled by circulating CO_2 gas. The fuel elements use natural uranium metallic fuel rods clad with a magnesium alloy. In the second-generation advanced gas-cooled (AGR) reactors, stainless steel-clad slightly enriched UO_2 fuel rods are used. These reactors are used to generate steam at higher temperatures.

4. *High-temperature helium gas-cooled reactors (HTGR)*

In these reactors, graphite serves as moderator, reflector and core structure material. Coated-particle oxide or carbide fuel is used. The helium gas coolant (700–1000°C and 5–8 MPa) transfers heat to steam generators.

5. *Liquid-metal cooled fast breeder reactors (LMFBR)*

These reactors use liquid sodium in the primary system that transfers the heat from the core to an intermediate heat exchanger. From here, sodium transfers heat to the steam generator. The fuel consists of (U, Pu)O_2 pellets contained in stainless steel cladding.

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Location of nuclear power plant
To locate a nuclear power plant, geological, meteorological, hydrological, topographical special transport and radiological investigations are essential. Some of the major factors to locate a nuclear station considered are

(i) availability of cooling water
(ii) transportation facilities
(iii) distance from load centre
(iv) safety
(v) radioactive waste, disposal facility
(vi) foundation requirement

Table 16.4 shows major component materials used in different reactor systems.

Table 16.4 *Major Component Materials Used in Different Reactor Systems*

Reactor component	BWR	PWR	CANDU	AGR	LMFBR
Core fuel/cladding	UO_2/Zircaloy	UO_2/Zircaloy	UO_2/Zircaloy	UO_2/SS	UO_2/SS UO_2, 25 ± 5% PuO_2/SS
Control materials	B_4C/Type 304 SS UO_2-Gd_2O_3/ Zircaloy 2	AgInCd alloy B4C-Al_2O_3; borosilicate glass	B_4C/SS	B_4C/SS	B_4C/SS
Breeding blanket	NA	NA	NA	NA	UO_2/Type 316 SS
Reactor vessel shell	Low-alloy steel (SA533 Gr. B)	Low-alloy steels (SA533 Gr. B, SA508)	(SA533 Gr. Zircaloy tubes in Al calandria	Pre-stressed concrete	Type 304 SS
Cladding	Type 308L SS (SA264)	Type 308 SS: Inconel 617			NA

16.6.1 Pressurized Water Reactor

A pressurized water reactor (PWR) power plant is composed of two loops in series as shown in Figure 16.16. One is the coolant loop called *primary loop* and the other is the water steam or *working fluid loop*.

Referring to Figure 16.17, hot water from the reactor flows to a steam generator where heat is transferred to water and steam in the secondary cooling system, which operates at a lower pressure. The primary coolant then flows from the steam generator to the primary cooling pump, where it is pumped back to the reactor. The pressurizer is a pressure vessel with a heater at the bottom and a water spray at the top. The top of the pressurizer is filled with steam at primary system pressure (15.5 MPa). If the primary loop pressure drops, the heater is energized to increase the steam content in the pressurizer and thus increases the pressure of primary cooling system. If the primary system pressure is too high; cold pressurized water is sprayed into the steam, condensing the steam and hence reducing the primary system pressure.

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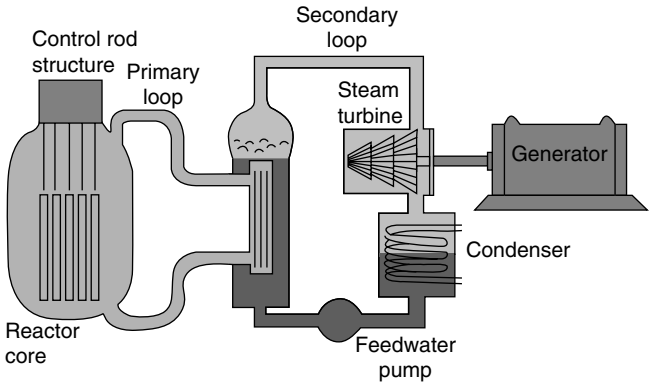


Fig. 16.16 *A PWR Power Plant*

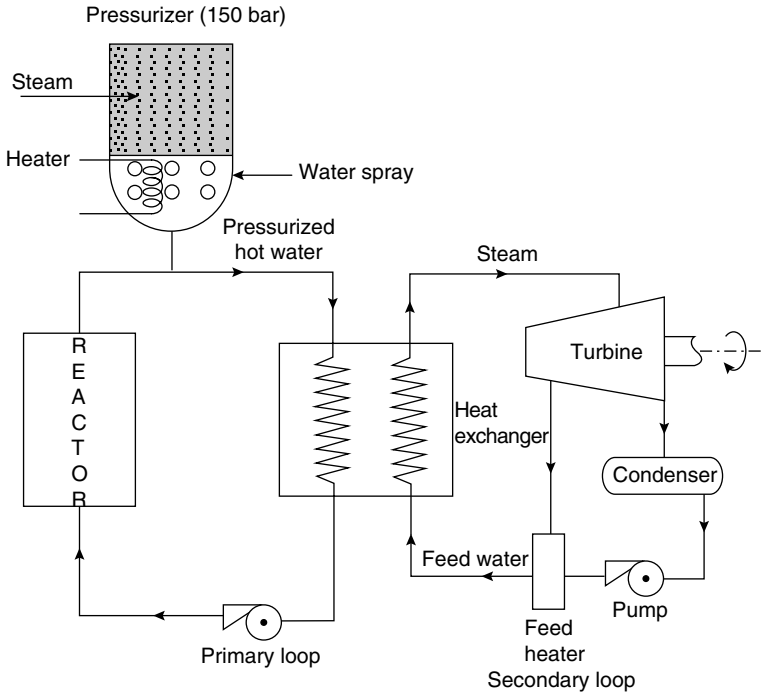


Fig. 16.17 *Schematic of a PWR Power Plant*

Advantages

(i) Due to high negative temperature coefficient, a PWR is safe and stable in operation and regulation.

(ii) Control rods need not be used for load change except only during start-up and slow-down reactions.

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(iii) It has a positive power demand coefficient (due to negative temperature coefficient) and responds greatly to more power demand.

(iv) Uses enriched fuel making the reactor more compact in size.

(v) It provides complete freedom to inspect and maintain the turbine, feed heaters and condenses during operation.

(vi) Water, which is used as a coolant, moderator and reflector is cheap in first cost and available in plenty.

(vii) It reduces the fuel cost by extracting more energy per unit weight of fuel.

(viii) A PWR is ideally suited to the utilization of fuel designed for higher burn-ups.

Disadvantages

(i) High primary circuit pressure requires a strong pressure vessel with subsequent high capital cost.

(ii) Due to low pressure in the secondary circuit (60–70 bar), thermodynamic efficiency is lower (20 per cent).

(iii) Reprocessing of fuel is difficult as it suffers radiation damage.

(iv) Presence of high temperature, high-pressure water and corrosion is severe. This means use of stainless steel for vessel and cladding, which further adds to plant cost.

(v) When γ-radiations pass through the pressure vessel, it results in uneven heating, which induces thermal stresses on the vessel.

16.6.2 Boiling Water Reactor (BWR)

In the BWR, the coolant is in direct contact with the heat-producing nuclear fuel and boils in the same compartment in which the fuel is located. The reactor pressure is maintained at 70 bar. As water and vapour coexist in the core, a BWR produces saturated steam at about 285°C. This operating temperature gives a Carnot efficiency of only 42 per cent with a practical operating efficiency of around 32 per cent lesser than the PWR. The coolant thus serves the triple function of coolant, moderator and working fluid. The disadvantage of this reactor is that any fuel leak might make the water radioactive and the radioactivity would reach the turbine and the rest of the loop in all probability (Figure 16.18).

A BWR power plant consists of a reactor, a turbine generator, a condenser and air ejector, a cooling system, etc.

Slightly sub-cooled liquid enters the reactor core at the bottom receiving sensible heat to saturation and some latent heat of vapourization. When it reaches the top of the core, it is converted into a very wet mixture of liquid and vapour. The vapour separates from the liquid, flows to turbine, does work, condenses in the condenser and is then pumped back to the reactor by feed pump.

The saturated liquid separating from the vapour at the top of the reactor (steam separator) flows downwards through the down comer either internally (Figure 16.19(a)) within the reactor or externally (Figure 16.19(b)) outside the reactor and mixes with the return condensate. The flow through the down comer may be either natural or by means of a recirculating pump. The ratio of recirculation liquid to the saturated vapour produced is called recirculation ratio. It ranges between 6 and 10.

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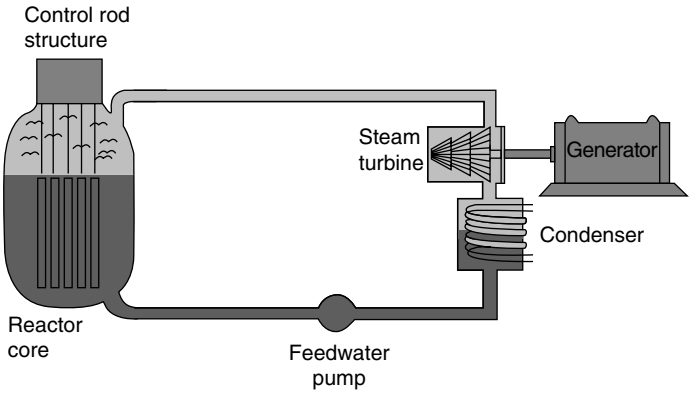


Fig. 16.18 *A BWR Power Plant*

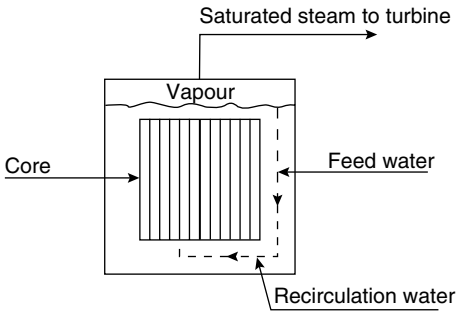


Fig. 16.19(a) *Internal Recirculation of BWR*

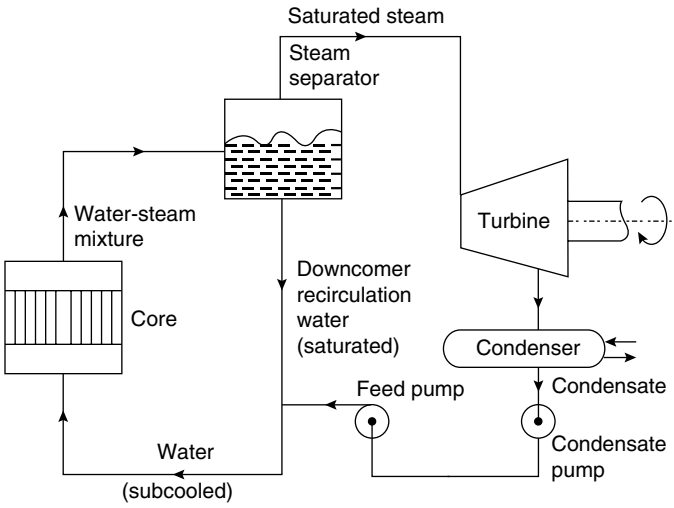


Fig. 16.19(b) *External Circulation of BWR*

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16.6.2.1 Boiling Water Reactor Cycles

Three different BWR cycles commonly used are as follows:

1. *Single-cycle internal circulation*

This system shown in Figure 16.20 uses internal recirculation system to generate power ranging from 50 to 100 MW.

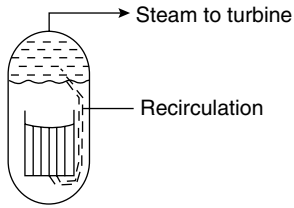


Fig. 16.20 *Single-Cycle Circulation*

2. *Single-cycle forced circulation*

This type shown in Figure 16.21 uses a forced circulation system using a circulating pump. The heat in the reactor is used to generate steam. It is useful to generate power ranging from 100 to 1000 MW.

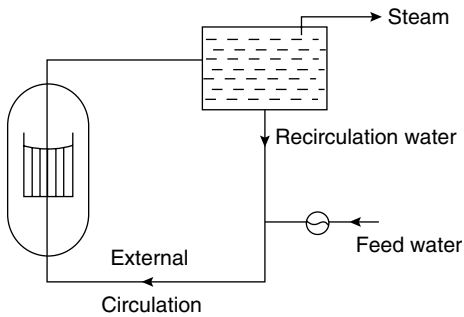


Fig. 16.21 *Forced Circulation (External)*

3. *Dual-cycle forced circulation*

This arrangement shown in Figure 16.22 has two cycles: steam and gas.

Advantages

(i) The reactor vessel is much lighter compared to a PWR as pressure inside the reactor vessel is considerably smaller than the PWR. This means cost of pressure vessel is less.

(ii) The reactor does not require steam generator, pressurizer circulating pump and connecting piping. This reduces the plant cost further.

(iii) Since boiling is allowed inside the reactor, the metal surface temperature is lower than PWR.

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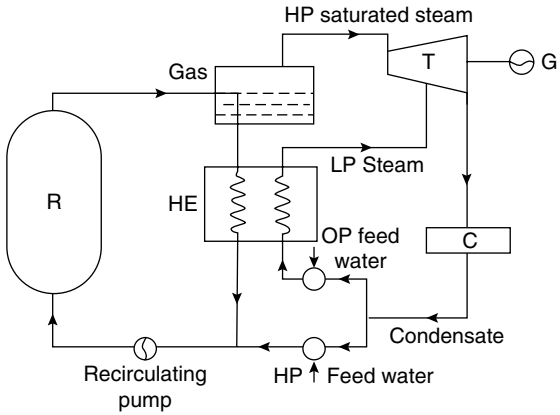


Fig. 16.22 *Dual-Cycle Forced Circulation*

(iv) Thermal efficiency of this reactor plant is considerably higher than PWR plant.

(v) BWR is more stable than PWR and much stable than any other type of reactor. It is also called as self-controlled reactor.

Disadvantages

(i) Light shielding of turbine and piping is necessary. Steam leaving the reactor is slightly radioactive.

(ii) The size of the vessel will be considerably less compared to PWR.

(iii) The possibility of 'burn out' of fuel is more in this reactor than PWR as boiling of water on the surface of fuel is allowed.

(iv) If it is not properly designed, the BWR might have negative power demand coefficient; when more power is demanded from reactor it may produce less.

16.6.3 CANDU Heavy Water Reactor

Heavy water composed of heavy hydrogen isotope 2H is used as a moderator and coolant in some power and research reactors. These reactors use heavy water as moderator and primary coolant and light water as secondary coolant. The Canadians have exported such reactors known as CANDU reactors.

The CANDU reactor (Figure 16.23) consists of a primary loop and a secondary loop. Fuel is burnt in the reactor releasing heat energy so that the primary coolant (heavy water) coming in contact with the reactor core gets heated up. The heated primary coolant is taken to the steam generator boilers where it gives the heat energy to the light water condensate. Steam is generated due to heat transfer and is used for power generation. The pressurizer maintains constant pressure of the primary loop system. Due to condensation, if the primary line pressure falls below the rated pressure, the heater gets activated heating the heavy water and thus increasing the line pressure due to formation of vapour. The primary coolant is pumped back to the reactor core using primary pumps. A separate system of heavy water is used as a moderator to reduce the energy of fast neutrons to thermal neutrons.

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Figure 16.24 shows how the steam from the collectors are further fed to the HP and LP turbines to generate power.

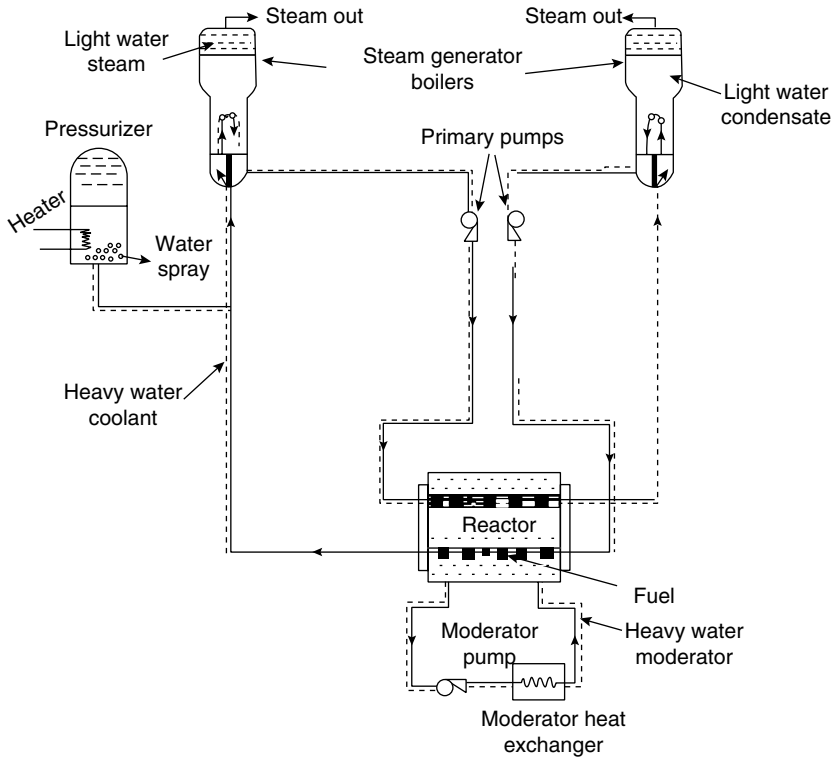


Fig. 16.23 *CANDU Reactor*

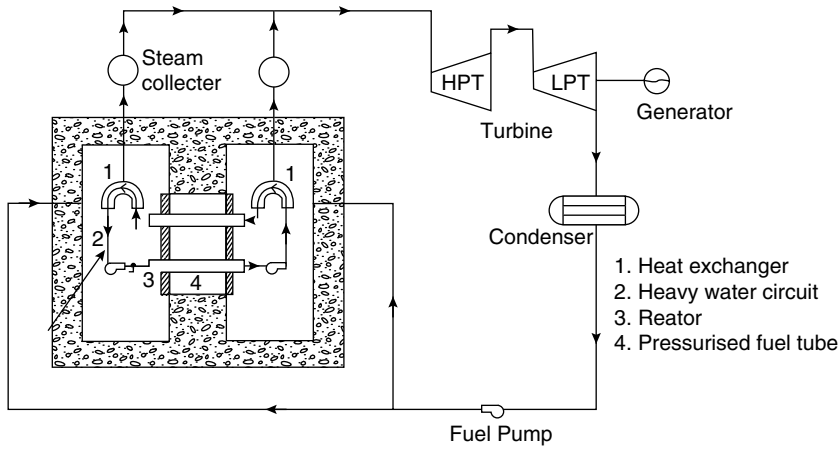


Fig. 16.24 *Simplified Steam Cycle of CANDU Reactor*

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Figure 16.25 shows fuel assembly of a pressurized heavy-water reactor (PHWR-CANDU). The main structural materials used are zirconium and aluminum alloys. The use of heavy water as a moderator provides good neutron economy and permits a wide range of possible fuel cycles (including Th, ^{233}U or U, Pu) and fuel management schemes. The UO_2 fuel elements are placed in Zircaloy-2 pressure tubes, which pass through an aluminum calandria containing the heavy water moderator another form of CANDU reactor working on the same principle.

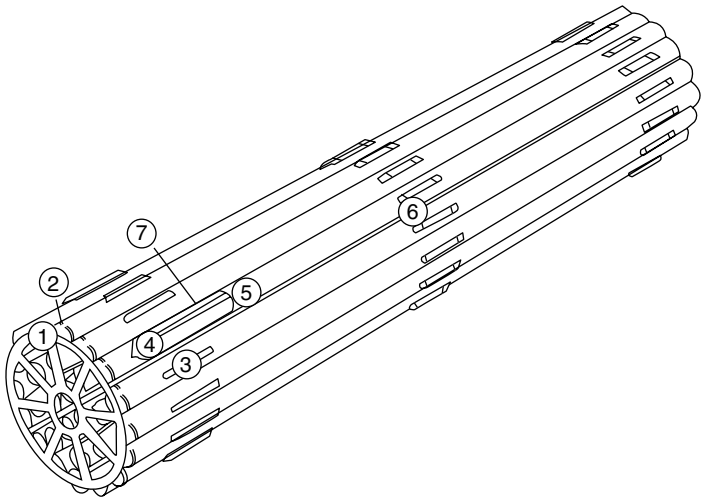


Fig. 16.25 *Fuel Bundle for CANDU Reactor* [1]

Note: (1) Zircaloy structural end plate; (2) Zircaloy end cap; (3) Zircaloy bearing pads; (4) VO_2 pellets; (5) Zircaloy fuel sheath; (6) Zircaloy spacers; and (7) graphite coating

Advantages

(i) The fuel need not be enriched.
(ii) No control rods are required; therefore, control is much easier than other types of reactors.
(iii) Fuel tubes are designed to withstand high pressure, therefore cost of the vessel is less.
(iv) Moderator will be kept at low temperature, which increases its effectiveness to slow down neutrons.
(v) This type of reactor has low fuel consumption and higher multiplication factor.
(vi) Time required for site construction will be less compared to PWR and BWR.

Disadvantages

(i) Loss of heavy water is high.
(ii) Leakage is a major problem as there are two mechanically sealed closure per fuel channel.

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(iii) Compared to PWR and BWR power density is considerably low; therefore, the reactor size is extremely large.

(iv) Very high standard of design, manufacture, inspection and maintenance are required.

16.6.4 Gas-Cooled Reactor

Gas-cooled reactors, namely Magnox and AGR belong to very earliest reactor designs in that they are graphite moderated and gas cooled. The early gas-cooled reactors popularly known as Magnox reactors (Figure 16.26) were built in the UK during 1956–1971, but have now been superseded. This reactor used the magnesium alloy to encase the fuel (natural uranium) and hence named after this. Fuel elements consisting of fuel rods encased in Magnox cans. These in turn are loaded into vertical channels in a core constructed of graphite blocks. To adjust the rate of the fission process and the heat output, control rods (strong neutron absorbers) in the vertical channels are either inserted or withdrawn from the core. The whole assembly is cooled by blowing carbon dioxide gas past the fuel cans, which are specially designed to enhance heat transfer. The hot gas converts water to steam in a steam generator. Although the early designs used a steel pressure vessel, which was surrounded by a thick concrete radiation shield, the later designs were constructed with a dual-purpose concrete pressure vessel with a radiation shield.

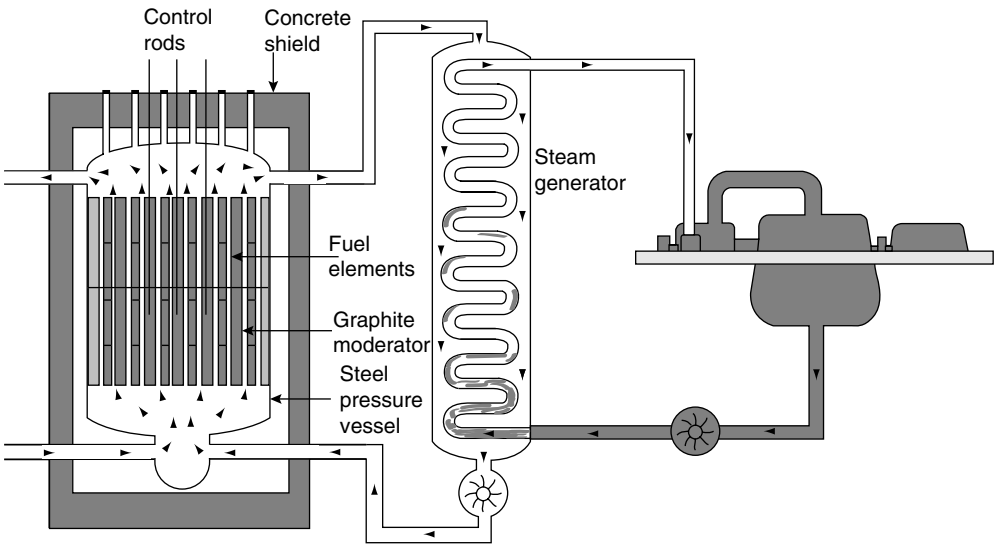


Fig. 16.26 *Gas-Cooled Reactor (Magnox)*

To improve cost effectiveness, higher thermal efficiencies and higher power densities, these reactors were modified to operate at higher temperatures. This system used higher cooling gas pressure and stainless steel cladding with uranium dioxide fuel. This reactor is known as the Advanced Gas-Cooled Reactor, or AGR, and uses graphite as the moderator. As shown in Figure 16.27, the steam generators and gas circulators are placed within a combined concrete pressure-vessel/radiation-shield.

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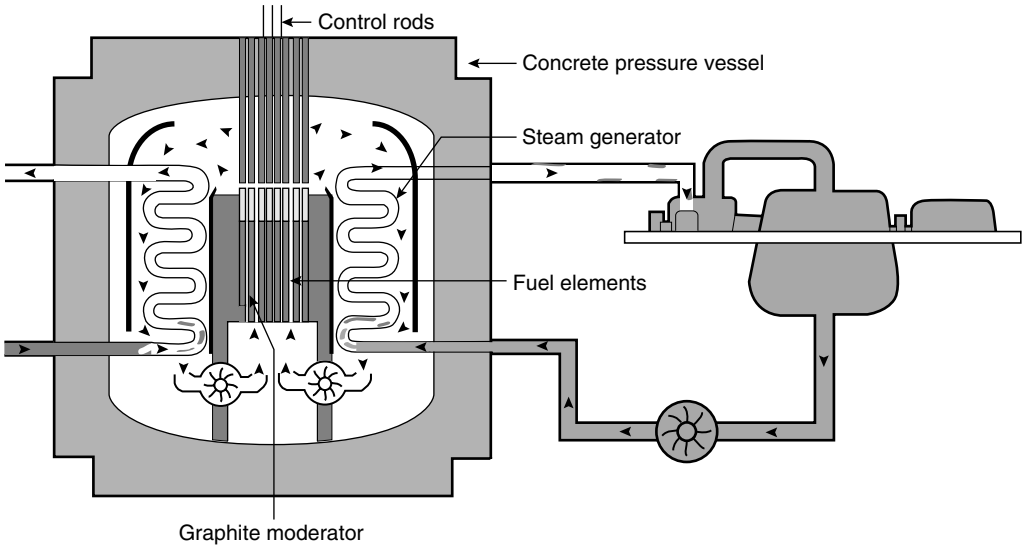


Fig. 16.27 *Gas-Cooled Reactor (AGR)*

16.6.5 Fast Breeder Reactors

If the reactor produces more fissionable material than it consumes, it is called a breeder reactor. Breeder fuel consists of both fertile and fissile materials. The number of neutrons released is sufficient to propagate the fission reaction and to produce more fissionable material by conversion of fertile isotopes to fissile isotopes. No moderator is used in the breeder reactor since fast neutrons are more efficient in transmuting U-238 to Pu-239.

The Pu-239 breeder reactor is commonly called a fast breeder reactor, and the cooling and heat transfer are done by a liquid metal. The metals used for this purpose are sodium and lithium. As sodium is most abundantly available it is most commonly used. The construction of the fast breeder requires 15–30 per cent higher enrichment of U-235 than a light-water reactor. The reactor fuel is surrounded by a 'blanket' of non-fissionable U-238. At this concentration of U-235, the cross-section for fission with fast neutrons is sufficient to sustain the chain reaction.

The two most commonly used breeding cycles are

(i) conversion of fertile U^{238} to fissile Pu^{239}

(ii) conversion of fertile Th^{232} to fissile U^{233}

Since uranium cycle uses high-energy neutrons, it is called as fast breeder reactor. These reactors use liquid sodium metal or pressurized helium as a coolant and no moderator to slow down the neutrons.

The following three types of fast breeder reactors are in use:

(i) Liquid metal fast breeder reactor (LMFBR)

(ii) Gas-cooled fast breeder reactor (GCFBR)

(iii) Molten salt breeder reactor (MSBR)

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16.6.5.1 Liquid Metal Fast Breeder Reactor (LMFBR)

The fuel consists of 80 per cent by weight of UO_2 and 20 per cent by weight of PuO_2 in small diameter stainless steel-clad tubes operating at temperatures of 670–700°C. The coolant used is sodium, which is a solid at room temperature but liquefies at 98°C. It has a wide working temperature since it does not boil until 892°C. The range of operating temperatures for the reactor falls within this so that it does not need to be pressurized as in a water–steam coolant system. It has large specific heat and hence efficiently transfers heat. Fuel assemblies and clusters of fuel pins are immersed in liquid sodium coolant flowing at low pressure through the reactor, entering the reactor core at 400°C and leaving at 620°C. Since primary sodium becomes intensely radioactive in the reactor, a secondary nonradioactive sodium coolant loop is used (Figure 16.28).

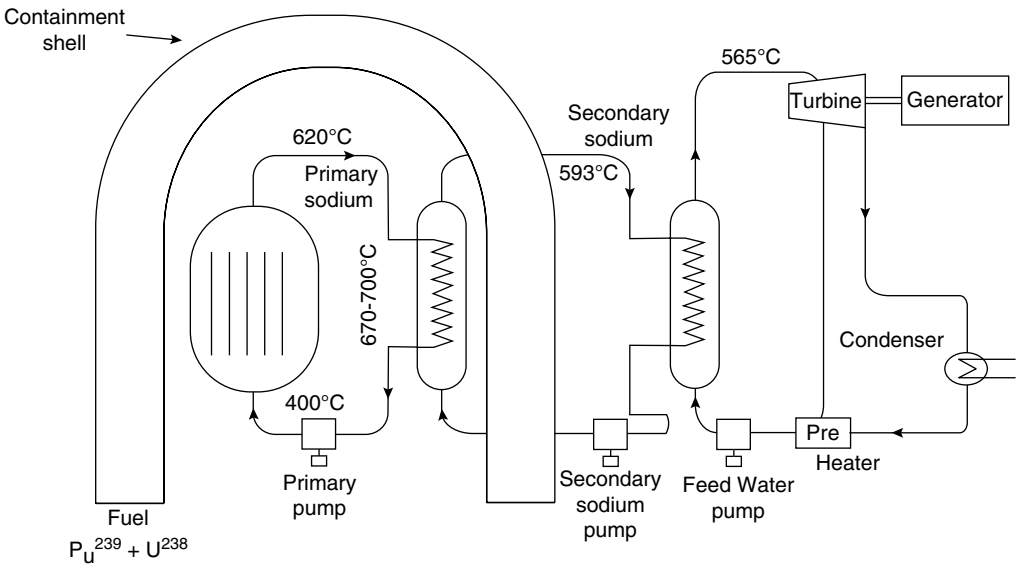


Fig. 16.28 *Liquid Metal-Cooled FBR*

The primary loop of the system may be either pool type (Figure 16.29) or loop type (Figure 16.30). In the pool-type system, the entire primary system, namely the reactor, primary heat exchangers, pumps are submerged in a large tank filled with molten sodium. The coolant is directly discharged from the heat exchanger to the tank and pumped from there further. This system is relatively insensitive to sodium leakage in the primary system, and compact also. However, the system is rather complex as the pool enclosure includes shield, inert gas closure, etc.

In the loop-type system, the heat exchangers, liquid metal pumps and other components of the primary system and the interconnecting piping arrangements are separated within a large building filled with an inert gas to prevent fire hazard from sodium leakage.

Advantages of using a liquid metal are

(i) excellent heat transfer
(ii) low-absorption cross section

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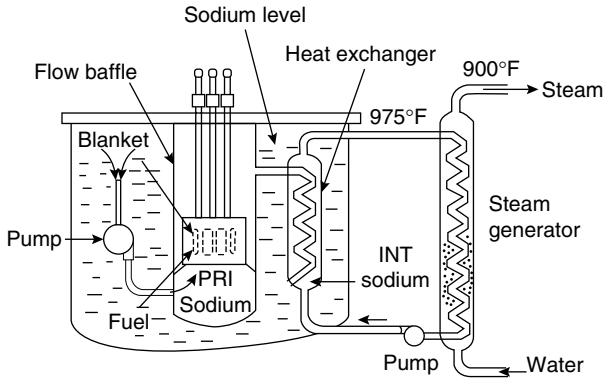


Fig. 16.29 *Pool Type*

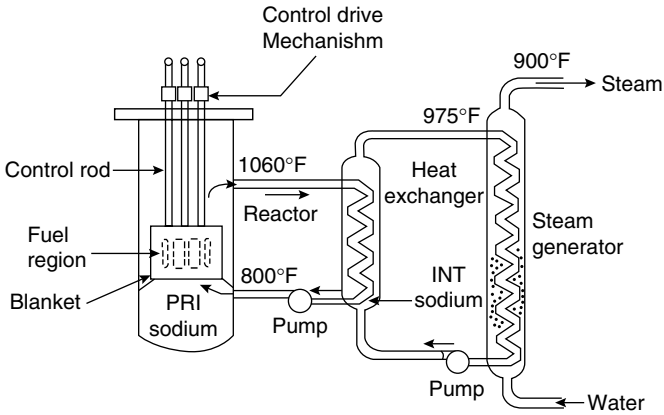


Fig. 16.30 *Loop Type*

(iii) low melting point
(iv) high BP
(v) highly specific
(vi) high thermal conductivity
(vii) cheap

Advantages

(i) High temperature can be achieved in the cycle, therefore thermal efficiency is high.
(ii) Low-cost graphite moderator can be used as it can retain its mechanical strength and purity at high temperatures.
(iii) Sodium as a coolant need not be pressurized.
(iv) The reactor size is comparatively small.
(v) It is best suited to thermal reactor with slightly enriched fuel.

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Disadvantages

(i) Leak of sodium is very dangerous as it comes out of reactor in a highly radioactive state.
(ii) Care must be taken to see that sodium does not come in contact with water as it becomes highly radioactive forming caustic soda with evolution of heat.
(iii) It is necessary to shield the primary and secondary cooling systems.
(iv) It is always necessary to keep the graphite and sodium separate.

16.6.5.2 Gas-Cooled Fast Breeder Reactor (GCFBR)

The GCFBR (Figure 16.31) is cooled by helium gas at 85 bar. Also, helium does not become radioactive under neutron bombardment and hence secondary coolant is not needed.

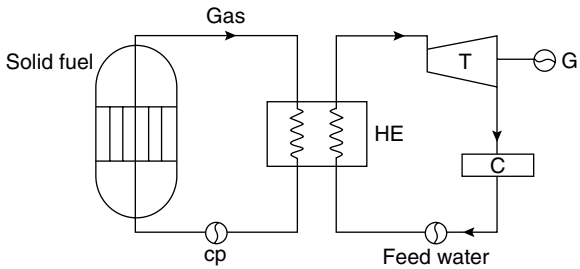


Fig. 16.31 *Gas-Cooled Reactor*

Advantages of using helium are as follows

(i) Using helium as a coolant poses less severe metallurgical and safety problems.
(ii) Helium is chemically inert, does not become radioactive and does not degrade the neutron spectrum. This leads to high conversion factor and negligible void reactivity coefficient.
(iii) Heat transfer coefficient of helium is much superior than sodium with artificial roughening of fuel rod surfaces.
(iv) Low doubling time (10–12 years) as compared to sodium-cooled reactors (20 years).

Disadvantages

(i) Reactor vessel needs to be pressurized to 70–80 bar.

Advantages

(i) Fuel processing is simple, as cladding problem of metallic fuel does not exist.
(ii) As uranium carbide and graphite can withstand high temperatures, limiting the fuel element temperature is not as serious as in other reactors.
(iii) Possibility of explosion in the reactor is avoided due to the use of CO_2 as coolant, which is a major problem in water-cooled plants.
(iv) Corrosion problem is eliminated.
(v) Due to low parasitic absorption, neutron economy is better.
(vi) Graphite remains stable under irradiation at high temperatures.

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Disadvantages

(i) Large vessel is required as power density is very low due to low heat transfer coefficient.
(ii) Leakage of gas is a major problem if helium is used instead of CO_2.
(iii) Loading of fuel is very elaborate and costly.

16.6.6 Organic Substance Cooled Reactor

Figure 16.32 shows an organic substance cooled reactor used in Piqua city, Ohio, USA. It uses enriched uranium as fuel and a mixture of orthometaterhenyl and paratherphenyl as coolant and moderator, and boron control rods, generating 11.4 MW capacity with an overall efficiency of 25 per cent.

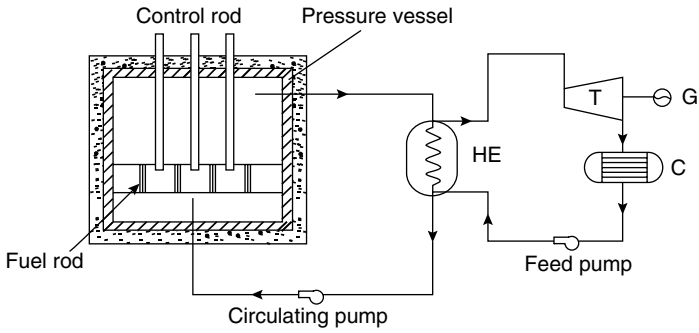


Fig. 16.32 *Organic Cooled and Moderated Reactor*

Advantages of using organic substance are

(i) Core operates at relatively high temperature at moderate pressure.
(ii) Less corrosive, hence low carbon steel can be used for piping. This makes low initial cost.
(iii) Steam does not become radioactive.
(iv) Do not freeze at low temperature.

Disadvantages

(i) Decompose when exposed to neutron, γ-radiation, leaving slurry deposits on fuel can surfaces reducing heat transfer.
(ii) Poor heat transfer properties as compared to water.
(iii) Highly inflammable.

Advantages

(i) Compact core design is possible as organic liquid is used as both coolant and moderator.
(ii) Core operating pressure is low due to high temperature at low vapour pressure.
(iii) A wide variety of fuels as uranium, uranium alloy, oxide or uranium carbide can be used with organic coolants due to their excellent moderating properties.
(iv) Low carbon steel can be used for vessel and piping as organic fluid does not corrode the tubes. This reduces initial capital cost.

Table 16.3 shows the summary of different types of thermal reactors.

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Table 16.3 *Summary of Different Types of Thermal Reactors*

Type	Fuel	Moderator	Coolant			Spent fuel processing	Steam cycle efficiency (%)	Main economic and safety characteristics
			Heat extraction	Outlet Temp (°C)	Pressure (psi)			
Magnox	Natural uranium metal (0.7% U^{235}) Magnesium alloy cladding	Graphite	Carbon dioxide gas heated by fuel raises steam in steam generator	360	300	Typically within one year, for operational reasons	31	Safety benefit that coolant cannot undergo a change of phase. Also ability to refuel whilst running gives potential for high availability
AGR	Uranium dioxide enriched to 2.3% U^{235} Stainless steel cladding	Graphite	Carbon dioxide gas heated by fuel raises steam in steam generator	650	600	Can be stored under water for tens of years, but storage could be longer in dry atmosphere	42	Same operational and safety advantages as Magnox but with higher operating temperatures and pressures, leading to reduced capital costs and higher steam cycle efficiency
PWR	Uranium dioxide enriched to 3.2% U^{235} Zirconium alloy cladding	Light water	Pressurized light water pumped to steam generator, which raises steam in a separate circuit	317	2235	Can be stored for long periods under water giving flexibility in waste management	32	Low construction costs resulting from design being amenable to fabrication in factory-built sub-assemblies. Wealth of operating experience now accumulated worldwide. Offload refuelling necessary

(*Continued*)

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Table 16.3 (*Continued*)

Type	Fuel	Moderator	Coolant			Spent fuel processing	Steam cycle efficiency (%)	Main economic and safety characteristics
BWR	Uranium dioxide enriched to 2.4% U^{235} Zirconium alloy cladding	Light water	Pressurized light water boiling in the pressure vessel produces steam, which directly drives a turbine	286	1050	As for PWR	32	Similar construction cost advantages to PWR enhanced by design not requiring a heat exchanger, but offset by need for some shielding of steam circuit and turbine. Offload refuelling necessary
CANDU	Unenriched uranium dioxide (0.7% U^{235}) Zirconium alloy cladding	Heavy water	Heavy water pumped at pressure over the fuel raises steam via a steam generator in a separate circuit	305	1285	As for PWR	30	Good operational record but requires infrastructure to provide significant quantities of heavy water at reasonable costs
RBMK	Uranium dioxide enriched to 1.8% U^{235}	Graphite	Light water boiled at pressure, steam used to drive a turbine directly	284	1000	Information not available	31	Information not available but operated in considerable numbers in the former USSR. Believed in the West to be inherently less safe

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As per the data released by the European nuclear society (Table 16.4), mainly PWRs are used in the nuclear power plants worldwide – 62 per cent according to the number, 68 percent according to the output – followed by BWR – 19 per cent according to the number, 21 per cent according to the output.

Table 16.4 *Nuclear Power Plants, Worldwide, Reactor Types, 18 January 2013*

Operational		
Reactor type	**Number**	**Electr. net output (MW)**
PWR	273	251,418
BWR	84	77,737
CANDU/D2O-PWR	48	24,201
GGR/AGR	15	8055
RBMK	15	10,219
FBR	2	580
Total	437	372,210

16.7 REACTOR CONTROL

Reactor power control is accomplished by using control rods and burnable poisons that contain neutron-absorbing materials. In PWRs, boric acid, a soluble chemical, in the primary coolant, also provides power control. By varying the concentration of boric acid is varied to control reactivity changes caused by depletion of fuel and build-up of fission products.

1. *Boron carbide*

Boron carbide is the most extensively used control material. It is used in thermal and fast reactors. The absorption of neutrons by ^{10}B results in the primary formation of ^{7}Li, helium and tritium as given by the following reactions:

$$B + n \rightarrow He + Li + 2.3 \text{ MeV}$$
$$B + n \rightarrow T + He + He$$

The fast neutron capture cross section of the ^{10}B isotope is greater than that of any other known isotope.

2. *Silver-base alloys*

An alloy of silver combined with 15 wt per cent cadmium and 5 wt per cent indium is used for control rod with suitable neutron-absorption properties over a wide range of neutron energies present in pressurized light-water reactors. This alloy clad in stainless steel or Inconel has been used as control rod material in PWRs.

3. *Hafnium*

Hafnium can be used to control the LWRs without excessive reactivity loss or damage over extended irradiation (approximately 40 year) for the lifetimes of the plants.

4. *Europium hexaboride*

There has been increasing interest in the use of europium hexaboride as an alternate control material to B_4C in fast breeder reactors.

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5. *Europium oxide*

Europium oxide is another material being considered as a neutron absorber material for use in the control rods. Many fast breeder reactors in the United States, Britain, Germany and Russia have tested Europium oxide. The BOR-60 fast breeder reactor (Russia) has operated satisfactorily since 1972 with europium oxide in one of the control rods.

6. *Burnable poisons*

Neutron-absorbing materials are also used in reactors to prevent power peaking in the early stages of operation of the core. They allow optimum burn-up of the fuel and power shaping in the core. PWRs use boric acid solution in the primary coolant as burnable poison to provide power control. Ceramic pellets containing burnable poisons are included in the fuel rods in most power reactors. Examples of these materials are boron carbide dispersions in alumina, gadolinium oxide dispersed in the uranium dioxide fuel and borosilicate glass.

16.8 RADIATION HAZARDS

Effects of nuclear radiation

Biological damage by interaction of radiation and tissue:

(i) Ionization – λ, β, γ – radiations ionize tissues into which they penetrate, resulting in complete damage of tissue.

(ii) Displacement – atom in the tissue is displaced by neutron and γ-radiation.

(iii) Absorption – Absorption of neutron by a tissue nucleus leads to radioactive nucleus, which results in change in chemical nature, malfunctioning of cell. Due to this, cell gets damaged leading to genetic modification.

Inhalation of radioactive material through air, food and water results in radiation hazard.

16.8.1 Handling of Nuclear Waste and Safety Measures

Wastes associated with nuclear power are as follows:

1. *Gaseous effluents*

Under normal operation, these are released slowly from the power plants into the biosphere and become diluted and dispersed harmlessly.

2. *Uranium mine and mill tailings*

Tailings are residues from uranium mining and milling operations. They contain low concentration of naturally occurring radioactive materials. They are generated in large volumes and are stored at the mines or mill sites.

3. *Low-level wastes*

Low-level waste (LLW) includes items that have become contaminated with radioactive material or have become radioactive through exposure to neutron radiation. This waste typically consists of contaminated protective shoe covers and clothing, wiping rags, mops, filters, reactor water treatment residues, equipments and tools, luminous dials, medical tubes, swabs, injection needles, syringes, and laboratory animal carcasses and tissues, etc. These contain less than 10 nC_i per gram of trans-uranium contaminants containing low but potentially hazardous concentrations

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of radioactive materials. These are generated in almost all activities (power generation, medical, industrial, etc) involving radioactive materials, require little or no shielding. These are disposed off in liquid form by shallow land burial.

4. *High-level wastes*

High-level wastes (HLW) are generated in reprocessing of spent fuel. They contain all fission products and contain most of the trans-uranium elements not separated during reprocessing. Such wastes are to be disposed off carefully. Since the only way radioactive waste finally becomes harmless is through decay, which for HLW can take hundreds of thousands of years, the wastes must be stored and finally disposed off in such a way that provides adequate protection to the public.

5. *Spent fuel*

This is unprocessed spent fuel that is removed from the reactor core after reaching its end-of-life core service. It is removed and then stored for three to four months under water in the plant site to give time for the most intense radioactive isotopes to decay before shipment for reprocessing or disposal.

Note: Units of nuclear radiation – Roentgen – amount of radiation that will on passing through pure air under standard condition produce 1 electrostatic unit of ions/cm^3 of air $\Rightarrow$ 86.9 ergs of energy absorbed/g of air, Rem – dose of absorbed radiation that will have the same effect that exposure to 1 Roentgen of γ-radiation will have $\Rightarrow$ 100 ergs/g of tissue, total dose driving a person's life time < 200 Rem weekly dose < 0.3 Rem.

Figure 16.33 shows a typical fuel cycle generating these wastes.

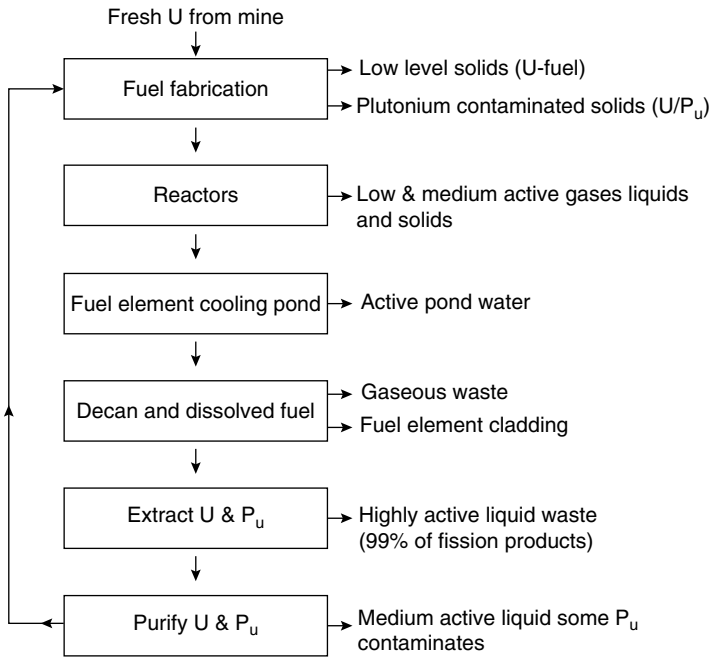


Fig. 16.33 *A Typical Fuel Cycle*

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16.8.2 Radioactive Waste Disposal

One of the major problems in the nuclear plants is the disposal of waste products that are highly radioactive. They emit large quantities of γ rays and these high-energy γ rays destroy all living matter through which they pass.

The radioactive products of 400 MW power station would be equivalent to 100 tons of radium daily and the radioactive effect of these plant products if exposed to atmosphere would kill all the living organisms within an area about 100 square miles.

The disposal of nuclear waste is a very difficult problem for the engineers and scientists.

In a nuclear fuel cycle, the solid, liquid and gaseous radioactive wastes are produced at different stages. These radioactive wastes must be disposed off in such a manner that there is no hazard to human and plant life. Moderate active solid wastes are buried in the ground. Moderate liquid wastes after preliminary treatments are discharged in deep pits or dry wells to keep them from seeping out into the surrounding ground. Active liquids are kept in concrete tanks and these tanks are buried in the ground till the radioactivity decays. Many times the radioactivity increases the temperature of the liquid waste or sometimes these liquids boil and the activity decreases with time. Gaseous wastes are discharged to atmosphere through high stacks if the wind permits.

The waste is disposed to air, ground and ocean.

1. *Air*

There are a lot of issues in the disposal of radioactive gases into the air. Because strong radioactive gases such as strontium and iodine are absorbed by the plants and they enter into the human body through food. Caesium is absorbed in muscle and strontium is absorbed in bones resulting in paralyses of the body. Generally, radioactive gases are collected and stored in a tank buried in the ground and disposed off to the atmosphere when radioactivity level is sufficiently low.

The amount of radioactivity presently disposed off in the air is well below the harmful level, but the problem will become serious when a large number of power reactors come up in operation.

2. *Ground*

This is one of the easy and cheapest methods of disposal because soil absorbs radioactive material easily. This disposal is suitable mostly in areas of low rainfall that are high above the ground water level.

Most of the radioactivities of waste are removed just by storage. The storage problem is simplified by separating caesium and strontium, which are extremely radioactive. These are generally stored in tanks that are buried in ground and then disposed off into the sea after 13 years of storage.

Defunct coal mines are used for waste disposal. The wastes are disposed off in salt heaps provided in the mines, because salt is a powerful absorber of radioactive emissions. Disposing off liquid waste by freezing is an easy and more economical method.

Disposal of LLW: The final deposition of the wastes is a major concern. Many countries are undertaking activities involving underground disposal in deep geological formations. These countries are investigating suitable sites and suitable methods of storage at these sites. These methods must be efficient enough to protect present and future populations from potential hazards. The suitable sites must be free of flowing ground water, but storage vessels must demonstrate reliability even in flowing water conditions.

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The disposal of low and intermediate level wastes is done at relatively shallow depths in many countries, namely packing the waste in solid form in concrete steel drums and burying (Figure 16.34).

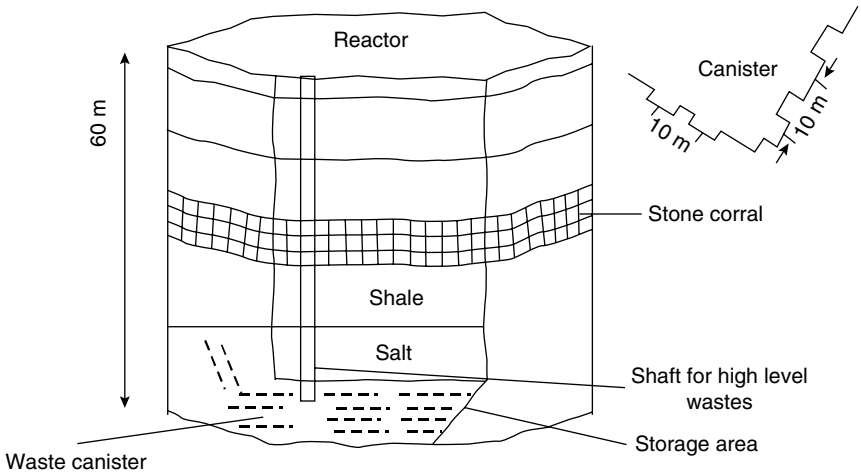


Fig. 16.34 *Disposal of LLW*

If spent fuel is to be disposed, it is buried in the earth, namely deep salt or granite formations below ground or below the sea bed.

Figure 16.35 shows a conceptual depository for the storage of HLW in rock salt formation for thousands of years. The solidified waste is placed in canisters that are stored in holes drilled in rock salt with a spacing of 10 m to allow efficient dissipation of energy without exceeding the temperature limits of either canister or salt. Each canister requires 100 m^2 of salt for cooling.

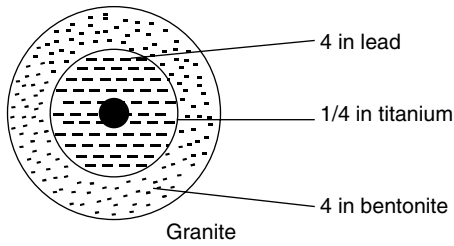


Fig. 16.35 *Canister for Vitrified High-Level Waste*

3. *Ocean*

In many places, the liquid waste is disposed off to the sea through the pipes carried from the plant to the point of disposal. While disposing off the waste into the sea, it should be ensured that the radioactivity level should not be very harmful to the fish life and seaweed, which is harvested for human use. In another method, the solid wastes should be cased into concrete

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blocks and these should be dropped into the sea at suitable places. The cost of disposal by this method is approximately ₹300 per cubic metre volume.

The danger of disposal into the sea water is indirect and depends on the absorption of radioactive elements such as caesium and strontium by plankton and then through the biological chain to the edible fishes and then to humans.

It is necessary to keep the radioactive solid waste at a depth of 6 m in water nearly for 100 days. It was found that after 100 days cooling of radioactive waste of 28 MW plant in water still had a radioactivity equal to million grams of radium. About 50 per cent of the radioactive elements disappear during cooling.

16.8.2.1 Different Methods for Nuclear Waste Disposal

In this section, different types of nuclear wastes and the methods used to handle them are discussed. These are as follows:

1. Fission fragments → weak, intensity active; isotopes → spent fuel stored under 6 m deep water to cool (100 days) → intensely active
2. Radioactive waste – gaseous liquid, solid
   Solid → buried at depths of few metres
   Gaseous → discharged to atmosphere through high stacks
   Liquid → after preliminary treatment discharged in dry wells or deep pits
3. Waste solution → stored in a stainless steel tank enclosed in a concrete wall, buried under earth (10 m) and provided with cooling oil to keep temperature at 50°C
4. Low-level solid waste → cast in cement in steel drum → drums buried below soil or kept at the bed of ocean
5. Medium-level solid waste → incorporated into cement cylinders
6. High-level liquid waste → stored in special steel tanks in concrete walls, water cooled → taken to storage area and disposed within 10 cm thick lead wall surrounded by 6 mm titanium.

Some other locations and methods used for nuclear waste disposal are as follows:

(i) *Geological formations*

(a) Rock salt

- ➤ Powerful absorber of radioactive emission
- ➤ Good thermal conductivity
- ➤ Cavities and tunnels can be easily made

(b) Argillaceous sediments – boreholes are provided at 160–260 m depths in a 100-m thick bed of clay

(c) Hard rocks – igneous, metamorphic, sedimentary rocks

(ii) *Ocean*

Floors of deep ocean provide safe, potential disposal sites for solidified high-level radioactive waste

(iii) *New methods to treat HLW*

(a) HARVEST Process – (highly active residues vitrification engineering study). This process was developed in 1970. In this process, highly active liquid waste is mixed with

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glass forming chemical in a steel container. The container is heated in a furnace to about 1000°C. It involves the following process. This mixture fuses to glass, which is further cooled, sealed and then discharged to the storage area (Figure 16.36).

Liquid waste + glass-forming material → heated steel vessel

↓

Mixture fuses to glass

↓

Cooled, sealed, discharged to storage

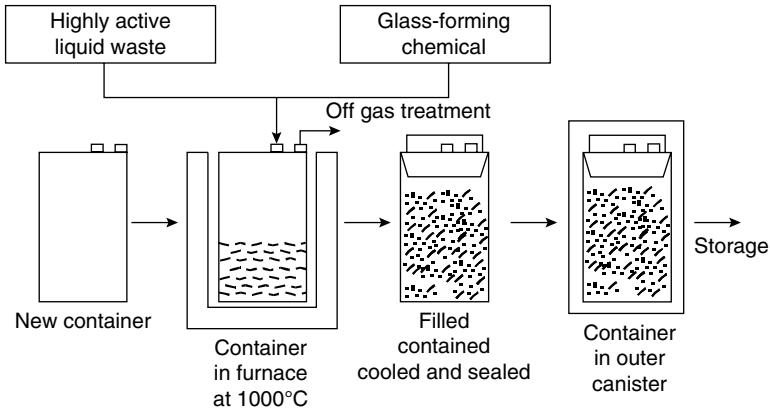


Fig. 16.36 *Waste Treatment (HARVEST Process)*

(b) AVM (*Atelier de Vitrification Marcoule*) Process

Figure 16.37 shows AVM (*Altier de –Vitrification Marcoule*) process used for liquid waste disposal. The process is self-explanatory. In this process, the liquid waste fed to a container is heated in a furnace and further mixed with glass powder. The mixture that comes out is melted in a melting furnace and collected in a vessel. In the subsequent steps, the container is closed and the exterior body of the vessel is decontaminated before disposing off into the storage area.

(iv) *Nuclear waste calcining*

Liquid → solid means small volume and easy storage. Calcining the liquid waste results in free flowing non-corrosion granules.

Features

- ➤ Spent fuel is reduced to a liquid
- ➤ Extraction of usable fuel from liquid
- ➤ Calcining the liquid waste

Off-gas clean-up system

Figure 16.38 shows an off-gas clean up system. Hot gas is quenched and passed through scrubbers and dried in condensers, demisters. It is then reheated and passed through silica gel

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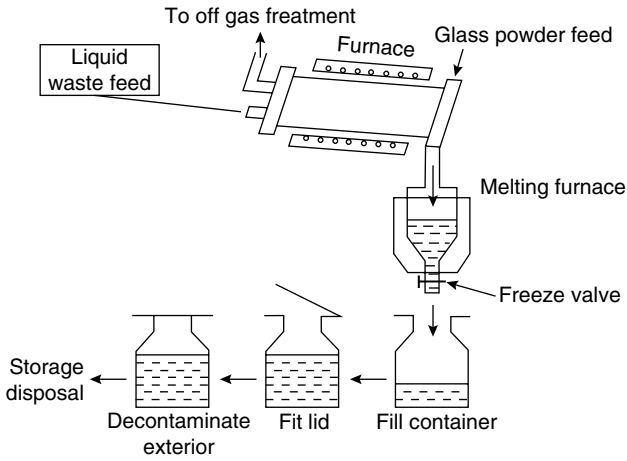


Fig. 16.37 *AVM Process*

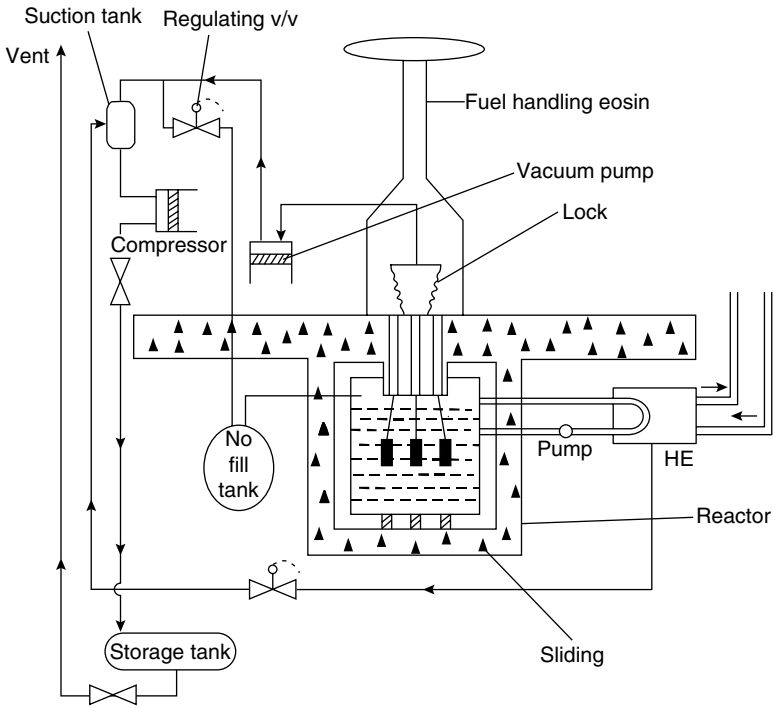


Fig. 16.38 *Off-Gas Clean-Up System*

absorber, demisters, filters, atmosphere protection system and then discharged to atmosphere through a stack 80 m high.

Gas disposal system – krypton, iodine gases, tritium, CO_2

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Crypton – cryogenic treatment of dissolved off-gas stream, packed in gas cylinder under pressure
Iodine – caustic scrubbing or HNO_3 scrubbing
Tritium – voloxidetion, pyrochemical processing, isotope enrichment
CO_2 – Caustic scrubbing
Rupture in the fuel element cladding releases gaseous fission products.

16.9 NUCLEAR POWER GENERATION IN INDIA

As per the information available in the Nuclear Power Corporation of India (NPCIL) (http://www.npcil.nic.in), nuclear power generation (starting from the year 2006–2007 till June 2013) from the operational plants in million units are given in Table 16.5.

Table 16.5 *Nuclear Power Generation (2006–2007 to 2013–2014)*

Year	Gross generation (MUS)	Capacity factor(%)	Availability factor (%)
2013–2014 (upto June – 2013)	7660	75	81
2012–2013	32,863	80	90
2011–2012	32,455	79	91
2010–2011	26,472	71	89
2009–2010	18,803	61	92
2008–2009	14,927	50	82
2007–2008	16,930	54	83
2006–2007	18,634	63	85

Most of the power plants commissioned in India are of pressurized heavy water reactor type except unit-1 and unit-2 of Tarapur Atomic Power Station (TAPS), Maharashtra. These plants use natural uranium fuel and heavy water as coolant and moderator. At the ground level, India's nuclear power programme has 20 reactors with an installed capacity of 4800 MW. Kudankulam 1 will add 1000 MW.

But the NPCIL has ambitious plans. Four indigenous Pressurized Heavy Water Reactors (PHWRs) of 700 MW totalling 2800 MW are being constructed by the NPCIL in other parts of the country; they are expected to be commissioned by 2016–2017.

A further 10 of the same size are planned, which means domestic reactors of nearly 10,000 MW capacity are in the pipeline. Some of them will be in new locations such as Gorakhpur in Haryana, Mahi Banswara in Rajasthan and Chutka and Bhimpur in Madhya Pradesh. All these efforts demand a huge requirement of natural uranium.

Table 16.6 shows the Total Nuclear Power Plant Capacity of India to 4780 MW

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Table 16.6 *Nuclear Power Plants of India (as on January 2011)*

State	Plant	Unit	Type	Capacity (MWe)	Date of commercial operation
Maharashtra	Tarapur Atomic Power Station (TAPS)	1	BWR	160	28 October 1969
	Tarapur Atomic Power Station (TAPS)	2	BWR	160	28 October 1969
	Tarapur Atomic Power Station (TAPS)	3	PHWR	540	18 August 2006
	Tarapur Atomic Power Station (TAPS)	4	PHWR	540	12 September 2005
Rajasthan	Rajasthan Atomic Power Station (RAPS)	1	PHWR	100	16 December 1973
	Rajasthan Atomic Power Station (RAPS)	2	PHWR	200	1 April 1981
	Rajasthan Atomic Power Station (RAPS)	3	PHWR	220	1 June 2000
	Rajasthan Atomic Power Station (RAPS)	4	PHWR	220	23 December 2000
	Rajasthan Atomic Power Station (RAPS)	5	PHWR	220	4 February 2010
	Rajasthan Atomic Power Station (RAPS)	6	PHWR	220	31 March 2010
Tamil Nadu	Madras Atomic Power Station (MAPS)	1	PHWR	220	27 January 1984
	Madras Atomic Power Station (MAPS)	2	PHWR	220	21 March 1986
Karnataka	Kaiga Generating Station (KGS)	1	PHWR	220	16 November 2000
	Kaiga Generating Station (KGS)	2	PHWR	220	16 March 2000
	Kaiga Generating Station (KGS)	3	PHWR	220	6 May 2007
	Kaiga Generating Station (KGS)	4	PHWR	220	20 January 2011
Uttar Pradesh	Narora Atomic Power Station (NAPS)	1	PHWR	220	1 January 1991
	Narora Atomic Power Station (NAPS)	2	PHWR	220	1 July 1992
Gujarat	Kakrapar Atomic Power Station (KAPS)	1	PHWR	220	6 May 1993
	Kakrapar Atomic Power Station (KAPS)	2	PHWR	220	1 September 1995

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16.10 QUESTIONS

16.10.1 Objective Questions

1. The efficiency of a nuclear power plant in comparison to a conventional thermal power plant is
   (a) same
   (b) more
   (c) less
   (d) maybe less or more depending on size
2. Isotopes of same elements have
   (a) same atomic number and different masses
   (b) same chemical properties but different atomic numbers
   (c) different masses and different atomic numbers
   (d) different chemical properties and same atomic numbers
3. The mass number of a substance represents the sum of total number of
   (a) protons and neutrons in an atom
   (b) protons and electrons in an atom
   (c) neutrons and electrons in an atom
   (d) protons and neutrons in a nucleus
4. Amongst the following, the fissionable materials are
   (a) U233 and Pu239
   (b) U231 and Pu233
   (c) U235 and Pu235
   (d) U238 and Pu239
5. A nuclear unit becoming critical means
   (a) it is generating power to rated capacity
   (b) it is capable of generating much more than rated capacity
   (c) there is danger of nuclear spread
   (d) chain reaction that causes automatic splitting of the fuel nuclei has been established
6. Moderator in nuclear plants is used to
   (a) reduce temperature
   (b) extract heat from nuclear reaction
   (c) control the reaction
   (d) cause collision with the fast moving neutrons to reduce their speed
7. The most commonly used moderator in nuclear plants is
   (a) heavy water
   (b) concrete and bricks
   (c) graphite and concrete
   (d) graphite

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8. Breeder reactor has a conversion ratio of
   (a) unity
   (b) more than unity
   (c) less than unity
   (d) zero
9. Fast breeder reactor uses
   (a) boiler
   (b) direct cycle of coolant system
   (c) double circuit system of coolant cycle
   (d) multi-pass system
10. One gram of uranium will produce energy equivalent to approximately
   (a) 1 tonne of high-grade coal
   (b) 4.5 tonnes of high-grade coal
   (c) 10 tonnes of high-grade coal
   (d) 100 tonnes of high-grade coal
11. Which of the following nuclear reactor does not need a heat exchanger for generation of steam?
   (a) gas cooled
   (b) liquid metal cooled
   (c) pressurized water
   (d) boiling water
12. The commonly used material for shielding is
   (a) lead or concrete
   (b) lead and tin
   (c) graphite or cadmium
   (d) thick galvanized sheets
13. The main interest of shielding in nuclear reactor is protection against
   (a) X-rays
   (b) infrared rays
   (c) a, P and y rays
   (d) neutrons and gamma rays
14. Reflector in nuclear plants is used to
   (a) return the neutrons back into the core
   (b) shield the radioactivity completely
   (c) check pollution
   (d) conserve energy
15. The energy required to be applied to a radioactive nucleus for the emission of a neutron is
   (a) 1 MeV
   (b) 2.4 MeV
   (c) 4.3 MeV
   (d) 7.8 MeV

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16. Ferrite material is
    (a) the most fissionable material
    (b) the basic fuel for nuclear paints
    (c) the basic raw material for nuclear plants
    (d) the material that absorbs neutrons and undergoes spontaneous changes leading to the formation of fissionable material
17. Enriched uranium is one in which
    (a) percentage of U^{235} has been artificially increased
    (b) percentage of U has been artificially increased
    (c) percentage of U^{234} has been artificially increased
    (d) extra energy is pumped from outside
18. In fast breeder reactors
    (a) any type of moderator can be used
    (b) graphite is used as the moderator
    (c) heavy water is used as the moderator
    (d) moderator is dispensed with
19. In nuclear fission each neutron that causes fission releases
    (a) no new neutron
    (b) at least one new neutron
    (c) one new neutron
    (d) more than one new neutron
20. The breeding gain in case of thermal breeder reactor as compared to fast breeder reactor is
    (a) same
    (b) lower
    (c) higher
    (d) higher/lower depending on the size of reactor
21. Gas-cooled reactor uses the following materials as moderator and coolant
    (a) graphite, CO_2
    (b) graphite, air
    (c) heavy water, CO_2
    (d) lead, H_2
22. A nuclear fission produces energy of following order in MeV
    (a) 20
    (b) 200
    (c) 2000
    (d) 20,000
23. The process by which a heavy nucleus is split into two light nuclei is known as
    (a) splitting
    (b) fission
    (c) fusion
    (d) chain reaction

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24. The first unclear power plant in India is located at
    (a) Kota
    (b) Kalapakkam
    (c) Tarapur
    (d) Baraeilly
25. Boiling water reactor uses the following as moderator, coolant and working fluid
    (a) ordinary fluid
    (b) heavy water
    (c) molten lead
    (d) hydrogen gas
26. U_{235} will undergo fission by
    (a) High energy (fast) neutrons alone
    (b) Low energy (slow) neutrons alone
    (c) Either fast or slow neutrons
    (d) Medium energy neutrons
27. U238 will undergo fission by
    (a) High energy (fast) neutrons alone
    (b) Low energy (slow) neutrons alone
    (c) Either fast or slow neutrons
    (d) Medium energy neutrons
28. A reactor capable of converting a ferrite material into fissile isotopes is called
    (a) Regenerative reactor
    (b) Fast breeder reactor
    (c) Breeder reactor
    (d) Boiling water reactor
29. Hydrogen is preferred as better coolant in comparison to CO_2 because former
    (a) Is lighter
    (b) Is inert
    (c) Has high specific heat
    (d) Is a good conductor
30. Natural uranium is made up of
    (a) 99.282% U_{238}, 0.712% U_{235}, 0.006% U_{234}
    (b) 99.282% U_{235}, 0.712% U_{238}, 0.06% U_{234}
    (c) 99.282% U_{234}, 0.712% U_{238}, 0.006% U_{235}
    (d) 99.282% U_{235}, 0.712% U_{234}, 0.006% U_{238}
31. Pick up the wrong statement Fast breeder reactors
    (a) Operate at extremely high power densities.
    (b) Are liquid-metal cooled
    (c) Produce more fuel than they consume
    (d) Use water as coolant.

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32. A pressurized water reactor employs pressurizer for the following application
    (a) To maintain constant pressure in primary circuit under varying load
    (b) To Supply high pressure steam
    (c) To increase pressure of water in primary circuit
    (d) To provide subcooled water at high pressure
33. The function of control rods in nuclear plants is to
    (a) Control temperature
    (b) Control radioactive pollution
    (c) Control absorption of neutron
    (d) Control fuel consumption
34. Breeder reactors employ liquid metal coolant because it
    (a) Acts as good moderator
    (b) Produces maximum steam
    (c) Transfers heat from core at a fast rate
    (d) Breeds neutrons
35. In triggering fission, which type of neutrons are more effective
    (a) Fast
    (b) Slow
    (c) In bulk
    (d) Static
36. A fission chain reaction in uranium can be developed by
    (a) Slowing down fast neutrons so that Uz fission continues by slow motion neutrons
    (b) Accelerating fast neutrons
    (c) Absorbing all neutrons
    (d) Using moderator
37. In triggering fission, the following types of neutrons are desirable
    (a) Fast moving
    (b) Slow moving
    (c) Critical neutrons
    (d) Neutrons at rest
38. Each fission of U235 produces following number of fast neutrons per fission
    (a) 1 neutron
    (b) 1 neutrons
    (c) 1 2 neutrons
    (d) 2 neutrons
39. A boiling water reactor uses following as fuel
    (a) Enriched uranium
    (b) Plutonium
    (c) Thorium
    (d) U

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40. A fast breeder reactor
    (a) Uses graphite rods as moderator
    (b) Has powerful moderator
    (c) Has no moderator
    (d) Uses ferrite material as moderator
41. A fast breeder reactor uses
    (a) 90% U-235
    (b) U-238
    (c) U-235
    (d) Pu-239
42. The coolant used in boiling water reactor is
    (a) CO_2
    (b) Pressurized water
    (c) Mixture of water and steam
    (d) Liquid metal
43. In boiling water reactor, moderator is
    (a) Coolant itself
    (b) Ferrite rod
    (c) Graphite rod
    (d) Liquid sodium metal
44. The efficiency of a nuclear power plant in comparison to conventional and nuclear consideration is
    (a) Higher cost of nuclear fuel
    (b) High initial cost
    (c) High heat rejection in condenser
    (d) Lower temperature and pressure conditions
45. The fuel needed, with reflector in nuclear power
    (a) More
    (b) Less
    (c) Same
    (d) Zero

Answers

1. c	2. b	3. d	4. a	5. d	6. d	7. d	8. b	9. c	10. b
11. d	12. a	13. d	14. a	15. d	16. d	17. a	18. d	19. d	20. b
21. a	22. b	23. b	24. c	25. a	26. c	27. a	28. a	29. c	30. a
31. d	32. a	33. c	34. c	35. b	36. a	37. b	38. d	39. a	40. c
41. a	42. c	43. a	44. d	45. b					

16.10.2 Review Questions

1. Give the classification of nuclear reactors.
2. Explain the factors that control the site selection of nuclear power plant.
3. Sketch and explain liquid metal cooled reactor and also list its merits.

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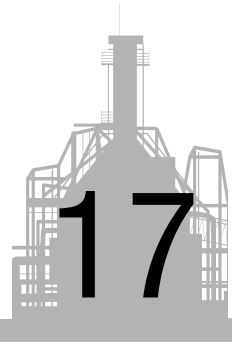
4. Write short notes on nuclear waste disposal.
5. Explain with a sketch the elements of a nuclear reactor.
6. Describe the working of sodium graphite reactor with a neat sketch.
7. Write short notes on fusion and fission reactions.
8. Make a neat sketch and explain the working of a gas cooled reactor.
9. Explain the phenomenon of chain reaction and the difficulties encountered with and the methods adopted for disposal of the different wastes in nuclear power plants.
10. What are the various factors to be considered while selecting the site for nuclear reactor?
11. What do you understand by terms binding energy and mass defect? How they are related to each other?
12. What are the different types of nuclear reactions taking place? Explain the significance of each in nuclear power generation.
13. What do you understand by nuclear fission? What are the essential requirements to cause nuclear fission?
14. What is chain reaction? How it is maintained?
15. What is the difference between controlled and uncontrolled chain reaction? Explain with neat sketches and with examples.
16. What different moderators are used in practice? What different properties make them suitable moderators?
17. What do you understand by fertile material and breeding. What is the importance of breeding in power engineering?
18. Draw a neat diagram of PWR and BWR and explain the advantages and disadvantages. What are the conditions which prefer PWR over BWR?
19. Draw a neat diagram of CANDU type reactor and give its advantages and disadvantages of this reactor over other types.
20. What are the outstanding features of advanced gas cooled reactors over other types?
21. What factors control the selection of a particular type of reactor?
22. What do you understand by breeding? What factors control the breeding?
23. Draw a neat diagram of breeder reactor and list out its advantages and disadvantages. Why only sodium is used as a coolant in breeder reactors?
24. Why the moderator is not required in breeder reactor?
25. List out the advantages and disadvantages of nuclear plants over conventional thermal plants.
26. What are the desirable properties of a good moderator?
27. Why shielding of a reactor is necessary?
28. What do you understand by thermal shielding?
29. What are the desirable properties of a shielding material?
30. Discuss the problems of health hazard from nuclear radiations.
31. Explain how radioactive wastes are disposed off.
32. Define multiplication factor and criticality.
33. Give the advantages and disadvantages of thermal and fast reactors.
34. Discuss the merits and demerits of PWR in comparison to a gas cooled reactor.

16.10.3 Reference

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Power Plant Economics



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17.3 Terminology used in power supply
17.4 Cost of electrical energy
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17.1 INTRODUCTION AND BASIC DEFINITIONS

The load required by the consumer does not remain constant with respect to time (hour, day or month), and it fluctuates according to his requirements. The problem of variable load is an important one because each kilowatt-hour energy is to be put on the transmission line at as low a production cost as possible. The cost of generation and transmission is dependent not only on the improved operating conditions, such as turbine and generators efficiencies, but also on the first cost of equipment, which can be reduced by using simplified control and by eliminating the various auxiliaries and regulating devices.

Figure 17.1 shows the general arrangement of the electrical power generation, transmission and distribution system. The energy is sent to the substations that are located at the ends of the primary distribution system. The energy from the substation is then carried through the feeders to the distribution transformers. Each transformer is connected to the systems of one or more customers by short low-voltage lines.

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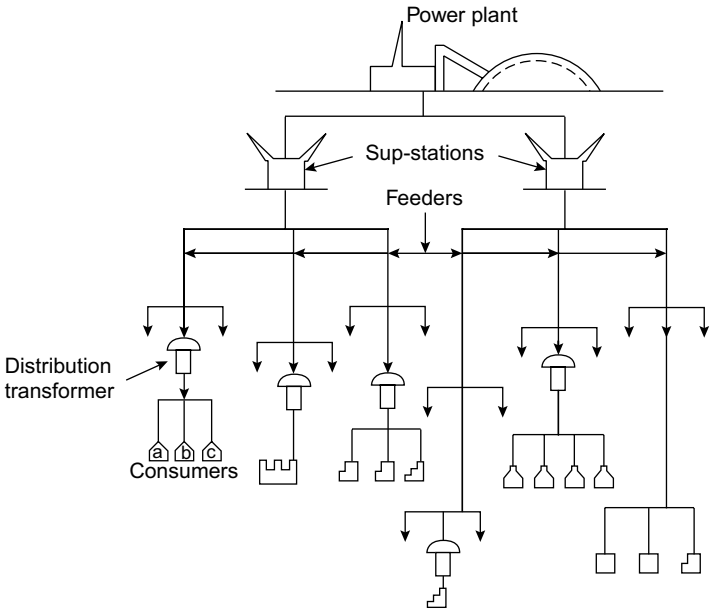


Fig. 17.1 *General Arrangement of the Electrical Power Generation*

17.2 TYPES OF LOADS

In a power station, load on the system refers to a device that taps the electric energy from an electric energy system. Load on the plant may be inductive (induction motor), resistive (electric bulb) or capacitive. Various types of loads on the system are as follows:

(a) *Domestic load:* This load consists of lights, fans, refrigerators, television, water pumping motors, etc. Domestic loads vary during different times of a day, and hence has low load factor ranging from 10 to 12 per cent.

(b) *Industrial load:* This load consists of all types of power requirements from industries and varies from industry to industry. Large-scale industries require 500 kW and above; medium-scale industries require 25–100 kW, whereas small-scale industries require around 25 kW.

(c) *Municipal load:* This load consists of load due to street lighting, water treatment, pumping and drainage purpose. Street lights are on during the night time, whereas water pumping is done generally during off-peak hours, to ensure improved load factor.

(d) *Commercial load:* This load consists of lighting for shops, fans and restaurants. This type of load happens more during day time as compared with domestic load. Due to seasonal usage of heaters and air conditioners, load on the system also varies.

(e) *Irrigation load:* This load is due to electric supply to irrigation pumps and motors and is generally supplied during night hours.

(f) *Traction load:* This load consists of railways, trolley buses, tram cars, etc., with wide variations during peak (morning and evening) and off-peak hours.

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17.2.1 Load Curves

Since a consumer of electric power always uses the power as and when required, the load will also be changing with time and will not be constant.

"A curve showing the load demand (variations) of consumer with respect to time is known as load curve." If the time is in hours, then the load curve is known as daily load curve. If the time is in days, the load curve is known as monthly load curve; and if the time is in months, the load curve is known as yearly or annual load curve. The load curve shows how the load varies with respect to time. Load curve is useful in predicting the annual requirements of energy and capacity of the power plant required to take the peak load.

Figure 17.2 shows a load curve of a consumer. The area under the load curve gives the total energy consumed by the customer.

The energy consumption of the customer is given by an expression if the load curve is drawn on an hourly basis.

$E = \int_0^{24} P$. dt. Where P is power/load in kilo-watt.

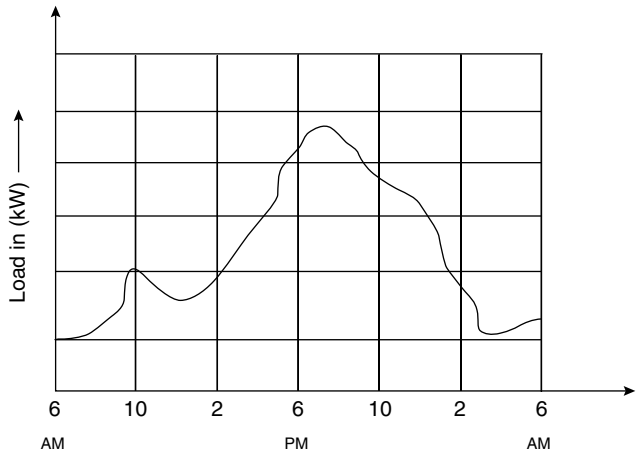


Fig. 17.2 *Chronological Daily Load Curve*

Figure 17.3 shows two customers A and B consuming the same amount of energy but with a different nature of consumption.

In the second case, the peak load is far greater than the first therefore the generating capacity of the plant required to supply the load of B is greater than the capacity required to supply the load of A. The plant designed for customer B is not only bigger in size, but it also runs under load (part load) conditions for majority of the period. Therefore, the cost of energy supplied to B may be hundred per cent greater than the cost of energy supplied to A even the total energy consumed by both customers is same.

Most complex problem of present-day operation of modern power plants arises from the variability of the load demanded by the consumers according to the requirements with respect to time. For economical and better operation of the power plant, constant magnitude load is always

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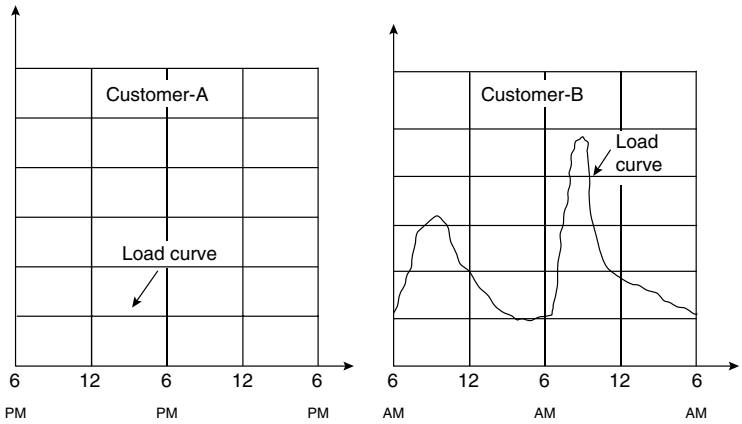


Fig. 17.3 *Constant and Variable Demand Load Curves*

desirable. Because it allows the plant to work at highest efficiency and requires simplified control and regulating devices.

In the next section, the load curves for different types of customers who are supplied from the power generating plant are shown. They include industrial, domestic, commercial, etc. The load curve of each customer is different from the other as per the activities of the customer.

(a) *Load curve for domestic customers*

The load curve shown in Figure 17.4 is typical of a residential community. During the early morning hours, the energy is required for lights, refrigerators, water heaters, etc. After the breakfast (at 9 AM), the demand decreases and fairly remains constant until about 4 PM. The cooking appliances then cause a slight rise in demand at 4 PM. After 4 PM, the early sunset of winter brings the lights into action and total load rapidly approaches its peak about 5 PM during the month of December. The high demand occurs at about 8 PM and then the load drops fairly rapidly.

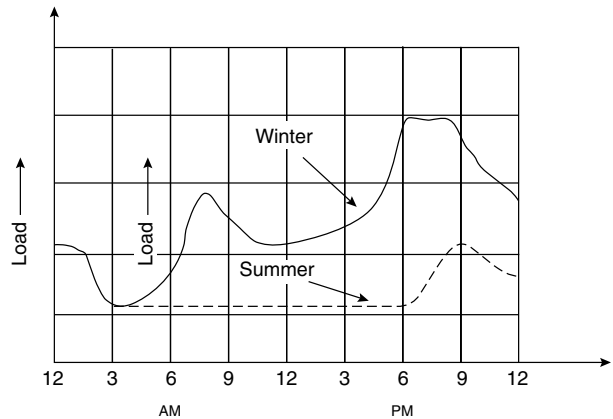


Fig. 17.4 *Residential Load Curve*

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(b) *Industrial load curve*

The load curve shown in Figure 17.5 is typical of a one shift in an industry. In the early morning hours, the demand is generally for lighting and auxiliary drives for heating boiler plants as well as some processes that require continuous energy supply such as refrigeration and electric furnaces. The energy demand increases from 5 to 8 AM as some of the factory machinery starts running for warming prior to operation. By 8 AM, the entire industry starts running and energy demand remains constant until shortly before noon. Load falls off as some of the machines are shut down during lunch hours. By 2 PM again, the load attains same level as at 8 AM Shortly before 4 PM, the load starts to drop as the shift of work ends. At 6 PM, most of the machines are shut down and load gradually tapers off until 10 AM when the minimum demand is reached and continues till the start of next working day.

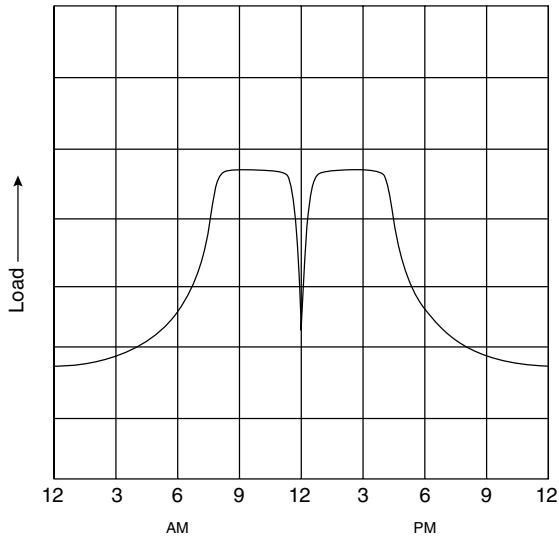


Fig. 17.5 *Industrial Load Curve for One Shift*

(c) *Urban traction load curve*

The load curve shown in Figure 17.6 is typical of a traction load. From the midnight, until 3 AM, the demand tapers off as the service reaches its minimum level and continues until about 5 AM. As the early factory workers start for their work, the required train services increase rapidly and the consequent load continuously rises as the factory workers are followed by office workers, school children and early shoppers. The peak load reaches at about 9.30 AM. The load after 10 AM rapidly diminishes as some of the trains return to the yards. The minimum day-load is reached during the noon hours and then rises continuously until the evening rush hours are in full swing with most of the workers go back to their homes. The load (after 6 PM) then falls rapidly. After midnight, the load again follows the cycle described.

(d) *Commercial load curve*

The load curve shown in Figure 17.7 is typical of commercial loads of shops and offices. The lighting in the shops and offices starts at 6 AM for cleaning and sweeping, and it reaches peak

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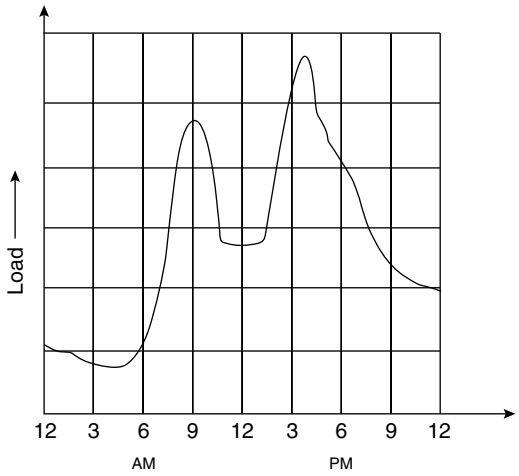


Fig. 17.6 *Urban Traction Load Curve*

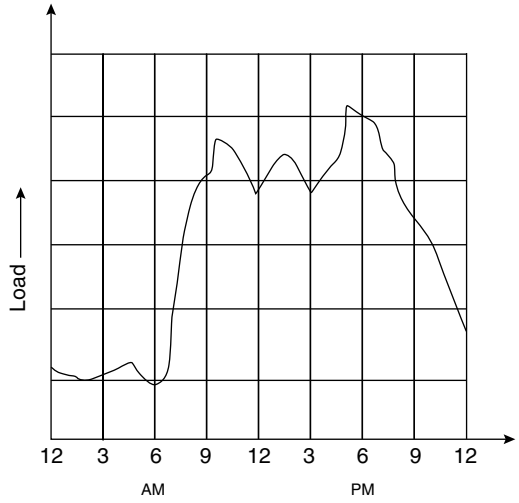


Fig. 17.7 *Commercial Loads of Shops Offices and Hospitals*

at 10 AM when the offices and shops start. The load remains constant more or less during 10 AM–4 PM. It further increases during 4–6 PM as more lights are required due to cloudy sky. Then, the load rapidly falls during 7 PM–12 night as the offices remain closed. The same load (night lights in shops and offices) remains during 12 night–6 AM.

(e) *Street lighting load*

The load curve shown in Figure 17.8 is typical of a street lighting load. Street lighting is the only form of load that does not exhibit peak demands. Normally, all lights are turned on and are turned off almost simultaneously. The total demand remains more or less constant during the hours of

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darkness. The turning on and turning off of lights is usually synchronized with the time of sunset and sunrise, respectively. Hence, the road lights will be functioning for a much shorter time during summer time than winter.

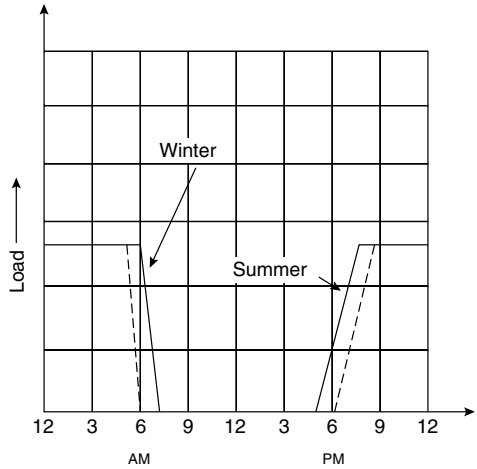


Fig. 17.8 *Street Lighting Load*

(f) *Water pumping load*

A typical water pumping load curve is shown in Figure 17.9. Generally, the true appearance of load curve will not be as smooth as drawn. The lighting load is smoothest due to the constant nature of the energy requirements. The traction load, spot welding and stamping curves exhibit violent fluctuations as high demand during starting and no demand during stopping. For loads

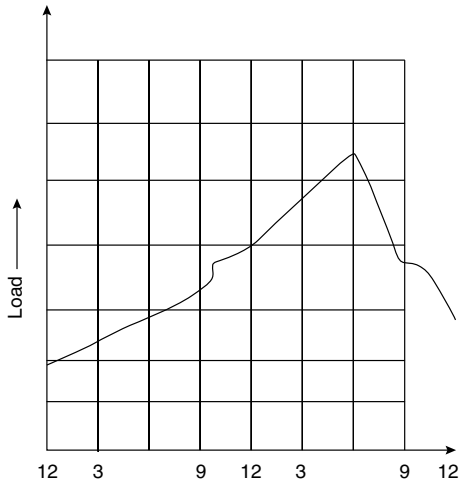


Fig. 17.9 *Water Pumping Load*

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of wide fluctuations, usual practice is to read the integrating metres every hour and plot the difference in readings as an equivalent constant load for corresponding periods on the curve. This gives the step-shaped curve. Such loads take wide and erratic swings and need special treatment of generator governing systems.

17.2.2 Drawing Load Duration Curve

The demand load curve of a power plant is generally found out by adding all the loads mentioned above in addition the effects of the factors mentioned below:

(a) Events such as political broadcast, election results, football game or cricket match

(b) Abrupt and unexpected changes in weather

(c) The changes in atmospheric temperature influence the electric loads

(d) Wind speed variations that effect on the requirement of electric power. More wind raises the cooling rate of buildings creating demand for more heating energy. In many cases, the effect of wind is more pronounced than the change in atmospheric temperature.

The effect of addition of different types of loads is to reduce the fluctuations of load required by the number of consumers and supplied by a single power plant.

17.2.2.1 Load Duration Curve

Load duration curve is a rearrangement of daily load curve with loads set up in descending order of magnitude. The areas under the load duration curve and corresponding load curve are equal and give kilowatt-hour of energy for that period.

To obtain the load duration curve from chronological load curve, the daily load curve is cut into many vertical strips. It is then arranged in descending order. The graphical method for constructing load duration curve from the load curve is shown in Figure 17.10.

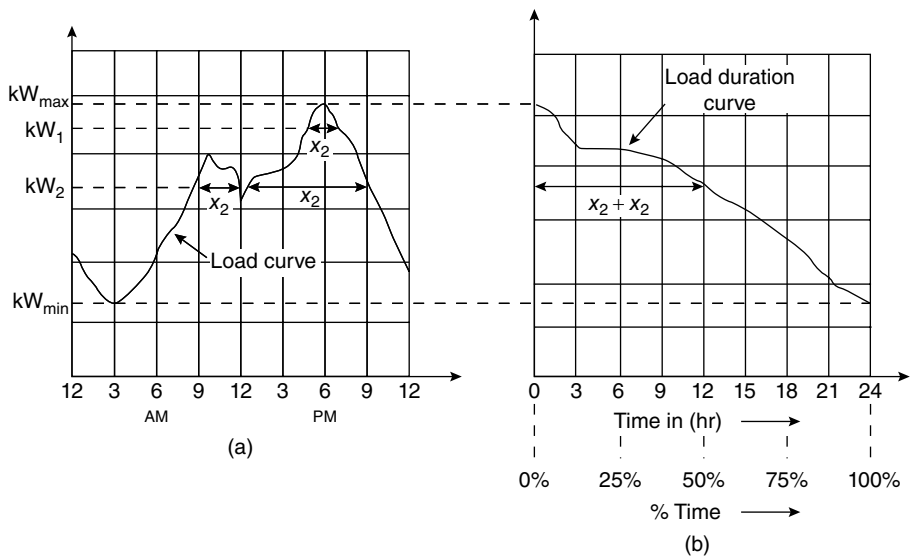


Fig. 17.10 *Preparing Load Duration Curve from Load Curve*

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(a) For plotting the load duration curve, mark the abscissa at any load ordinate equal to the length of the abscissa intercepted by that load ordinate on the load curve.

(b) The intercept is that point at the maximum demand (kW_{max}), and it is plotted at zero hours as shown in Figure 17.10(b).

(c) At load kW_1, the intercept is x_1 hours and is plotted at x_1 hours on the load duration curve.

(d) At load kW_2, the intercepts are $x_2 + x_3$ hours and is plotted accordingly as shown in Figure 17.10.

(e) At minimum load kW_{min}, the intercept covers the entire period of 24 hr and plotted accordingly.

Advantages of load duration curve the following:

(i) Any point of the load duration curve is a measure of the number of hours in a given period during which the given load has prevailed.

(ii) Load duration curve gives the advantage of summarizing loads for a day, week, month or year.

(iii) It is also useful for power plant design because in one simple curve, a whole year can be summarized showing peak demand, the variations in demand down to minimum, the length of the time they existed and total energy involved.

Figure 17.11 demonstrates a simplified version of drawing a load duration curve (Figure 17.11(b)) from a load curve (Figure 17.11(a)), as explained above.

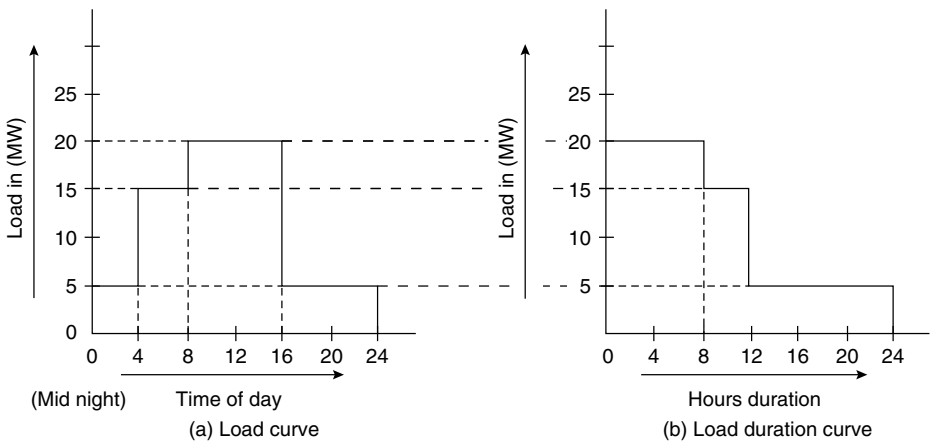


Fig. 17.11 *A Simple Demonstration of Drawing Load Duration Curve from the Load Curve: (a) Load Curve and (b) Load Duration Curve*

17.2.3 Effect of Variable Load on Power Plant Design and Operation

The necessity of supplying a variable load influences the characteristics and method of use of power plant equipments. The generation of power must be regulated according to the demand followed by governing to achieve the same. Another requirement of a power plant is a quick response to the load.

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In all variable load problems, the major problem is that the generator (and prime mover) must be able to take varying load as quick as possible without the change in voltage and frequency. When the load on the generator increases, the rotor and prime-mover speed decreases, thereby reducing the frequency. With the decrease in speed of the prime mover which is due to the increase in load on generator, the governor must act, admitting more fuel in case of thermal plants and more water in case of hydel plant, enough to bring the speed back to normal and pick up the load. The frequency stabilizers are used to maintain the frequency constant which may change due to response of the equipments.

The raw materials used in thermal power plant are fuel, air and water and to produce variable power according to requirement by varying the raw material. With an increase in load on the plant, the governor admits more steam and maintains the turbine speed. The governor response to this point has followed rapidly the change of load but beyond this point, changes are not so rapid. Because, the steam generator operates with unbalance between heat transfer and steam demand long enough to suffer a definite decrease of steam pressure. With fluctuating steam demand, it becomes very difficult to secure good combustion and steady steam pressure because efficient combustion requires the co-ordination of so many services. The co-ordination between the different components and processes is not as simple as supplying of more raw materials, but the reason being that there is certain time lag present in combustion and heat transfer that is not present in electrical generators.

The design of thermal plants for variable loads is always more difficult than diesel or hydraulic plants, and it is always desirable to allow the thermal plant to operate as base load plant.

It is always necessary in case of power plants to keep the reserve capacity in order to meet the peaking loads. This reserve is particularly required for an individual or isolated plant. This reserve capacity always increases the charges of electrical energy supplied. Therefore, it is always desirable to keep the reserve capacity as small as possible.

17.2.4 Methods to Meet Variable Load

Variable load demand can be met with an interconnected hydro-steam generation system, where peak loads are taken up by steam power plant. At peak loads, the generating capacity of water turbine may be kept at a higher level than the water flow capacity at peak loads, and lesser than or equal to maximum flow of river. This arrangement ensures that the hydro unit generates adequate energy at low cost when sufficient water flow is available. Referring to Figure 17.12(a), during high flow season, the hydro plant is operated as base load plant and the steam plant is operated as peak load plant. During low flow season, namely summer days, steam plant is operated as base load plant, whereas the hydro plant is run as peak load plant to cater the peak demand, as shown in Figure 17.12(b).

Interconnection of two plants offers the following advantages:

(a) Economic operation since combined system offers a much lower cost than a single, small capacity system

(b) Enhanced and reliable power supply

(c) Reduction in total installed capacity

(d) Low capital and maintenance costs

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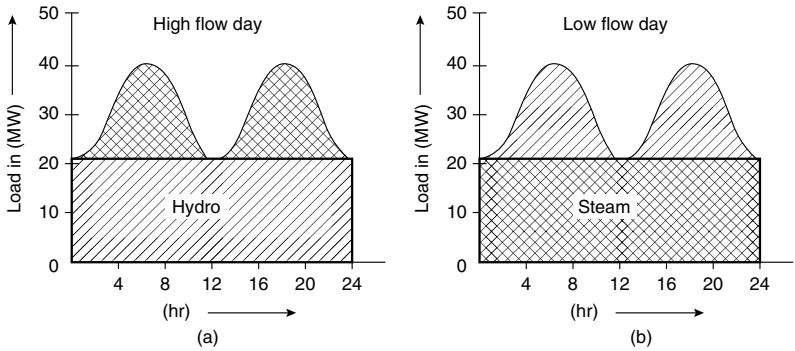


Fig. 17.12 *Methods to Meet Variable Load*

17.2.5 Prediction of Future Loads

The capacity of a power station is decided by knowing the maximum power demand of the area at which the power station is to be installed. The following two methods are used to forecast the load requirements:

(1) *Statistical method:* The future load is judged based on the data method for the past several years.

(2) *Field survey method:* The existing requirements of the area for different loads such as industrial, agricultural, municipal and residential are found. Future load requirement is decided taking into account the factors such as population growth, standard of living, climate of the region and industrial development.

17.3 TERMINOLOGY USED IN POWER SUPPLY

Some common terms used in connection with power supply are defined as follows:

(a) *Connected load:* The connected load is the sum of ratings in kilo-watt of the equipments installed in the consumer's premises. The connected loads in the premises of a consumer are shown in Figure 17.13.

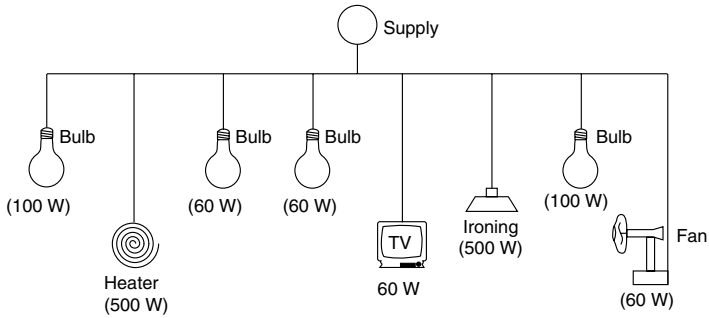


Fig. 17.13 *Load in a Consumer's Premises*

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The total connected load in the consumer's premises
= 100 + 500 + 60 + 60 + 60 + 500 + 100 + 60 = 1,440 watts.

(b) *Maximum demand:* The maximum demand is the maximum load that a consumer uses at any time. It is always less than connected load or equal to connected load. When all the equipments fitted in the consumer's house run to their fullest extent simultaneously, then the maximum demand becomes equal to connected load. Generally, maximum demand is always less than connected load. This is because all the equipments never run simultaneously and never run at full load. If ironing is done in day time and the bulbs are on in the evening with TV and fan is running, then the simultaneous maximum demand is

100 + 60 + 60 + 60 + 100 + 60 = 440 W.

Say the heater is used in evening time during winter and all bulbs are on as well as the TV is running, then the maximum demand is

100 + 500 + 60 + 60 + 60 + 100 + 60 = 940 W.

Thus, the maximum demand of a consumer depends on the time of day as well as his habits.

(c) *Demand factor:* It is defined as the ratio of maximum demand to connected load. The maximum value of the demand factor is unity. Each device will reach its own maximum demand at some time during its operation, but the demand factor measures the extent to which it contributes towards the maximum demand of a group of devices of which it is a part.

For the above-mentioned example, the demand factors for day time and evening time are given below:

Demand factor (F_d) = 0.43 (day time)
= 0.61 (evening time)

The demand factors for different types of loads are listed in Table 17.1.

Table 17.1 *The Demand Factors for Different Types of Loads*

S. no	Type of load	Capacity or type	Demand factor
1	Residence lighting	< 0.25 kW	1.00
–	–	< 0.50 kW	0.60
–	–	>1 kW	0.50
Restaurant	0.70	–	–
Office	0.70	–	–
Theatre	0.70	–	–
2	Commercial lighting	Small industry	0.60
School	0.55	–	–
Hotel	0.50	–	–
3	General power services	0–10 H.P.	0.75

(*Continued*)

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Table 17.1 (*Continued*)

S. no	Type of load	Capacity or type	Demand factor
–	–	10–20 H.P.	0.65
–	–	20–100 H.P.	0.55
–	–	> 100 H.P.	0.50

(d) *Average load:* The average load is the ratio of the area under the load curve (energy in kW hr) to the time period (24 hr) considered to draw the load curve.

$$\text{Average load} = \frac{\text{Area under load curve}}{24}$$

$$= \frac{\text{Energy consume in 24 house}}{24}$$

It can be noted that two load curves may represent the kilo-watt hour production, yet the unit cost of production for one may be more than the other. To describe an operating condition, some information other than the magnitude of kilo-watt hour energy produced is needed. The relationship between the peak load and average load over the time specified is required to describe an operating condition of the power plants. If the plant runs continuously at an average load, it generates the same amount of energy that the actual load curve shows.

Figure 17.14 shows the load curve drawn with load versus time period.

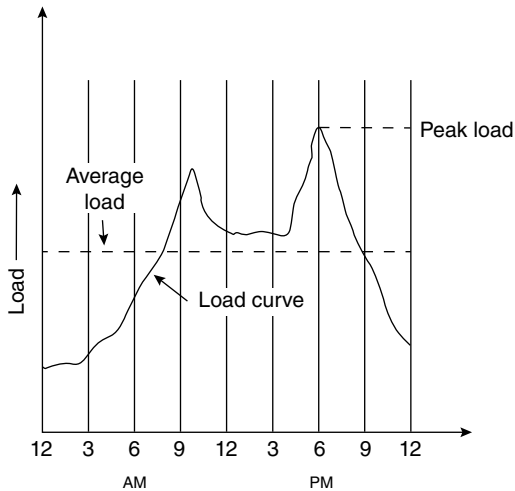


Fig. 17.14 *Load Curve*

(e) *Load factor:* It is the ratio of average load to the peak or maximum load determined by the consumer. The load factor is always less than unity. The power plants that are used to take the base load run on a high load factor and the plants that are used to take the peak load run on low load factors.

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$$\text{Load factor } (F_1) = \frac{\text{Average load}}{\text{Maximum load}}$$

The load factor can also be given by

$$\text{Load factor} = \frac{\text{Total energy consumption in 24 hours}}{\text{Peak load in 24 hours}}$$

The load factors for different types of consumers are listed in Table 17.2.

Table 17.2 *The Load Factors for Different Types of Consumers*

Nature of load	Load factor
Residential load	
Lights, heaters, TV and radio sets, air conditioners and refrigerators, electric cookers and water heaters and small pumping sets	0–15%
Commercial load	
Lighting in shops and those used for advertising and electrical equipments, in restaurants and markets	25–30%
Industrial load	
Small-scale industries (below 25 kW)	30–50%
Medium- and large-scale industries (100–500 kW)	60%
Heavy industries (> 500 kW)	80%
Municipal load	
Road lights and traffic signals	25%

Low load factor is an indication of high cost of electric production as the power plant is not used to its full capacity for the whole period (24 hr) considered. For the low production cost of the electric energy, it is always desirable to run the plant to its full capacity for the maximum period of operation to give high load factor. This is the reason why the base load plants of very high capacity run at high load factor (nearly unity) and peak load plants are allowed to run at low load factors to reduce the overall production cost.

(f) *Diversity factor:* The diversity factor is the ratio of the sum of the maximum demands of the individual consumers and the simultaneous maximum demand of the whole group during a particular time.

$$\text{Diversity factor} = \frac{\text{Sum of individual demands}}{\text{demand}}$$

As the total maximum demand required at any time during the day is less than the sum of the maximum demands due to diversity, the total load factor of the system increases and which is desirable for the economic operation of the power plant helps to supply more energy with less plant capacity and at a cheaper rate as the power factor of the system improves.

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The needs of the consumers are their maximum demands and their energy consumptions during a day. It is always necessary to provide the generating capacity equal to their maximum as possible. This is possible only if the load factor of all consumers when combined approaches to unity. This is not possible in practice owing to variable load characteristics when a number of consumers with different load requirements at different times during the day are to be supplied. An attempt should be made to supply these loads in such a way as to smooth the load curve of the system and obtain as high a load factor as possible and practicable.

The diversity factors for different types of consumers are given in Table 17.3.

Table 17.3 *The Diversity Factors for Different Types of Consumers*

Elements of system	Residence	Commercial	General	Large load	Load	Power users
Between consumers	2.00	1.46	1.44	–	–	–
Between transformers	1.30	1.30	1.35	–	–	1.05
Between feeders	1.15	1.15	1.15	–	–	1.05
Between substations	1.10	1.10	1.10	–	–	1.14
Consumer to transformer	2.00	1.46	1.44	–	–	–
Consumer to feeder	2.60	1.90	1.95	–	–	1.15
Consumer to substation	3.00	2.19	2.24	–	–	1.32
Consumer to generator	3.29	2.41	2.46	–	–	1.45

High diversity factor is always desirable for the economic operation of the plant because the load factor increases with an increase in diversity factor. The effective demand of a consumer on the generator is determined by multiplying his connected load by the demand factor and dividing the product by the diversity factor.

$$\text{Maximum effective demand} = \frac{\text{Connected load demand factor}}{\text{Diversity factor}}$$

The peak demand of a system is sum of the individual demands of the devices that are functioning at the time of the peak. At the time of the system peak demand, the demand of a particular group of similar consumers is seldom at the maximum value that it may reach at some other time of the year

Peak diversity factory,

$$\text{System peak demand} = \frac{\text{Maximum effective demand of consumar group}}{\text{Demand of consumer group at the time of system peak demand}}$$

$$\text{System peak demand} = \frac{\text{Maximum effective demand}}{\text{Peak diversity factor}}$$

When peak diversity factor is not given, it can be taken as unity.

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(1) *Plant capacity factor:* The load and diversity factors do not give any idea about the reserve capacity required in the station. to overcome this disadvantage the following two factors are used. *Capacity factor is defined as the ratio of actual energy produced to the maximum possible energy that could have been produced during the same period.*

$$\text{Capacity factor} = \frac{\text{Average load during time } t \text{ considered}}{\text{Plant capacity during time } t \text{ considered}}$$

$$= \frac{\text{Average load}}{\text{Plant capacity}} = \frac{\text{Peak load Load factor}}{\text{Plant capacity}}$$

Significance: The capacity factor shows how near the plant runs to its full rating. When there are several plants on a system, the most efficient (base load) plant runs close capacity factor equal to unity as a base load plant and least efficient acts as a peaking plant. The difference between load and capacity factors is an indication of reserve capacity. The high values of demand factor, load factor, diversity factor and capacity factor are always desirable for economic operation of the plant and to produce the energy at a cheaper rate.

(2) *Plant use factor: It is the ratio of energy produced in given time to maximum possible energy economic that could have been produced during the actual number of hours of operation*. It shows the extent to which the plant capacity is used to meet the peak demand.

$$\text{Plant use factor} = \frac{\text{Annual energy produced}}{\text{Capacity of plant} \;\; \text{no. of hours} \;\; \text{plant is in operation during year}}$$

Some important observations on plant use factor are as follows.

- As the plant use factor approaches 1, it indicates the need for additional capacity of the plant.
- The high value of plant is use factor indicates that the plant is used most efficiently. A low utilization factor means the plant is used only for standby purposes on a system comprised of several stations or the capacity has been installed well in advance of need.
- In some inter-connected system, the use factor may exceed unity (1.1–1.2). This indicates that loads have been carried in excess of the rated capacity of the equipment. Hence, the plant carries 10–20 per cent more load than its rated capacity as the equipments are always designed to take nearly 10–20 per cent more load than rated.

From the above equations, we can write

- Capacity factor = load factor × use factor

$$\text{Load factor} \times \text{use factor} = \frac{\text{Average load}}{\text{Peak load}} \times \frac{\text{Peak load}}{\text{Plant capacity}}$$

$$= \frac{\text{Average load}}{\text{Plant capacity}} = \text{Capacity factor}$$

The effect of age on the plant capacity factor and the effect of plant capacity factor on the production cost are shown in Figure 17.16. The annual plant capacity factor (particularly, thermal

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and nuclear) decreases with the age of the plant, because the efficiency of the components decreases with an increase in age, as shown in Figure 17.15.

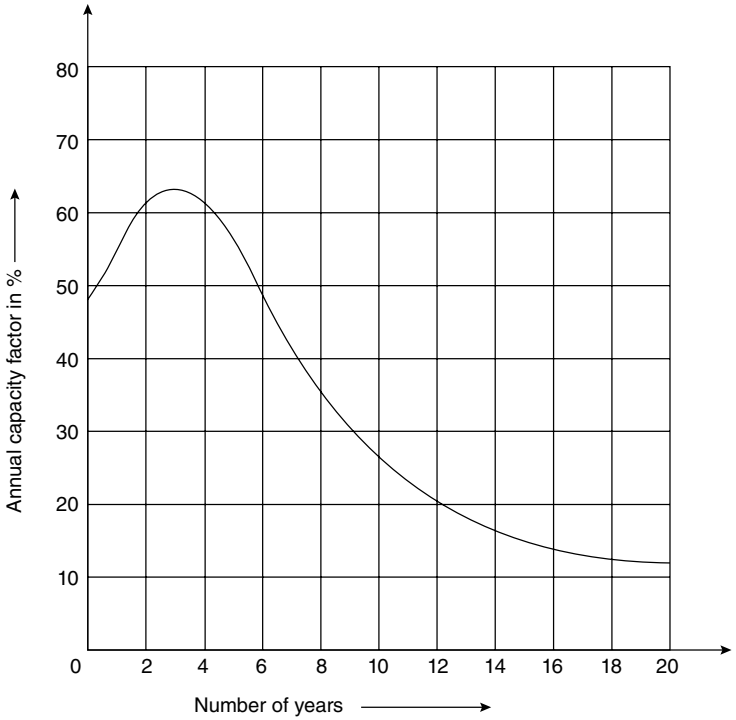


Fig. 17.15 *Effect of Age*

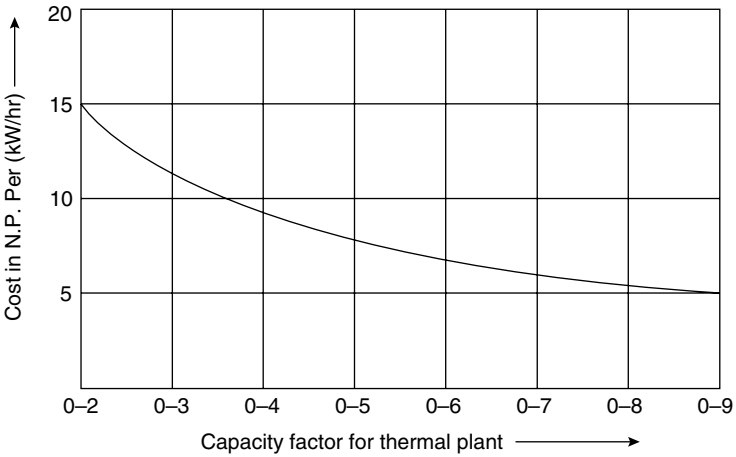


Fig. 17.16 *Effect of Capacity Factor on the Cost of Production*

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- The relation between the capacity factor and use factor in terms of times can be represented as follows:

$$\text{Capacity factor (C.F.)} = \frac{L_a}{L_p} = \frac{\text{Actual power generated/year}}{\text{Maximum possible power generation \year}}$$

$$= \frac{L_a \times 8760}{L_p \times 8760} \tag{1}$$

$$\text{Use factor (U.F.)} = \frac{\text{Actual energy produced per year}}{L_p\, t\,(\text{actual hours of running the plant})\,\text{per year}} \tag{2}$$

From Eqs (1) and (2),

$$\frac{\text{C.F.}}{\text{U.F.}} = \frac{L_a\, 8760}{L_p\, 8760} \times \frac{L_p\, t}{L_a\, 8760} = \frac{t}{8760}$$

Figure 17.17 graphically represents of different factors discussed above.

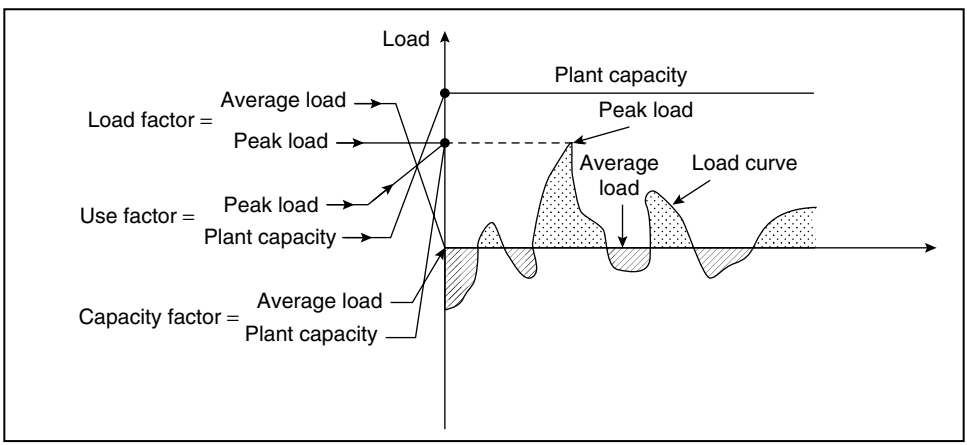


Fig. 17.17 *Graphical Representation of Different Factors*

Example 17.1

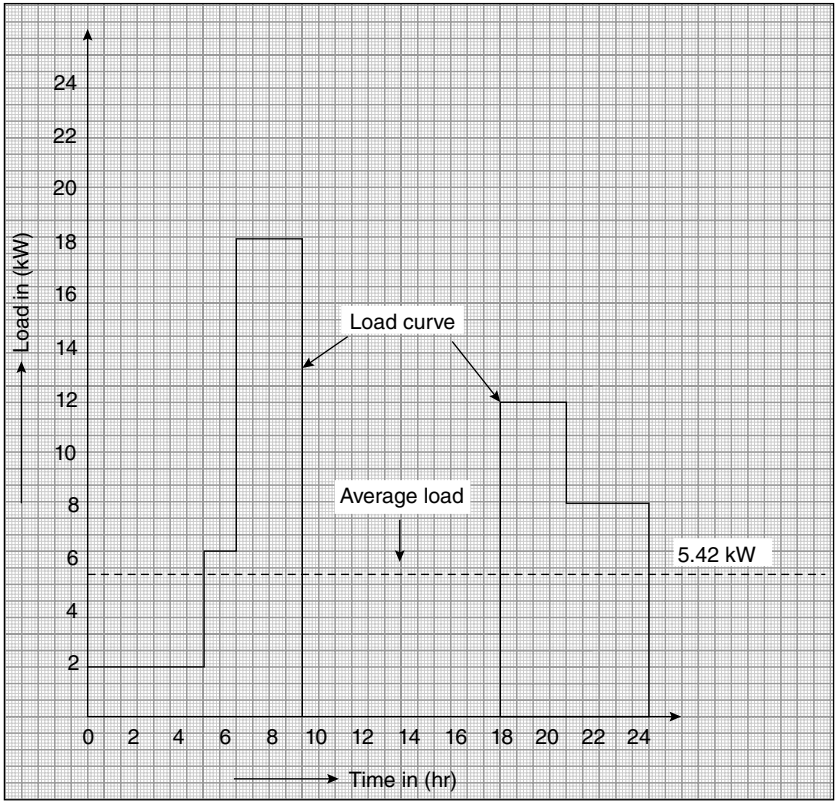
A residential load of a locality is given below:

Time (hr)	0–5	5–6	6–9	9–18	18–21	21–24
Load (kW)	2	6	18	zero	12	8

Draw the load curve and find out the load factor and energy consumed during 24 hr.

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Solution: The load curve is drawn as shown in figure



Total energy consumed during 24 hr

$$= 2 \times 5 + 6 \times 1 + 18 \times 3 + 12 \times 3 + 8 \times 3$$
$$= 10 + 6 + 54 + 36 + 24 = 130 \text{ kW-hr}$$

$$\therefore \text{ Average load} = \frac{136}{24} = 5.42 \text{ kW}$$

$$\text{Load factor} = \frac{5.42}{18} = 0.301$$

Example 17.2

A power station supplies the following loads to the consumers:

Time (hr)	0–6	6–10	10–12	12–16	16–20	20–22	22–24
Load (MW)	30	50	60	40	80	70	40

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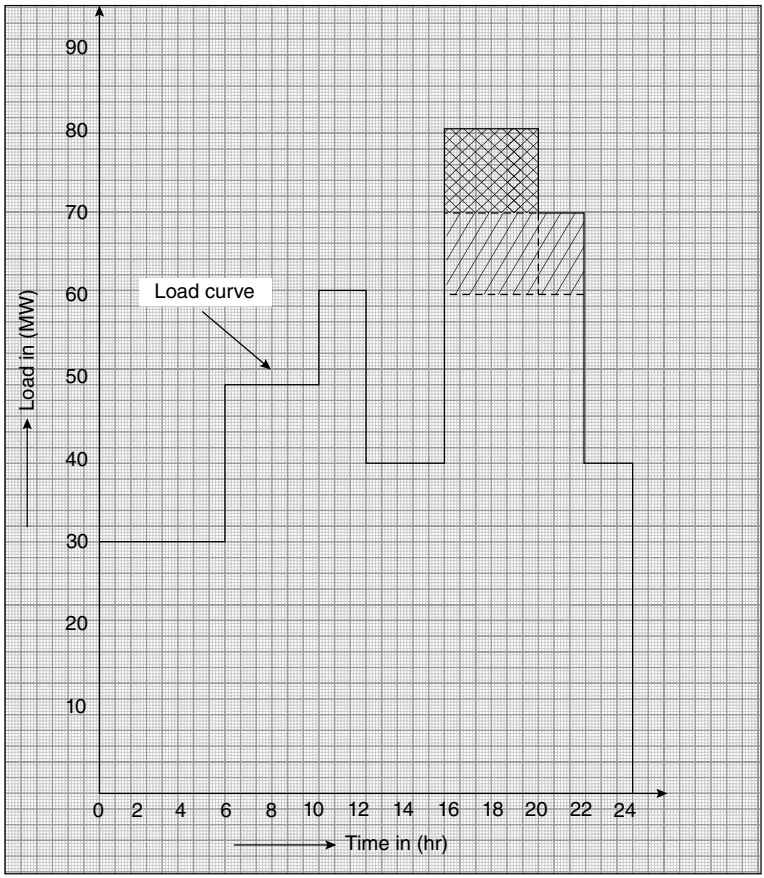
Find (a) the load factor of the plant and (b) what is the load factor of a standby equipment of 30 MW capacity if it takes up all loads above 60 MW?

Solution: The load curve is drawn for the given loads as shown in figure below.
Energy generated = area under the load curve

$$= 30 \times 6 + 50 \times 4 + 60 \times 2 + 40 \times 4 + 80 \times 4 + 70 \times 2 + 40 \times 2$$
$$= 180 + 200 + 120 + 160 + 320 + 140 + 80 = 1{,}200 \text{ MW-hr}$$

$$\text{Average load} = \frac{1200}{24} = 50 \text{ MW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Peak load}} = \frac{50}{80} = 0.625$$



(b) If the load above 60 MW is supplied by a standby unit of 20 MW capacity, the energy generated.

$$= 20 \times 4 + 10 \times 2 = 80 + 20 = 100 \text{ MW-hr}$$

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Time during which stand-by unit remains in operation,
= 6 hr (from the load curve)

$$\therefore \text{Average load} = \frac{100}{6} = 16.7 \text{ MW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Peak load}} = \frac{50}{80} = 0.625$$

Example 17.3

The peak load on a power station is 35 MW. The loads having maximum demands of 15, 10, 5 and 7 MW are connected to the power station. The capacity of the power station is 40 MW and annual load factor is 50 per cent. Find (a) the average load on the power station, (b) energy supplied per year, (c) demand factor and (d) diversity factor.

Solution:

(a) Average load

$$\text{Load factor} = \frac{\text{Average load}}{\text{Peak load}}$$

$$\therefore \text{Average load} = 35 \times 0.5 = 17.5 \text{ MW}$$

(b) Energy supplied per year
= Average load × 8,760 kW-hr
$= 17.5 \times 10^3 \times 8{,}760 = 153.3 \times 10^6$ kW-hr

(c) $$\text{Demand factor} = \frac{\text{Maximum demand}}{\text{Connected load}} = \frac{35}{15+10+5+7} = \frac{35}{37} = 0.946$$

(d) $$\text{Diversity factor} = \frac{\text{Sum of individual maximum demands}}{\text{Simultaneous maximum demand}}$$

$$= \frac{15+10+5+7}{35} = \frac{37}{35} = 1.06$$

Example 17.4

The maximum load on a thermal power plant of 65 MW capacity is 50 MW at an annual load factor of 60 per cent. The coal consumption is 1 kg per unit of energy generated and the cost of coal is ₹ 600 per tonne. Find (a) annual revenue earned if the energy is sold at Re. 1 per kW-hr and (b) the capacity of the plant.

Solution:

(a) Annual revenue earned

$$\text{Annual load factor} = \frac{\text{Average load}}{\text{Peak load}}$$

$$\therefore \text{Average load} = 50 \times 0.6 = 30 \text{ MW} = 30 \times 10^3 \text{ kW}$$

Energy generated per year $= 30 \times 10^3 \times 8{,}760 = 262.8 \times 10^6$ kW-hr

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$$\text{Coal required per year} = \frac{262.8\times10^6}{1000} = 262.8 \times 10^3 \text{ tonnes.}$$

$$\text{Cost of coal per year} = 262.8 \times 10^3 \times 600 = ₹\,15{,}768 \times 10^4$$

$$\text{Cost of energy sold} = \frac{262.8\times10^6}{1} \times 1 = ₹\,26{,}280 \times 10^4$$

$\therefore$ Revenue earned by the power plant per year

$$= 26{,}280 \times 10^4 - 15{,}768 \times 10^4 = ₹\,10{,}51{,}20{,}000$$

Example 17.5

A thermal power plant consists of two 70 MW units each running for 8,000 hr and one 30 MW unit running for 2,000 hr per year. The energy produced by the plant is 876×10^6 kWh per year. Determine plant load factor and plant use factor. Consider maximum demand is equal to plant capacity.

Solution:

$$\text{Plant capacity} = 70 \times 2 + 30 = 170 \text{ MW}$$

$$\text{Average load} = \frac{\text{Energy produced per year}}{8760} = \frac{876\times10^6}{8760}$$

$$= 10^5 \text{ kW} = 100 \text{ MW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Max load}} = \frac{100}{170} = 0.5882 = 58.82\%$$

$$\text{Plant use factor} = \frac{\text{Actual energy produced}}{\left(\text{Max. possible energy which can be produced by the plant}\right)}$$

Max. possible energy which can be produced by the plant,

$$= 2 \times 70 \times 8{,}000 + 1 \times 30 \times 2{,}000$$

$$= 1.18 \times 10^6 \text{ MWh}$$

$$= 1{,}180 \times 10^6 \text{ kWh}$$

$$\therefore \text{ Plant use factor} = \frac{876\times10^6}{1180\times10^6} = 0.742$$

Example 17.6

A power plant supplies the loads having maximum demands of 40, 50 and 30 MW respectively. The load factor of the plant on the basis of annual load curve is 60 per cent and the diversity factor of the load is 1.2. Determine (a) the maximum load on the power plant, (b) the capacity of the power plant required to take the loads and (c) annual energy supplied by the power plant.

Solution:

(a) Diversity factor

$$\text{Diversity factor} = \frac{\text{Sum of individual max. demands}}{\text{Simultaneous max.demamd}}$$

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$\therefore$ Simultaneous max. demand $= \dfrac{40+50+30}{1.2} = \dfrac{120}{1.2} = 100$ MW

This will be the maximum load on the power plant.

(b) Capacity of the power plant

The capacity of the plant can be taken as sum of individual maximum demands.

$= 40 + 50 + 30 = 120$ MW

(c) Load factor

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum load}}$$

$\therefore$ Average load $= 0.6 \times 100 = 60$ MW.

$\therefore$ Energy supplied per year = average load $\times$ 8,760

$= 60 \times 10^3 \times 8{,}760$ kWh $= 525.6 \times 10^6$ kWh

Example 17.7

On the basis of annual operation, the use factor and capacity factor of a central plant are 0.6 and 0.4, respectively. Find out the number of hours of its operation during the year.

Solution:

$$\text{Use factor} = \frac{E}{P_c \times t} \tag{1}$$

$$\text{Capacity factor} = \frac{\text{Average load} \times 8760}{P_c \times 8760} = \frac{E}{P_c \times 8760} \tag{2}$$

Where E = energy produced per year (kWh)

P_c = plant capacity in kW

t = number of hours the plant remained in operation during the year.

Dividing Eqs (1) and (2), we obtain

$$\frac{\text{Use factor}}{\text{Capacity factor}} = \frac{E}{P_c \times t} \times \frac{P_c \times 8760}{E}$$

$$\frac{0.6}{0.4} = \frac{8760}{t}$$

$$t = 5{,}840 \text{ hr}$$

Example 17.8

A diesel power plant consists of two units of 1 MW capacity each and one unit of 200 kW capacity. The fuel used has a calorific value of 40,000 kJ/kg and the fuel consumption is 0.25 kg/kWh. Determine the quantity of fuel required for a month of 30 days and its cost if the fuel cost is ₹ 4,000 per tonnes. Also find the overall efficiency of the plant. **Take plant capacity factor on monthly basis = 50%.**

Solution:

$$\text{Capacity factor} = \frac{E}{P_c \times t}$$

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Where E = energy generated during the month.
P_c = plant capacity.
And t = time in hours during the month.

$$P_c = 1{,}000 + 1{,}000 + 200 = 2{,}200 \text{ kW.}$$
$$t = 30 \times 24 = 720 \text{ hr.}$$
$$\therefore \quad 0.5 = \frac{E}{2200 \times 720}$$
$$\therefore \quad E = 2{,}200 \times 720 \times 0.5 = 792 \text{ MWh/month.}$$

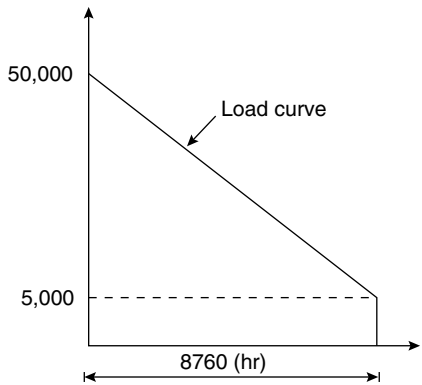
Fuel consumption per month,

$$= 0.25 \times 792 \times 1{,}000 = 198{,}000 \text{ kg}$$
$$= 198 \text{ tonnes.}$$
$$\therefore \text{ Fuel cost} = 198 \times 4{,}000 = ₹\, 7{,}92{,}000/\text{month}$$
$$\therefore \text{ Cost of energy} = \frac{792000}{600 \times 720} = ₹\, 1.83/\text{kW-hr}$$
$$\text{Overall efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{792{,}000 \times 3600}{198000 \times 40{,}000} = 0.36 = 36\%$$

Example 17.9

An yearly load duration curve of a gas turbine power plant is a straight line from 50,000 to 5,000 kW. The load is taken by a power plant that consists of two units of 30,000 kW each and one unit of 20,000 kW. Determine (a) load factor and (b) capacity factor of the plant.

Solution:
The load curve is shown in figure below.



The energy generated per year by the plant = area under the load curve

$$= 8{,}760 \times 5{,}000 + \frac{1}{2} \times 8{,}760\ (50{,}000 - 5{,}000)$$
$$= 8{,}760\ (5{,}000 + 22{,}500)$$
$$= 8{,}760 \times 27{,}500$$
$$= 240.9 \times 10^6 \text{ kWh/year}$$

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$$\text{Average load} = \frac{8760 \times 275,00}{8760} = 27,500 \text{ kW}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}} = \frac{27500}{50000} = 0.55$$

$$\text{Capacity factor} = \frac{\text{Energy generated/year}}{\text{Plant capacity} \times 8760}$$

$$\text{Plant capacity} = 30,000 \times 2 + 20,000 = 80,\ 000 \text{ kW}$$

$$\text{Capacity factor} = \frac{27500 \times 8760}{80,000 \times 8760} = 0.344$$

Example 17.10

A load duration curve for an industrial load is served by hydro-thermal plants follows a straight line. The maximum and minimum loads are 30 and 10 MW, respectively. The hydel-plant is available at the time of minimum regulating flow which is just enough to take a peak load of 75 MWh per day. On investigation, it was found economical to pump water from lower reservoir to higher reservoir using the excess power from steam plant during peak period allowing steam plant to run at 100 per cent load all the times. Determine the capacities of hydel and steam plants required for the purpose.

Assume efficiency of steam to electric conversion = 45%

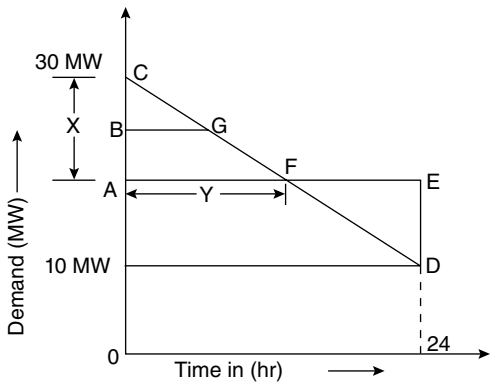
Solution:

The load duration curve for the combined system is shown in figure. Let OA be the capacity of thermal plant CGB is the energy available in low flow period and FED be the energy available during off-peak period. BGFA is the corresponding energy supplied from pump storage plant. As steam-electric conversion is 45%

$$\therefore \text{BGFA} = 0.45 \times \text{area FED} \tag{1}$$

$$\text{Area BGFA} = \text{area CAF} - \text{area CBG}$$

$$= \frac{1}{2} xy - 75$$



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$$\text{and area FED} = \frac{1}{2}\ \text{ED} \times \text{FE} = \frac{1}{2}\ (20 - x)\ (24 - y)$$

Substituting in Eqn. (1) we have,

$$\frac{1}{2}\ xy - 75 = 0.45 \times ½ \times (20 - x)\ (24 - y)$$

$$\therefore\ \frac{1}{2} xy - 150 = 0.45\ (20 - x)\ (24 - y)$$

$$= (480 - 24x - 20y + xy)\ 0.45$$

$$= 216 - 10.8x - 9y + 0.45xy \qquad (2)$$

Also $$\frac{x}{y} = \frac{20}{24}\ \text{(Similar triangles)} \qquad (3)$$

Substituting $y = 1.2x$ from Eqn. (3) into Eqn. (2), we obtain

$x(1.2x) - 150 = 216 - 10.8x - 9\ (1.2x) + 0.45x\ (1.2x)$

$\therefore\ 1.2x^2 - 150 - 216 + 10.8x + 10.8x - 0.54x^2 = 0$

$\therefore\ 0.66x^2 + 21.6x - 371 = 0$

$$x = \frac{21.6 \pm \sqrt{21.6^2 + 4 \times 371 \times 0.66}}{2 \times 0.66}$$

$$= \frac{21.6 \pm 38.03}{1.32}$$

$$x = 12.45$$

$\therefore\ x$ = Hydel plant capacity = 12.45 MW

Steam plant capacity = 30 − 12.45 = 17.55

Example 17.11

The yearly duration curve for a certain plant can be considered as a straight line from 20 to 4 MW power supplied with one generator rated at 5 MW and two units of 10 MW capacity each. Determine

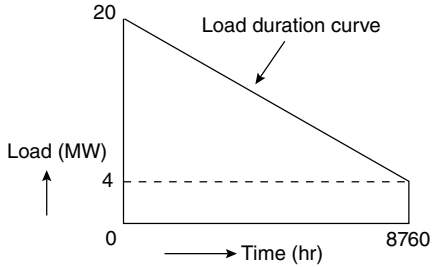
(a) Installed capacity

(b) Plant factor

(c) Maximum demand

(d) Load factor

(e) Utilization factor



Solution:

The load duration curve is shown below:

(a) Installed capacity = 2 × 10 + 5 = 25 MW

(b) Plant factor or capacity factor or use factor

$$\text{Plant factor} = \frac{\text{Average load}}{\text{Capacity of the plant}}$$

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$$\text{Average load} = \frac{\text{Total energy in load curve for the period}}{\text{Total number of hours in the period}}$$

$$= \frac{8760 \times 4 + \frac{1}{2} \times 8760 \ (20 - 4)}{8760} = \frac{105120}{8760}$$

$$\text{Average load} = 12 \text{ MW}$$

$$\text{Plant factor} = \frac{12}{25} = 0.48$$

(c) Maximum demand = 20 MW

(d) $$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}}$$

$$= \frac{12}{20} = 0.6$$

(e) $$\text{Utilization factor} = \frac{\text{Maximum load}}{\text{Rated capacity of the plant}}$$

$$= \frac{20}{25} = 0.8 = 80\%$$

Example 17.12

A power plant has the following annual factors: Load factor = 70 per cent; capacity factor = 50 per cent. Use factor = 60 per cent. Maximum demand is 20 MW. Find out the following:

(a) Annual energy production

(b) Reserve capacity over and above peak load

(c) Hours during which the plant is not in service per year

Solution:

$$\text{Load factor} = \frac{\text{Average load}}{\text{peak load}}$$

Average load = $20 \times 0.7 = 14$ MW

(a) Annual energy produced = $14 \times 10^3 \times 365 \times 24$

$= 122.64 \times 10^6$ kW-hr

$$\text{Annual capacity factor} = \frac{\text{Average load}}{\text{peak capacity}}$$

$$\text{Plant capacity} = \frac{14}{0.5} = 28 \text{ MW}$$

(b) Reserve capacity over and above peak load,

$28 - 20 = 8$ MW

(c) $$\text{Annual use factor} = \frac{\text{Energy generated per year}}{\text{Plant capacity} \times t_1}$$

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where t_1 is the actual number of hours of the year for which plant remains in operation.

$$0.6 = \frac{122.64 \times 10^6}{28 \times 10^3 \times t_1}$$

$$\therefore\ t_1 = 7300 \text{ hr/year}$$

Hours not in service = 8,760 − 7,300 = 1,460 hr/year

Example 17.13

A base load power station and a standby power station share a common load as follows:
Base load station annual output = 150×10^6 kWh
Base load station capacity = 25 MW
Maximum demand on base load station = 30 MW
Stand by station capacity = 18 MW
Maximum demand on standby station = 15 MW
Stand-by-station annual output = 14×10^6 kWh
Maximum demand on standby station = 15 MW
Determine the following for both power stations: (a) load factor and (b) capacity factor

Solution:
Base load station,

$$\text{Average load} = \frac{150 \times 10^6}{365 \times 24} = 17123.3 \text{ kW}$$

$$\text{(i) Load factor} = \frac{\text{Average load}}{\text{Maximum demand}} = \frac{17123.3}{30 \times 10^3} = 0.57$$

$$\text{(ii) Capacity factor} = \frac{\text{Energy generated}}{\text{Capacity of plant} \times 365 \times 24}$$

$$= \frac{150 \times 10^6}{35 \times 1000 \times 24 \times 365} = 0.489$$

Stand by station,

$$\text{Annual average load} = \frac{14 \times 10^6}{365 \times 24} = 1598.17 \text{ kW}$$

$$\text{(i) Load factor} = \frac{\text{Average load}}{\text{Maximum demand}} = \frac{17123.3}{15 \times 1000} = 1.14$$

$$\text{(ii) Capacity factor} = \frac{\text{Energy generated}}{\text{Capacity} \quad 24 \times 365} = \frac{14 \times 10^6}{18 \times 24 \times 365 \times 1000} = 0.088$$

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Example 17.14

The peak load on a power station is 40 MW. The loads having maximum demands of 18, 12, 8 and 9 MW are connected to the power station. The capacity of the power station is 50 MW, annual load factor is 62 per cent find,

(i) Average load on the power station

(ii) Energy supplied per year

(iii) Demand factor

(iv) Diversity factor

Solution:

Power station capacity = 50 MW

Maximum demand on power station = 40 MW

Maximum demands on the transformers

$$= 18 + 12 + 8 + 9 = 47 \text{ MW}$$

Annual load factor = 62% = 0.62

plant operating period = $0.62 \times 365 \times 24 = 5431.2$ hr

(i) Average load on station = maximum demand × load factor

$$= 40{,}000 \times 0.62$$

$$= 24{,}800 \text{ kW}$$

(ii) Energy supplied per year = average load × 365 × 24

$$= 24{,}800 \times 8{,}760$$

$$= 217248.0 \text{ MWh}$$

(iii) $$\text{Demand factor} = \frac{\text{Maximum demand}}{\text{Sum of connected load}}$$

$$= \frac{40000}{47000} = 0.851$$

(iv) $$\text{Diversity factor} = \frac{\text{Maximum demand}}{\text{Sum of connected load}}$$

$$= \frac{47000}{40000} = 1.175$$

Example 17.15

The maximum demand of a power station is 26,000 kW and daily load curve is described as follows:

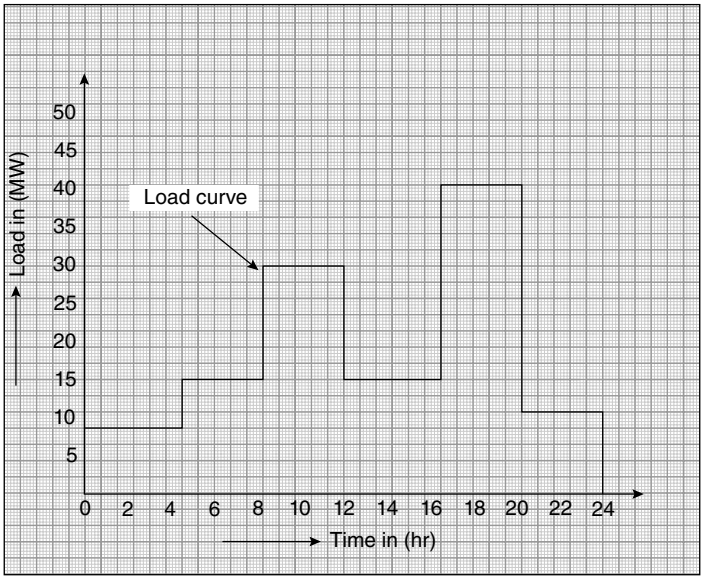
Time (hr)	0–4	4–8	8–12	12–16	16–20	22–24
Load (MW)	8	15	28	15	42	8

(i) Determine the load factor

(ii) What is the load factor of stand by equipment rated at 8 MW that takes up all load in excess of 15 MW. Calculate use factor also.

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Solution:
The daily load curve is shown below:



(i) Load factor

Energy generated = area under load curve

$$= 8 \times 4 + 4 \times 15 + 28 \times 4 + 15 \times 4 + 42 \times 4 + 8 \times 4$$

$$= 464 \text{ MWh} = 464 \times 10^3 \text{ kWh}$$

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}}$$

$$\text{Average load} = \frac{464 \times 10^3}{24} = 19333.33 \text{ kW}$$

$$\therefore \quad \text{Load factor} = \frac{19333.33}{26000} = 0.744$$

(ii) Load factor of stand by equipment

Standby equipment supplies, $42 - 15 = 27$ MW for 4 hr $(16 - 20)$

$28 - 15 = 13$ MW for 4 hr $(8 - 12)$

$\therefore$ Energy generated by stand by equipment,

$$= (27 \times 4 + 13 \times 4) \times 10^3 = 160 \times 10^3 \text{ kWh}$$

Time for which standby equipment is in operation = 8 hr

$$\text{Average} = \frac{160 \times 10^3}{8} = 20{,}000 \text{ kW}$$

$$\text{Load factor} = \frac{20000}{24 \times 10^3} = 0.833$$

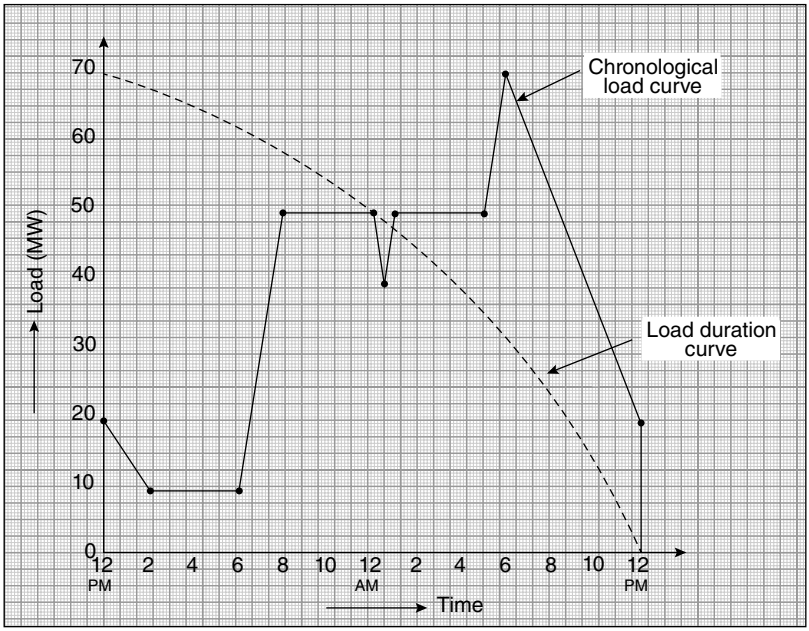
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Example 17.16

Draw the chronological daily load curve and the load duration curve from the following data. Find the load factor and utilization factor of the plant serving this load if its capacity is 90 MW.

Time	12 PM	2 AM	6 AM	8 AM	12 AM	12.30 PM	1 PM	5 PM	6 PM	12 PM
Load (MW)	18	8	8	48	48	38	48	48	68	18

Solution:
Chronological load and load duration curve is shown below:



$$\text{Average load for the period} = \frac{\text{Total energy in load curve for period}}{\text{Total number of hours period}}$$

Total energy in load curve for period

$$= \frac{(18+8)}{2} \times 2 + 8 \times 4 + \frac{(8+48)}{2} \times 2 + 48 \times 4 + \frac{1}{2}\left(\frac{48+38}{2}\right) + \frac{1}{2}\left[\frac{(38+48)}{2}\right] + 48 \times 4 + \frac{(48+68)}{2} \times 1 + \frac{(48+8)}{2} \times 6$$

$$= 26 + 32 + 56 + 192 + 86 + 86 + 192 + 58 + 198$$
$$= 926$$

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$$\text{Average load} = \frac{926}{24} = 38.58 \text{ MW}$$

$$\text{Load factor} = \frac{38.58}{68} = 0.57 = 57\%.$$

$$\text{Utilization factor} = \frac{\text{Maximum load}}{\text{Rated capacity of the plant}}$$

$$= \frac{68}{90} = 0.75 = 75\%.$$

Example 17.17

A power station has a maximum demand of 80×10^3 kW and daily load curve is defined as follows:

Time (hr)	0–6	6–8	8–12	12–14	14–18	18–22	22–24
Load (MW)	40	50	60	50	70	80	40

(a) Determine the load factor of power station.

(b) What is the load factor of standby equipment raid at 25 MW that takes up all load in excess of 60 MW? Also calculate its use factor.

Solution:

The load curve is shown below:

$$\text{Energy generated} = 40 \times 6 + 50 \times 2 + 60 \times 4 + 50 \times 2 + 70 \times 4 + 80 \times 4 + 40 \times 2$$
$$= 1{,}360 \text{ MWh} = 1{,}360 \times 10^3 \text{ kWh}$$

$$\text{Average load} = \frac{1360 \times 10^3}{24} = 56{,}666 \text{ kW}$$

Maximum demand $= 80 \times 10^3$ kW

(a) $$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}} = \frac{56{,}666}{80 \times 10^3} = 0.71$$

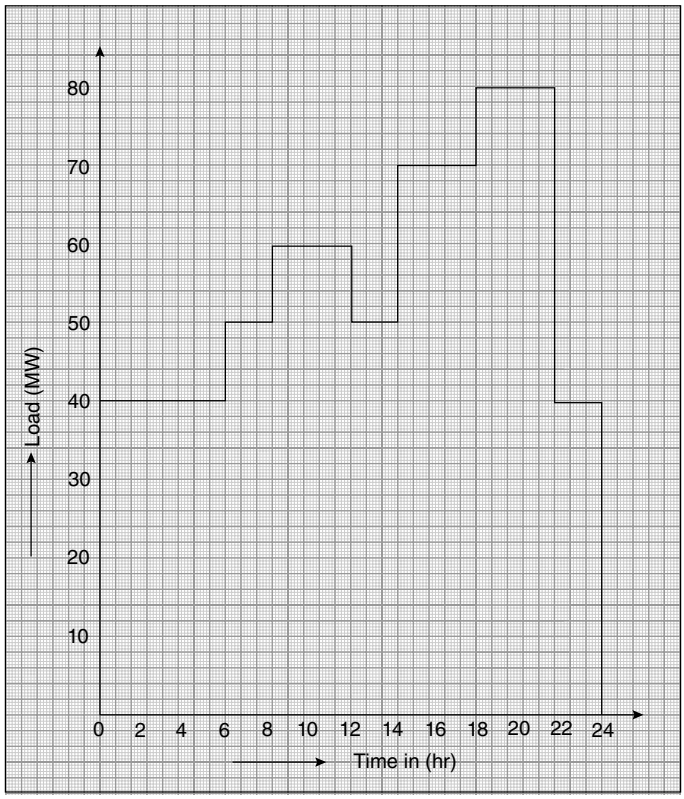
(b) Standby equipment supplies,
$70 - 60 = 10$ for 4 hr
$80 - 60 = 20$ for 4 hr

Energy generated by standby equipment,
$$= 10 \times 4 + 20 \times 4$$
$$= 120 \times 10^3 \text{ kWh}$$

Standby equipment remains in operation for 8 hr,

$$\text{Average load} = \frac{120 \times 10^3}{8} = 15 \times 10^3 \text{ kW}$$

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$$\text{Load factor} = \frac{15\times10^3}{20\times10^3} = 0.75$$

$$\text{Use factor} = \frac{E}{C\times t_1} = \frac{120\times10^3}{25\times10^3\times8} = 0.6$$

Example 17.18

A central power station has annual factors as follows:
Load factor = 60%
Capacity factor = 40%
Use factor = 45%
Power station has a maximum demand of 15,000 kW. Determine (a) annual energy production, (b) reserve capacity over and above peak load and (c) hours per year not in service.

Solution:

(a) *Annual energy production*

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum demand}}$$

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Average load = $0.6 \times 15{,}000 = 9{,}000$ kW

Annual energy produced = $9{,}000 \times 365 \times 24$

$$E = 78.84 \times 10^6 \text{ kWh}$$

(b) *Reserve capacity above peak load*

(b) Capacity factor = $\dfrac{E}{C\,t}$

$$C = \frac{78.84 \times 10^6}{0.4 \times 365 \times 24} = 22{,}500 \text{ kW}$$

Reserve capacity = $22{,}500 - 15{,}000 = 7{,}500$ kW

(c) *Hours per year not in service*

$$\text{Use factor} = \frac{E}{C \times t_1}$$

$$\therefore\ t_1 = \frac{78.84 \times 10^6}{0.45 \times 22500} = 7{,}786.67 \text{ hr}$$

Hour per year not in service = $365 \times 24 - 7{,}786.67$

$= 973.33$ hr

Example 17.19

A power station has two 60 MW unit each running for 7,000 hr a year and an 30 MW unit running for 1,500 hr a year. The energy produced per year is 700×10^6 kWh, calculate the following:

(a) Plant load factor

(b) Plant use factor

Solution:

(a) *Plant load factor*

$$\text{Average load} = \frac{700 \times 10^6}{365 \times 24} = 79908.67 \text{ kW}$$

capacity of plant = $60 \times 2 + 30 = 150 \times 10^3$ kW

$$\therefore \quad \text{Load factor} = \frac{700 \times 10^6}{365 \times 24 \times 150 \times 10^3} = 0.533$$

(b) *Plant use factor*

Energy that could be generated = $2 \times 60 \times 7{,}000 + 30 \times 1{,}500$

$= 885 \times 10^6$ kWh

Actual energy generated = 700×10^6 kWh

$$\therefore \text{ Use factor} = \frac{700 \times 10^6}{885 \times 10^6} = 0.79$$

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17.4 COST OF ELECTRICAL ENERGY

The total cost of power generated by a power station is sub-divided as follows:

(a) *Fixed cost*

It includes the following:

- Capital cost: Cost of land, building, equipment, installation and cost of designing and planning the station
- Capital cost of primary distribution systems such as cost of sub-stations, transmission lines, etc.
- Interest, taxes and insurance on the capital cost
- General management cost
- Depreciation cost

(b) *Energy cost*

It includes the following:

- Cost of fuel
- Cost of operating labour
- Cost of maintenance, labour and materials
- Cost of supplies such as (1) water for feeding boilers, condensers and for general use, (2) lubricating oils and (3) water treatment chemicals, etc.

(c) *Customer charges*

These charges depend on the number of customers. Various costs include the following:

- Capital cost of secondary distribution system and depreciation cost, taxes and interest on this capital cost.
- Cost of inspection and maintenance of distribution lines and the transformers
- Cost of labour required for meter reading and office work
- Publicity cost

(d) *Investors profit:* The investor expects a satisfactory return on the capital investment. The rate of profit varies according to the business conditions prevailing in different localities.

17.4.1 Depreciation

Depreciation cost is the amount to be set aside per year from the income to meet the depreciation caused by the age of service, wear and tear of the machinery (Figure 17.18). The amount collected per year is helpful in replacing and repairing the equipment. Several methods are used to calculate depreciation. Two most commonly used methods are the following:

(a) *Straight line method:* According to this method, depreciation occurs uniformly according to a straight line law.

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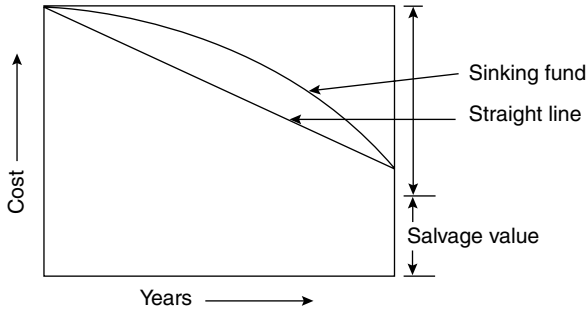


Fig. 17.18 *Depreciation Methods*

If A = annual depreciation reserve
= annual deposit in the reserve fund, in rupees
P = principal sum or capital cost of plant
S = salvage value or residual value of power plant at the end of n year
n = expected useful life, in years

then,

$$A = \frac{P - S}{n}$$

The amount collected every year as depreciation fund does not depend on the interest it may draw. Any amount collected due to the interest is considered as income.

(b) *Sinking fund method:* In this method, the amount collected every year consists of annual instalments plus the interest on all the instalments

If I = rate of compound interest, then annual depreciation reserve is

$$A = (P - S)\left[\frac{I}{(1+I)^N - 1}\right]$$

Example 17.20

If the expected life of equipment is 15 years and the interest earned on the capital investment is 5 per cent, find the depreciation rate.

Solution:

Depreciation rate is given by the following equation:

$$A/S = \left[\frac{I}{(1+I)^N - 1}\right]$$

$$= \left[\frac{0.05}{(1+0.05)^{15} - 1}\right] = 0.0463$$

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Example 17.21

The original value of an equipment is ₹ 2,50,000 and its salvage value at the end of its useful life of 20 years is ₹ 25,000. Find the value of the equipment at the end of 10 years of its use by the following methods: (a) straight line depreciation and (b) sinking fund depreciation, when it is compounded annually at 8%.

Solution:

(a) *Straight line method*

Original value = ₹ 2,50,000
Salvage value = ₹ 25,000
Total depreciation = 2,50,000 − 25,000
= ₹ 2,25,000

Depreciation per year,

$$= \frac{2,25,000}{20} = ₹\,11,250$$

Depreciation at the end of 10 years,

$$= 11,250 \times 10 = ₹\,1,12,500$$

Value of the equipment at the end of 10 years,

$$= 2,50,000 - 1,12,500$$
$$= ₹\,1,37,500$$

Or

$$A = \frac{\mathrm{P-S}}{\mathrm{n}} = \frac{2,50,000 - 25,000}{20}$$

$$= ₹\,11,250 \text{ per year}$$

At the end of 10 years amount collected

$$P_1 = \mathrm{A} \times \mathrm{n} = 11,250 \times 10 = ₹\,1,12,500$$

Value of the equipment is = 2,50,000 − 1,12,500 = ₹ 1,37,500

(b) *Sinking fund method*

Annual deposit,

$$\mathrm{A} = (P - S)\left[\frac{I}{(1+I)^N - 1}\right]$$

$$= (2,50,000 - 25,000)\left[\frac{0.08}{(1+0.08)^{20} - 1}\right]$$

$$= ₹\,4,916.75$$

Total amount collected at the end of 10 years.

$$(P - S)10 = \mathrm{A}\left[\frac{(1+I)^N - 1}{I}\right] = 4916.75\left[\frac{(1+0.08)^{10} - 1}{0.08}\right]$$

$$= ₹\,71,226.81$$

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Value of the equipment at the end of 10 years.

$$= 2{,}50{,}000 - 71{,}226.81$$

$$= ₹\,1{,}78{,}773.19$$

17. 5 ENERGY RATES (TARIFFS) FOR ELECTRICAL ENERGY

Cost of generation of electric energy consists of fixed cost and running cost. Since the electricity generated is to be supplied to the consumers, the total cost of generation has to be recovered from the consumers. For this purpose, some form of energy rates are evolved known as tariffs.

The energy rate (tariff) must include the following:

(a) Recovery of cost of capital investment in generating equipment, transmission and distribution system, and other operating equipment.

(b) Recovery of cost of the operation, supplies and maintenance of equipment

(c) Recovery of cost of maintenance equipment, billing and collection costs, and miscellaneous services.

(d) An adequate return on the total capital investment

If Y = total amount of the bill

d = a constant charge made on each bill

e = unit charge for metered or allowed maximum demand

during billing period

x = Maximum demand during billing period

f = unit charge for energy, Rs/kWh

z = total energy consumed during billing period, kWh

then,

$$Y = d + ex + fz$$

The above equation is known as general energy rate equation.

Any energy rate or tariff should have the following characteristics: (i) simplicity, (ii) fairness, (iii) uniformity and (iv) incentive.

17.5.1 Different Types of Tariffs

Different types of tariffs are discussed in this section.

(a) *Flat rate demand:* This is one of the earliest forms of energy tariffs. In this case, the total demand and the energy consumption were fixed. It is based on the number of lamps installed and a fixed number of hours of use of these lamps in 1 month or year. The rate is expressed as price per lamp per unit of demand (kW) of the consumer (Figure 17.19). This energy rate eliminates the use of metering equipment. This tariff is expressed as follows:

$$Y = \text{ex}$$

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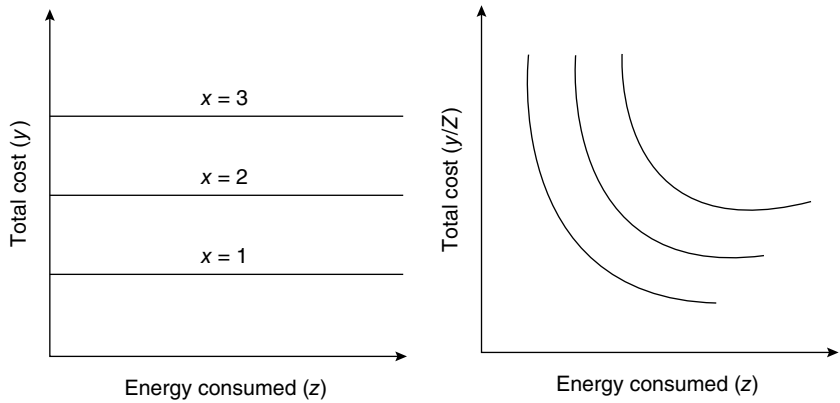


Fig. 17.19 *Flat Rate Demand*

(b) *Straight line metre rate:* This is the simplest form of tariff. In this method, the charge per unit is constant. This system does not encourage the increased consumption of electricity unless the tariff is very low (Figure 17.20). According to this method, the amount to be charged from the consumer depends on the energy consumed in kilo-watt hour, which is recorded by a means of a kilowatt hour metre. This tariff is written as $Y = fz$

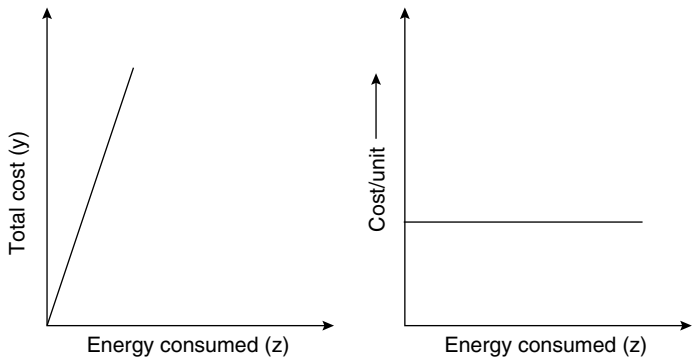


Fig. 17.20 *Straight Line Meter Rate*

The disadvantages of this system is that a consumer who is not using the energy need not pay any amount even though he has incurred some expense to the power station due to its readiness to serve him.

(c) *Step meter rate:* In this system, there are different tariffs for different steps of energy consumption (Figure 17.21).

E.g. 15 paise unit for a consumption less than 200 kWh
10 paise/unit for a consumption more than 200 kWh, etc.

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In this system, the rates of unit go on decreasing with the increased consumption of electric energy.

i. e. $y = fz$ if $o \le z \le A$

$y = f_1 z_1$ if $A \le z_1 \le B$

$y = f_2 z_2$ if $B \le z_2 \le C$

where A, B and C are different steps of energy consumption.

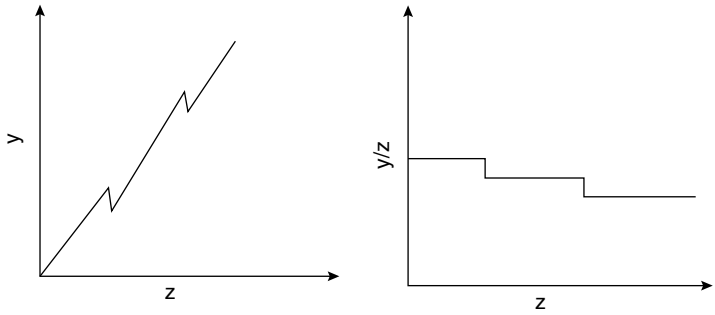


Fig. 17.21 *Step Meter Rate*

The disadvantages of this system is that a consumer who is not using the energy need not pay any amount even though he has incurred some expense to the power station due to its readiness to serve him.

(d) *Block metre rate:* The major disadvantage of the step metre rate method is the inconsistency in the total cost when the energy consumption approaches the upper limit of any step. If the consumption is increased so as to enter the next step for which the unit rate is less, the total energy cost reduces. This is modified in the block metre rate in which the total energy cost remains constant (Figure 17.22).

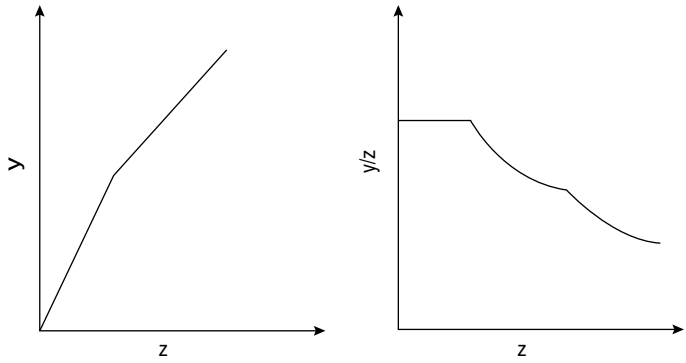


Fig. 17.22 *Block Metre Rate*

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In this method, different rates are charged for different blocks of consumption. The size of the block depends on the size of the consuming body. With increased consumption of electric energy, the unit rate will go on decreasing.

For example,

35 paise/unit for the first 15 units/month

20 paise/unit for any part of the next 25 units/month

10 paise/unit for the excess over 25 units/month

It is expressed as follows:

$$Y = f_1 z_1 + f_2 z_2 + f_3 z_3 \ldots$$

where f_1, f_2 and f_3 are the unit charge for energy in Rs/kWh and z_1, z_2, z_3, etc., are the total energy consumed during billing period, kWh.

(e) *Two part tariff (Hopkinson demand rate):* In all the previous methods, the total energy cost is not adequately allocated. The charges were based on only the item, whereas the total energy cost varies with three items. In the two-part tariff system, the charges for demand and energy are included. The demand charge is based on a maximum monthly demand and the energy charge is based on follow on block rate.

Thus, $y = ex + fz$

A separate meter is required to record the maximum demand. This tariff is used for industrial loads.

(f) *Three part tariff (Doherty rate):* According to this system, the customer pays some fixed amount in addition to the charges for maximum demand and energy consumed. The fixed amount to be charged depends on the occasional increase in fuel price, rise in wages of labour, etc.

Thus, $y = ex + fz + \mathrm{C}$

(g) *Wright demand rate:* The Wright demand rate gives much more incentive to the consumer to lower his demand by lowering both the demand and energy charges. Keeping the demand at a minimum, this system improves the load factor. This system is recommended for industrial consumers.

(h) *Power factor tariffs:* In this system, a penalty is imposed on the consumer if the power factor of load drops below a specified limit. Two methods are used.

(1) Maximum kilovolt-amps demand tariff: In this method, the maximum demand charge is not measured on the basis if maximum demand in kilo-watt but on the basis of maximum demand in Wright demand rate.

(2) Power factor penalty tariff: In this method, the power factor below a certain stated value say 0.8 or 0.85 is penalized. A certain sum is added to the kilo-watt maximum demand charge for every 0.1 fall in the power factor.

Example 17.24

The load demand of an industrial consumer is given below:

300 kW for 1 hr/day

250 kW for 7 hr/day

75 kW for 8 hr/day

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Estimate the yearly energy cost if two part tariff is ₹ 100/kW + 10 paise per kWh.

Solution:

Maximum demand = 300 kW

$\therefore$ Demand charge = 300×100

= ₹ 30,000

Considering 6 days a week,

total units consumed per year,

$= 300 \times 1 \times 6 \times 52 + 250 \times 7 \times 6 \times 52 + 75 \times 8 \times 6 \times 52$

$= 6 \times 52\ (300 + 1{,}750 + 600)$

= 8,26,800 kWh

$\therefore$ Energy cost = $8{,}26{,}800 \times 0.10$ = ₹ 82,680

$\therefore$ Total annual cost = ₹ 30,000 + ₹ 82,680

= ₹ 1,12,680

$$\therefore \text{Average cost per unit} = \frac{112680}{826800} \times 100$$

= 13.6 paise

Example 17.25

The annual consumption of a customer is $1{,}500 \times 10^3$ units and the annual load factor is 40 percent. If the load factor improves to 100 per cent, what will be the saving in the average unit cost. The two part tariff is ₹ 110 per kW per year + 10 paise per kWh.

Solution:

Assuming the customer is domestic customer.

$$\text{Average load} = \frac{1500 \times 10^3}{24 \times 365} = 171.23 \text{ kW}$$

$\therefore$ Maximum demand = 171.23/0.4 = 427.5 kW

Demand charge = 427.5×110 = ₹ 47,025

Energy charge = $1{,}500 \times 10^3 \times 10/100$

= ₹ 1,50,000

$\therefore$ Total cost = demand charge + energy charge

= Rs 1,97,025

$$\therefore \text{Average unit rate} = \frac{197025}{1500 \times 10^3} \times 100 = 13.14 \text{ paise}$$

If the consumer takes his demand energy at 100 per cent load factor, then

$$\text{Maximum demand} = \frac{171.23}{1} = 171 \text{ kW}$$

Demand charge = 171.23×110

= ₹ 18,835.3

Energy charge = ₹ 1,50,000

$\therefore$ Total cost = demand charge + energy charge

= 18,835.3 + 1,50,000

= ₹ 1,68,835.3

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$$\text{Average unit rate} = \frac{168835.3}{1500 \times 10^3} \times 100$$
$$= 11.26 \text{ paise}$$

Hence, there is a saving of 1.88 paise per unit.

Example: 17.26

A two tariff is quoted as below:
Demand:

- First 5 kW of maximum demand at Rs 50/kW
- Next 5 kW of maximum demand at Rs 40/kW
- Excess over 10 kW maximum demand at Rs 25/kW

Energy:

- First 100 units at 25 paise/kWh
- Next 100 units at 20 paise/kWh
- Next 300 units at 15 paise/kWh
- Excess over 500 units at 10 paise/kWh

Calculate the monthly bill and the total cost per unit in following cases:

(a) Maximum demand 7 kW and energy consumed 175 kWh

(b) Maximum demand 15 kW and energy consumed 600 kWh

Solution:

(a) $\text{Demand charge} = 50 \times 5 + 40 \times 2$
$= 250 + 80$
$= ₹\,330/\text{year}$

$$\text{Energy charge} = 100 \times \frac{25}{100} + 75 \times \frac{25}{100}$$
$$= 25 + 15$$
$$= ₹\,40.$$

$$\text{Monthly bill} = \frac{330}{12} + 40$$
$$= ₹\,67.50$$

$$\text{Unit rate} = \frac{67.50}{175} \times 100$$
$$= 38.6 \text{ paise}$$

(b) $\text{Demand charge/year} = 5 \times 50 + 5 \times 40 + 5 \times 25$
$= ₹\,575$

$\text{Energy charge/month} = 100 \times 0.25 + 100 \times 0.20 + 300 \times 0.15 + 100 \times 0.1$
$= ₹\,100$

$$\therefore \quad \text{Monthly bill} = \frac{575}{12} + 100$$
$$= ₹\,147.92$$

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$$\therefore \text{Total cost per unit} = \frac{147.92 \times 100}{600}$$
$$= 24.7 \text{ paise}$$

17.6 SELECTION OF TYPES OF GENERATION AND GENERATING EQUIPMENT

After selecting the type of drive based on availability of cheap fuels or water resources such as steam, gas diesel or water power, further selection of the design and size of the equipment is done primarily based on economic consideration and the lowest unit cost of production. The working efficiency of all types of equipment with larger sizes of plants and with high load factor operation is generally higher. Increased plant size also reduces with the capital cost per unit installation. However, with a bigger size, investments are heavier. When the plant size is bigger, it is possible that load factor lower than optimum may be reached.

17.6.1 Steam Power Plants

In case of steam power plants, the choice of steam conditions, namely pressure and temperature is an important factor affecting operating costs. By operating the plant at higher pressure and temperature, the cycle efficiency increases at the increased capital cost. Further improvement in efficiency is accomplished by reheating and regeneration. However, the capital cost further increases which should be balanced with gain in operating cost. The use of boiler accessories such as air pre-heaters and economizers may be further considered from economic view point.

17.6.2 Diesel Engine Plants

Based on thermodynamic considerations selection of diesel engine (DG) plants is made. Operating the engines at higher compression ratio (pressure) improves the engine efficiency but at the cost of initial investment. A supercharged engine could be utilized only if the cost of the supercharger is justified with substantial gain in engine power. If the engine is a four-stroke engine its thermal efficiency is higher and if the engine is a two-stroke engines, it offers the benefits of lower weight and cost.

17.6.3 Gas Turbine Power Plant

A gas turbine plant is an obvious choice if the gaseous fuel is available in plenty and the cost of inclusion of other equipment such as intercooler, regenerator, re-heater, etc., results in the gain in thermal efficiency.

17.6.4 Hydro-Electric Power Plant

As compared with thermal stations an hydro-electric power plant has little or no fuel cost, operating cost. If sufficient water is available to meet peak load demand, then the power can be produced at a small cost. For plants of smaller capacity generally the capital cost per unit installed is higher and the unit cost of conveying water to the power house is more if the quantity of water is small. The added advantage of the hydro-plant is that plant capacity may be increased by increasing the height of dam or by diverting water from other streams into the head reservoir. In order reduce the operating costs further, hydro-power plants may be made automatic or remote controlled.

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17.7 SELECTION OF THE NUMBER AND SIZE OF UNITS

As the load on the power plant is not constant and varies continuously, it is not economical to run a single generating unit meeting the variable load. The efficiency of a single generator unit falls drastically when the load on it is low. Hence, the number and size of the generating units is decided based on the annual load curve of the station. The number and size of the units selected must perfectly match with the station load curve to ensure highest efficiency of the generating units (75% maximum).

Consider the following load curve Figure 17. 23(a). The curve shows wide variations in load for the 24 hr considered from a minimum of 60 kW to a maximum of 500 kW. For this kind of situation, single unit is not recommended. However, the total generating capacity can be matched by selecting a number of generating units to fit the load curve as shown in Figure 17. 23(b). In this case, three units can be employed according to the load on the station, viz. three in numbers.

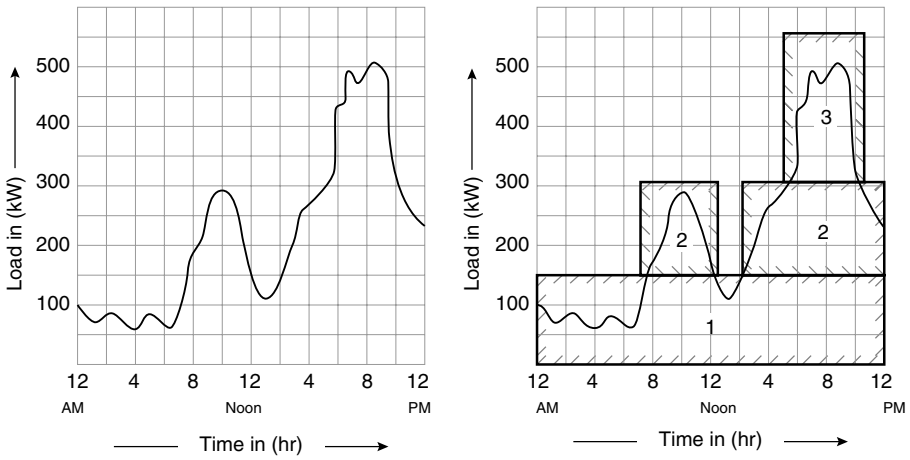


Fig. 17 23 *Selection of Size and Number of Generating Units*

The operating schedule and the sequence for the three units is summarized in Table 17.4.

Table 17.4 *The Operating Schedule and the Sequence for the Three Units*

Time duration	Unit being run
12 midnight–7 AM	• Unit no. 1
7 AM–12 noon	• Unit no. 1 and unit no. 2 both run
12 noon–2 PM	• Only unit no. 1 runs • Unit no. 2 is stopped
2–5 PM	• Unit no. 2 is started • Unit no. 1 and unit no. 2 both run
5–10.30 PM	• All three units: unit no. 1, unit no. 2 and unit no. 3 run
10.30 PM– 12 midnight	• Unit no. 3 is stopped • Unit no. 1 and unit no. 2 both run

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Thus by utilizing the load curve and proper mapping, it is possible to select the size and number of generating units, operating the plant at maximum efficiency.

The following points to be remembered while selecting the size and number of units:

(a) Number and size of the unit should approximately match with annual load curve of the plant.

(b) Selection of units of different capacities is preferable as units having same capacity may not meet the load requirements.

(c) As investment cost/kW of capacity increases with decreased size of the unit, selection of large number of smaller units must be avoided.

(d) A total of 15–20 per cent margin above the maximum demand should be provided to accommodate future load requirements.

(e) Provision for a spare generating unit must be made to facilitate the repairs and maintenance of working units.

17.8 FACTORS AFFECTING ECONOMICS OF GENERATION AND DISTRIBUTION OF POWER

The factors that influence the economics of power plant operation are load factor, demand and utilization factors.

(a) *Load factor:* Depending on the type of power generation and the load factor, cost of generating unit power also varies in a power plant.

(i) In a hydro-electric power station for a given quantity of water and staff to generate maximum output, the cost per unit generated at 100 per cent load factor would be half the cost per unit at 50 per cent load factor. This implies that hydro-electric power station should be run at its maximum load continuously on all units.

(ii) In case of a diesel plant, the cost per unit generated at 100 per cent load factor may be about 3/4th of the same cost at 50 per cent load factor. In other words, a diesel power station should be used for fluctuating loads or as a standby.

(iii) In a steam power station, fuel cost constitutes the major item in operating costs and does not vary in the same proportion as load factor. The cost per unit generated at 100 per cent load factor may be about 2/3rd of the cost per unit generated at 50 percent load factor. Hence, a steam power station should be run in such a way that all its running units are economically loaded.

(b) *Demand factor and utilization factor:* In order to ensure most economical operations of interconnected stations, it is essential to study the curves of all stations and the time of maximum demand occurring in a system.

(i) Unit cost is higher even in an efficient station, if worked at low utilization factor. All planning should be done to load the most efficient and cheapest power-producing stations to the fullest possible extent (base load stations). Typically, a base load station carries full load during 24 hr in three shifts. For stations having medium range efficiency, it is advisable to operate only during the two shifts of 16 hr.

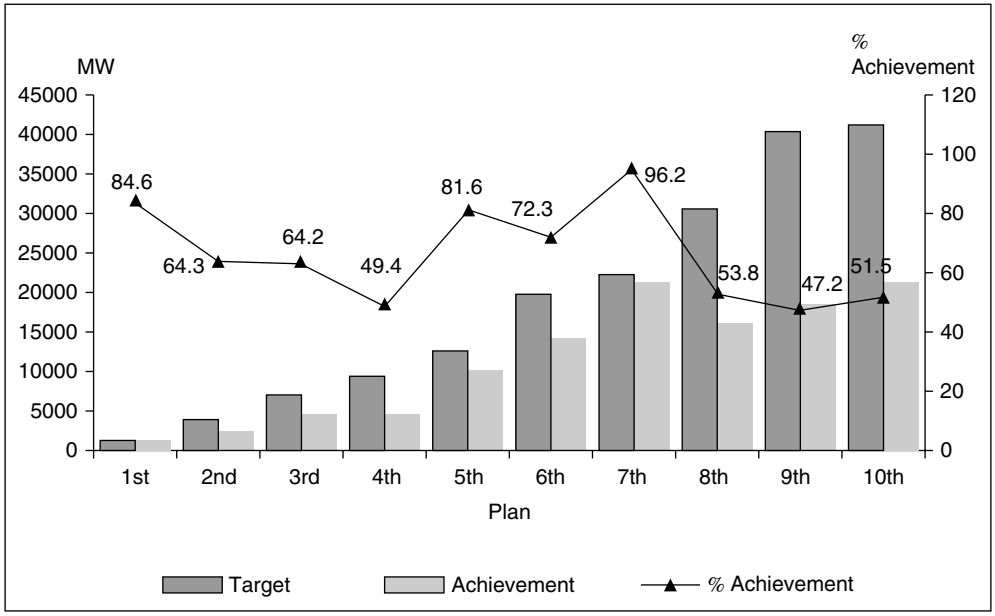
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(ii) The older or less-efficient stations are run as peak load or standby stations for shorter durations.

Modern power plants use units of large capacities that reduce space and to handle larger loads. These units operate near to full load for maximum economical benefit.

17.9 PLANNING FOR POWER GENERATION IN INDIA (SOURCE: KPMG INDIA)

As per the white paper on Implementation Challenges and Opportunities by KPMG in India, India has historically failed to meet its power sector targets by a significant margin and with tremendous opportunities ahead, the power sector continues to be affected by the shortfall both on generation as well as transmission side. The various proposals in generation and transmission are currently under different implementation stages. However, the power sector in India has been hit with a set of problems for meeting the planned targets. An indication of targets and actual additions is provided in the table below:



17.10 POWER ACHIEVEMENT, TARGET V/S ACHIEVED

Based on results of studies, CEA has recommended the following action plans for the Twelfth and Thirteenth Five-Year Plan for the power sector.

Twelfth Plan capacity addition requirement (Source: CEA)

(i) A Capacity addition of about 80,000 MW would be required to be added in the country during the Twelfth Five-Year Plan assuming likely capacity addition of about 62,374 MW

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during the Eleventh Plan, spinning reserve requirement of 5 per cent as per NEP, retirement of old inefficient thermal units of about 4,000 MW and revised reliability criteria of CEA.

(ii) Development of renewable, hydro, nuclear and gas-based projects are given high priority in order to minimize CO_2 emissions. Hydro-capacity addition of 9,204 MW has been considered during Twelfth Plan based on progress of actual construction at project site and taking into account uncertainty in development of hydro projects relating to geological surprises, natural calamities, and environmental issues. The total capacity addition requirement in the country for Twelfth Plan under Base Case Scenario has been proposed by CEA in Table 17.5.

Table 17.5 *The Total Capacity Addition Requirement in the Country for Twelfth Plan*

Total capacity (excluding renewable) – 79,690 MW
Hydro – 9,204 MW
Nuclear – 2,800 MW
Thermal – 67,686 MW
• Coal – 66,600 MW
• Gas – 1,086 MW
• Renewable – 18,500 MW

Indigenous capacity required to be developed – 79,690 MW (1,200 MW imports have been assumed from neighbouring countries).

Thirteenth Plan capacity addition requirement (Source: CEA)

A capacity of 79,200 MW would be required to be added in the country during Thirteenth Plan considering capacity addition of about 80,000 MW during Twelfth Plan. For further reading, readers may visit the following url: http://www.cea.nic.in/

17.11 QUESTIONS

17.11.1 Objective Questions

1. The area under the load curve is
   (a) Units of power generated
   (b) Units of power spent
   (c) Has no meaning
   (d) None of the above
2. The value of diversity factor is
   (a) less than 1
   (b) more than 1
   (c) 0
   (d) negative

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3. Diversity factor is used to determine
   (a) average load
   (b) units of power generated
   (c) plant capacity
   (d) None of the above
4. Area under the load curve over 24 hr gives
   (a) average load
   (b) units of power generated
   (c) plant capacity
   (d) None of the above
5. In a hydro-steam system, steam plant is used as a base load plant during
   (a) high flow season
   (b) low flow season
   (c) any season
   (d) None of the above
6. In a hydro-steam system, steam plant is used as a peak load plant during
   (a) high flow season
   (b) low flow season
   (c) any season
   (d) None of the above
7. Installed capacity of a power station is
   (a) more than maximum demand
   (b) less than maximum demand
   (c) equal to maximum demand
   (d) None of the above
8. More efficient plants are generally used as
   (a) base load plant
   (b) peak load plant
   (c) no such restriction
   (d) None of the above
9. If the load factor on the power station is higher
   (a) Cost/unit power is less
   (b) Cost/unit power is more
   (c) Has no bearing on cost
   (d) None of the above
10. In an interconnected grid system, the diversity factor
   (a) increases
   (b) decreases
   (c) remains constant
   (d) has no effect

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Answers

1. a 2. a 3. c 4. a 5. b 6. a 7. a 8. a 9. a 10. a

17.11.2 Review Questions

1. Differentiate between load curve and load duration curve.
2. Differentiate plant capacity factor and plant use factor.
3. Briefly explain the following:
   (a) Load factor
   (b) Capacity factor
   (c) Use factor
   (d) Diversity factor
   (e) Demand factor
4. Define the following:
   (a) Connected load
   (b) Maximum demand
5. Define the diversity factor and state the advantages of the diversity of load in a power supply system. Show that the load factor of the power is improved by an increase in diversity of load.
6. Discuss the factors to be considered for 'plant selection' for a site.
7. Write short notes on types of tariffs for electrical energy.
8. What is meant by power plant economics? What are the fixed and operating costs?
9. Discuss the methods of determining the depreciation of electrical power plant.
10. What is the significance of two part tariff and three part tariff? Explain the advantages of each over other.
11. What is the significance of incremental rate for a power plant?
12. The loads on a power plant with respect to time for 24 hr are listed below:

Time (hr)	0–6	6–8	8–12	12–14	14–18	18–22	22–24
Load (MW)	40	40	60	50	70	80	40

Draw the load curve and find cut the load factor of the power station. If the loads above 60 MW are taken by stand by unit of 20 MW capacity, find out the load factor of the stand by unit.

Ans: 0.71 and 0.75

13. The daily load of a power station is given below:

Time	12 night	2 AM	6 AM	8 AM	12 noon	12.30 PM	1 PM	5 PM	6 PM	12 night
Load (MW)	20	10	10	40	40	30	40	40	60	20

   (a) Plot load and load duration curve for the system
   (b) Find out the load factor

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14. The following data are collected from a daily load curve of a power plant.

Load in kW	15,000	12,000	10,000	8,000	6,000	4,000	2,000
No of hours at load	875	876	1,750	2,630	4,380	7,000	8,740

(a) Draw the load duration curve and find out the load factor of the system.

(b) If the load is supplied by two plants one is acting as base load plant of 10,000 kW capacity and other as peak load plant of 7,500 kW capacity, find out the annual load factor, annual capacity factor and annual use factor for both plants.

15. A power plant has the following annual factors: load factor = 80 per cent, capacity factor = 60 per cent, use factor = 70 per cent. Maximum demand is 20 MW. Find the following:

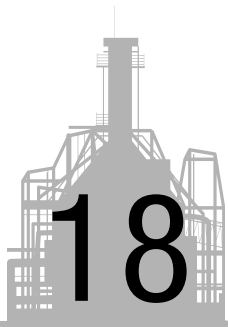
(a) Annual energy production

(b) Reserve capacity over and above peak load

(c) Hours during which the plant is not in service per year

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Environmental Aspects of Power Station



18

Contents

18.1 Environmental aspects
18.2 Global warming and green house effect
18.3 Thermal pollution of water and its control
18.4 Acid precipitation – acid rain and acid snow
18.5 Effluents from power plants and impact on environment
18.6 Environmental impact of power plant: social and economical issues of the power plants

18.1 ENVIRONMENTAL ASPECTS

The design, location, construction and operation of electrical power generation facilities have been affected by environmental concern. Hence, a power engineer must have concern for the environment while producing enough power to meet public demand at low cost. Government has passed environmental legislation that limits the amount of certain solid and gaseous pollutants that can be discharged to the atmosphere. The legislation also limits the amount of thermal energy that can be discharged to natural waters.

Even though power plants are not alone the largest contributors to environmental problems, the growing number of plants is of serious concern about the same.

18.1.1 Constituents of the Atmosphere

Major constituents of atmosphere are nitrogen and oxygen with traces of argon, carbon dioxide and helium. The atmosphere can be subdivided into a clean dry air, water vapour, and aerosols, namely water droplets, as well as ice crystals, salt grains and dust particles, which help in condensation of water.

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Table 18.1 gives the composition of the composition of clean dry air in the troposphere (i.e. at altitudes ≤ 16 km). Altogether, N_2, O_2 and Ar represent 99.96 per cent in volume. The concentration of CO_2 at ground level may vary between day and night by up to a factor of two, and all other composition of gases is homogeneous and constant. In addition, there are also traces of SO_2, O_3, NO and CO.

Table 18.1 *Constituents of Atmosphere*

Name	Molecular mass (amu)	% by volume	Name	Molecular mass (amu)	% by volume
N_2	28.013	78.084	He	4.003	0.0005
O_2	32.000	20.946	Kr	83.8	0.0001
Ar	39.948	0.934	CH_4	16.043	0.0002
CO_2	44.010	0.033	H_2	2.016	0.00005
Ne	20.183	0.0018	N_2O	44.013	0.00005

Figure 18.1 indicates the four layers of the earth's atmosphere. From the earth's surface, it is the troposphere that ranges from 0 to 8 miles – the next being the stratosphere that ranges from 8 to 30 miles. Next to stratosphere lies the mesosphere that ranges from 30 to 50 miles. The outermost layer is the thermosphere that ranges from 50 to 120 miles.

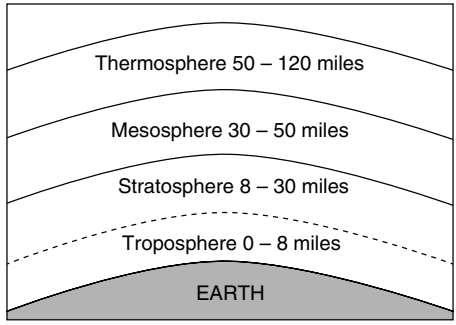


Fig. 18.1 *Layers of Atmosphere*

18.1.2 Different Pollutants Due to Thermal Power Plant and Their Effect on Human Health

The most commonly observed pollutants from fossil power plants (oil or coal based) are the following:

(a) Sulphur oxide (SO_x)
(b) Nitrogen oxides (NO_x)
(c) Carbon oxides (CO_x)
(d) Particulate matter or PM
(e) Thermal pollution

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Coal-fired power plants directly emit particulate matter (PM, also called "soot") in addition to gases that undergo chemical reactions to form fine particles in the atmosphere: SO_x and NO_x. These emissions of PM in conjunction with SO_x and NO_x increase the ambient concentration of PM less than 2.5 μm in diameter ($PM_{2.5}$) over hundreds to thousands of kilometres downwind of the plants. Continuous exposure to $PM_{2.5}$ may lead to increased mortality from cardiopulmonary diseases, lung cancer [1,2] and numerous other respiratory illnesses and associated morbidity.[3]

Although most modern power plants have some modern pollution controls, such as electrostatic precipitators (ESPs), the use of flue–gas desulfurization (FGD) is relatively rare in the developing countries. FGDs if installed can reduce sulphur dioxide emissions by 90 per cent, resulting in substantial human health risk reductions.

18.1.3 Environmental Control of Different Pollutants Such as Particulate Matter PM

PM comprises smoke, dust and other solids made of a wide variety of organics and metals. Both natural and human activities are responsible for PM in the atmosphere.

Natural causes include natural dust caused by wind, storms, volcanoes and natural fires, meteoritic dust and fog. Fog when combined with smoke forms smog. Human causes include dust and ash emanating from large industries, fossil-based power plants and other combustion processes. Out of these, power plants contribute to about one-third. Sulphur compounds are larger contributors to PM.

Effects

(a) Decreases visibility, increases soiling and corrosion and affects climatic conditions.

(b) Causes health hazard, which causes breathing problem with particulate sizes ranging between 2 and 1.25 μm. An individual breathes about 1 mg of particulate matter/day during times of heavy pollution. These particles deposit in the mucous lining and deeper parts of lungs.

(c) Particulate matter in the atmosphere is highly toxic; it absorbs toxic substances and obstructs respiratory passages. An annual mean of 100–200 $\mu g/m_3$ results in respiratory illness, whereas 300–600 $\mu g/m^3$ causes a large increase in the number of patients suffering from bronchitis.

18.1.4 Oxides of Sulphur

Sulphur in the atmosphere exists essentially in the following three forms:

1. Sulphur dioxide, SO_2 that comes from artificial causes.
2. Hydrogen sulphide, H_2S that comes from natural sources.
3. Various sulphates that come from sea spray and oxidation of SO_2.

- *Sulphur dioxide*

Since it comes from artificial causes, it is of primary concern even though it contributes less than 25 per cent of all sulphur in the atmosphere. It is estimated that more than 90 million tons of sulphur dioxide are discharged to the atmosphere every year.

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Effects

Acid rain and acid snow are caused by SO_2, hydrogen sulphide, H_2S and the oxides of nitrogen, NOx in the atmosphere. These gases carried along wind currents combine with water molecules in the water vapour of the atmosphere, forming tiny drops. These drops mainly contain nitric acid, HNO_3 and sulphuric acid, H_2SO_4. These drops when encountered by snow or rain-producing clouds result in acid rain which:

(a) Results in lowering of pH values of lakes and streams.

(b) Results in leaching of essential plant nutrients from soil and reduce nitrogen fixation by microorganism. This makes soil less fertile.

(c) Dissolve aluminium and cadmium out of soil minerals, allowing them to enter roots and kill trees.

18.1.5 Oxides of Nitrogen

Production of NOx is an endothermic reaction and its concentration is temperature dependent. NOx emissions can be reduced by lowering the combustion temperature and by eliminating hot spots in the furnace. It can also be inhibited by lowering the air–fuel ratio or by employing exhaust gas re-circulation. Lowering the air–fuel ratio restricts the amount of oxygen available for combustion while exhaust gas recirculation reduces the combustion chamber temperature.

18.1.6 Oxides of Carbon

Carbon monoxide, CO, methane, CH_4 and carbon dioxide, CO_2 are the most widely used contaminants.

Carbon monoxide is caused by natural causes partially, such as marsh gas, coal mines, vegetation and forest fires. This contribution is small when compared to human generated causes. A total of 90 per cent of CO is produced by motor vehicles, whereas power plants contribute less than 1 per cent. The CO additions to the entire earth's atmosphere are about 230 million tons/year.

Effects

(i) Like NO, it deprives body tissues from oxygen by absorbing haemoglobin.

(ii) A CO concentration of 100 ppm causes headache, 500 ppm causes collapse and 1,000 ppm is fatal.

Carbon dioxide

It is largely contributed by power plants. CO_2 added to nature by natural causes such as decay of organic matter contributes more than that by artificial causes. In addition, CO_2 contaminates the atmosphere; it is not considered as a pollutant as it is essential for plants.

CO_2 can be removed by photosynthesis. This process converts CO_2 back into oxygen and organic compounds in the presence of sunlight and water. However, more concentration of CO_2 results in green house effect.

18.2 GLOBAL WARMING AND GREEN HOUSE EFFECT

The concentration of CO_2 has increased to a level that is beyond the capacity of plant life and the oceans to completely remove it. 50 per cent of CO_2 added is retained in the atmosphere.

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The existence of CO_2 in the atmosphere causes green house effect. The atmosphere, analogous to the glass panes of a green house, transmits the radiation from sun. The surface of the sun is about 6,000 K and most of the radiation is in the form of short waves and visible portions of the spectrum, and only a portion of this radiation is absorbed or scattered back to space by the atmosphere. The transmitted radiation is largely absorbed by the surface of earth and thereby warming it.

Part of the resulting heat of the earth is transmitted by conduction, convection, evaporation and reradiated away from the surface. Because of the low surface temperature, this radiation is by infrared portion of the spectrum. The presence of CO_2 and H_2O in the atmosphere results in the absorption of large portions of *infrared* radiations (long wave) from the surface and partial of those radiations, back to earth. Thus, atmosphere is not transparent completely to the reradiated energy and traps much of the energy from the sun. Thus, growing levels of concentrations of CO_2 increases the surface temperature of the earth. This may result in climatic changes with disastrous consequences, such as melting of polar ice caps, raising the sea level and flooding the coastal areas of the world (Figure 18.2).

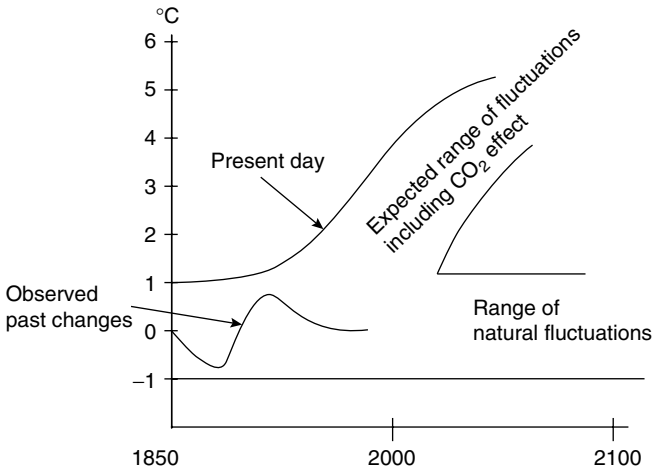


Fig. 18.2 *Change in Global Temperature with and without CO_2 Effect*

To counterbalance the effect of CO_2, PM in the atmosphere should be increased. PM helps in reflecting some of incidents from the sun back to space and thus prevents it from reaching the earth's surface.

18.3 THERMAL POLLUTION OF WATER AND ITS CONTROL

The discharge of thermal energy into natural waters is called thermal pollution. The addition of heat to the water reduces the water's ability to hold dissolved gases, including dissolved oxygen that is vital for aquatic life. If the water temperature exceeds 35°C, the dissolved oxygen contents are low to support life. At lower temperature, aquatic growth is enhanced and most plants and fishes grow at a faster rate. During cold weather, warm water discharge from a power plant is an excellent place for fishes.

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To compare power generating systems with respect to the amount of thermal energy discharged to the environment, a term known as thermal discharge index (TDI) is introduced. The TDI of any power system is the number of thermal energy units discharged to the environment for each unit of electrical energy produced by the plant, that is

$$\text{TDI} = \frac{\text{Thermal power to the environment, MW}_{\text{th}}}{\text{Electrical power output, MV}_{\text{e}}}$$

TDI should be as low as possible and preferably non-zero to satisfy second law of thermodynamics.

Let P_{th} = input thermal energy to power station

Pth η_{th} = electrical power output of the system

then

$$\text{TDI} = \frac{P_{\text{th}} \quad P_{\text{th}} \quad \text{th}}{P_{\text{th}} \quad \text{th}}$$

$$= \frac{P_{\text{th}}\left(1-\eta_{\text{th}}\right)}{\eta_{\text{th}} \times P_{\text{th}}}$$

$$= \frac{\left(1-\eta_{\text{th}}\right)}{\eta_{\text{th}} \times P_{\text{th}}}$$

In a nuclear reactor, thermal energy discharge is by condenser, using large volumes of water. For gas turbine and IC engine power plants, thermal energy is discharged to the atmosphere. For a fossil-fuelled power system, thermal discharge can be divided into the following three groups:

1. Electrical power output, P_{e}
2. Stack power, $P_{\text{s}} = P_{\text{th}}\,(1 - \eta_{\text{b}})$; η_{b} = boiler efficiency
3. Condenser power, Pc

Therefore,

total power $P_{\text{th}} = P_{\text{e}} + P_{\text{s}} + P_{\text{c}}$

$= \eta_{\text{th}}\, P_{\text{th}} + (1 - \eta_{\text{b}})P_{\text{th}} + P_{\text{c}}$

Therefore, $P_{\text{c}} = P_{\text{th}}\,[1 - \eta_{\text{th}} - (1 - \eta_{\text{b}})]$

$= P_{\text{th}}\,[\eta_{\text{b}} - \eta_{\text{th}}]$

18.4 ACID PRECIPITATION – ACID RAIN AND ACID SNOW

The tiny droplets of water in the clouds have the tendency to continuously absorb the suspended solid particles and gases in the atmosphere. The gases of sulphur oxides and nitrogen oxides while coming in contact with water molecules are converted into sulphuric and nitric acids as per the following chemical reactions. The non-metal oxide gases react with water and produce acids.

$SO_2 + H_2O \rightarrow H_2SO_3$

$2\,NO_2 + H_2O \rightarrow HNO_2 + HNO_3$

When large numbers of such tiny cloud droplets coalesce together, they form a larger water drop that falls in the form of precipitation: acid rain, acid snow, ice, sleet or fog. The pH of

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acid rain or snow varies from 3.6 to 5, whereas normally accepted pH of rain water is 5.6. Figure 18.3 shows the process of acid rain formation.

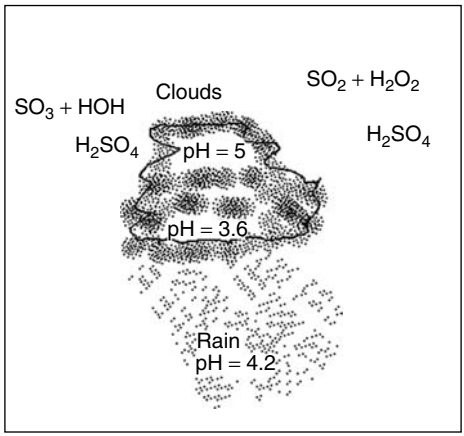


Fig. 18.3 *The Process of Acid Rain Formation*

18.5 EFFLUENTS FROM POWER PLANTS AND IMPACT ON ENVIRONMENT

18.5.1 Radiation from Nuclear Power Plant Effluents

Effects of nuclear radiation

Biological damage by interaction of radiation and tissue:

(i) Ionization – λ-, β-, γ-radiations ionize tissues into which they penetrate resulting in complete damage of tissue.

(ii) Displacement – atom in the tissue is displaced by neutron and γ-radiation.

(iii) Absorption – absorption of neutron by a tissue nucleus leads to radioactive nucleus, which results in change in chemical nature and mal-functioning of cell. Due to this, cell gets damaged, leading to genetic modification.

Inhalation of radioactive material through air, food and water result in radiation hazard.

18.5.1.1 Handling Nuclear Waste and Safety Measures

Wastes associated with nuclear power are as follows:

(i) *Gaseous effluents*

Under normal operation, these are released slowly from the power plants into the biosphere and become diluted and dispersed harmlessly.

(ii) *Uranium mine and mill tailings*

Tailings are residues from uranium mining and milling operations. They contain low concentration of naturally occurring radioactive materials. They are generated in large volumes and are stored at the mine or mill sites.

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(iii) *Low level wastes*

Low-level waste (LLW) includes items that have become contaminated with radioactive material or have become radioactive through exposure to neutron radiation. This waste typically consists of contaminated protective shoe covers and clothing, wiping rags, mops, filters, reactor water treatment residues, equipments and tools, luminous dials, medical tubes, swabs, injection needles, syringes, laboratory animal carcasses and tissues, etc. These contain less than 10 nCi per gram of trans-uranium contaminants containing low, but potentially hazardous, concentrations of radioactive materials. These are generated in almost all activities (power generation, medical, industrial, etc.) involving radioactive materials, require little or no shielding. These are disposed of in liquid form by shallow land burial.

(iv) *High-level wastes*

High-level wastes (HLWs) are generated in reprocessing of spent fuel. They contain all fission products and contain most of the trans-uranium elements not separated during reprocessing. Such wastes are to be disposed off carefully. Since the only way for radioactive waste finally to become harmless is through decay, which for HLWs can take hundreds of thousands of years; the wastes must be stored and finally disposed off in such a way that it provides adequate protection of the public.

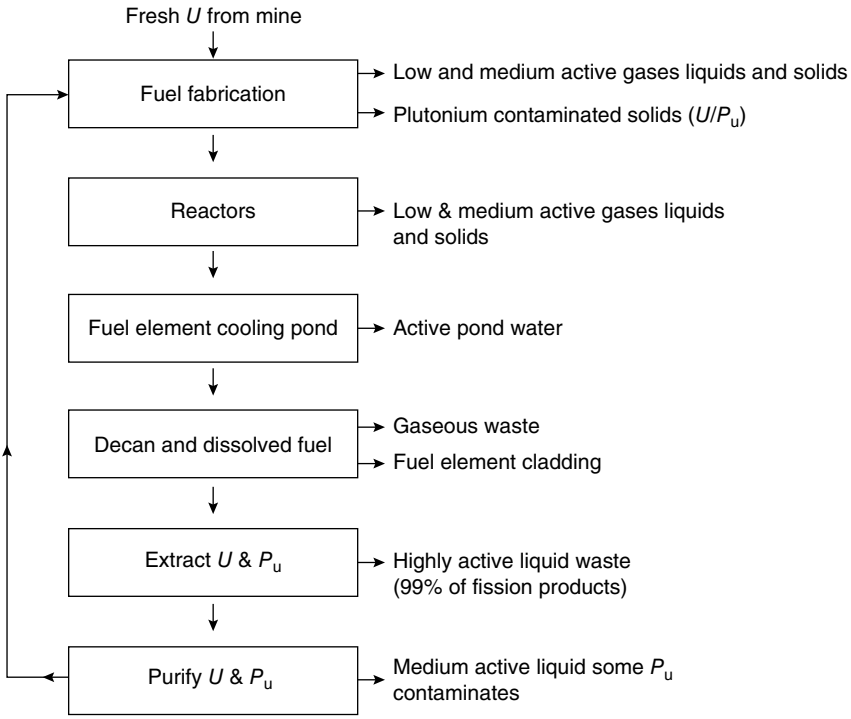


Fig. 18.4 *A Typical Fuel Cycle*

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(v) *Spent fuel*

This is unprocessed spent fuel that is removed from the reactor core after reaching its end-of-life core service. It is removed and then stored for 3–4 months under water in the plant site to give time for the most intense radioactive isotopes to decay before shipment for reprocessing or disposal.

Note: Units of nuclear radiation – Roentgen: amount of radiation that will, on passing through pure air under standard condition, produce 1 electrostatic unit of ions/cm_3 of air $\Rightarrow$ 86.9 ergs of energy absorbed/gm of air, Rem – dose of absorbed radiation that will have the same effect that exposure to 1 Roentgen of γ-radiation will have $\Rightarrow$ 100 ergs/gm of tissue, total dose driving a person's life time < 200 Rem weekly dose < 0.3 Rem.

Figure 18.4 shows a typical fuel cycle generating these wastes.

18.5.1.3 Radioactive Waste Disposal

One of the major problems in the nuclear plants is the disposal of waste products that are highly radioactive. They emit large quantities of γ-rays, and these high energy γ-rays destroy all living matter through which they pass.

The radioactive products of 400 MW power station would be equivalent to 100 tons of radium daily, and the radioactive effect of this plant products if exposed to atmosphere would kill all the living organisms within an area about 100^2 miles.

The disposal of nuclear waste is the great problem for the engineers and scientists.

In a nuclear fuel cycle, the solid, liquid and gaseous radioactive wastes are produced at different stages. These radioactive wastes must be disposed off in such a manner that these do not cause hazard to human and plant life. Moderate active solid wastes are buried in the ground. Moderate liquid wastes after preliminary treatments are discharged in deep pits or day well from which they sweep out into the surrounding ground. Active liquids are kept in concrete tanks, and these tanks are buried in the ground until their decay of radioactivity. Many times, the radio activity increases the temperature of the liquid waste or sometimes these liquids boil and the activity decreases with time. Gaseous wastes are discharged to atmosphere through high stacks if the wind permits.

The waste is disposed to air, ground and ocean.

(a) *Air*

There are lots of problems in disposing radioactive gases in to the air. Because strong radioactive gases such as strontium and iodine are absorbed by the plants and they enter in to the human body through food. Cesium is absorbed in muscle and strontium in bones and paralyses the health. Generally, radioactive gases are collected and stored in a tank buried in the ground and disposed off to the atmosphere when activity level is sufficiently low.

The amount of radioactivity presently disposed to the air is well below the harmful level, but the problem will get serious when large number of power reactors will come up in operation.

(b) *Ground*

This is one of the easy and cheapest methods of disposal because soil absorbs radioactive material easily. This disposal is suitable mostly in areas of low rainfall at points that are high above the ground water level.

Most of the radio activeness of waste is removed just by storage. The storage problem is simplified by separating cesium and strontium that are extremely radioactive. These are generally stored in tanks that are buried in ground and then disposed in to the sea after 13 years of storage.

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Vacated coal mines are used for waste disposal. The wastes are disposed in the salt heaps provided in the mines, because salt is a powerful absorber of radioactive emissions. It is the easy and more economical method of disposing off liquid waste by freezing.

18.5.2 Pollution Standards

As per the Environment (Protection) Rules, 1986 of the Government of India, with regard to PM emissions, the following standards should be adhered to by the thermal power plants (Table 18.2).

Table 18.2 *Government of India Standard for PM Emissions*

Industry	Parameter	Standard
	PM emissions	
	• Generating capacity 210 MW or more	150 mg/normal m^3
	• Generating capacity 210 MW or less	350 mg/normal m^3
	Condenser cooling waters (once through cooling system)	– 6.5–8.5 pH
	• Temperature	–5°C higher than the intake water temperature
	• Free available chlorine	–0.5
Thermal power plant	Boiler blowdowns	–
	• Suspended solids	100
	• Oil and grease	20
	• Copper (total)	1
	• Iron (total)	1
	Cooling tower blowdowns	–
	• Free available chlorine	0.5
	• Zinc	1
	• Chromium (total)	0.2
	• Phosphate	5
	• Other corrosion inhibiting material	Limit to be established on a case-by-case basis by the Central Board in case of Union Territories and State Board in the case of states
	Ash pond effluent	–
	• pH	6.5–8.5
	• Suspended solids	100
	• Oil and grease	20

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As per the study conducted by Green Peace India, India does not impose technology-based control for SO_2 and NO_x, and India's ambient air quality standards have not been used to impose such controls. With unregulated pollutants such as SO_2 and NO_x, emission rates can vary widely depending on the sulphur and nitrogen content of the coal being used and the operating temperature of the boiler. Table 18.3 extract from Green Peace India[4] shows technology-based pollution limits for several countries.

Table 18.3 *Technology-Based Pollution Limits for Selected Countries*

Country	PM (mg/m^3)	SO2 (mg/m^3)	NO_x (mg/m^3)
India	150	None	None
Japan (general)	50	Permit	200
China	30	100	100
EU LCP[1] Directive[6]	30	200	200
EU – IE[2] Directive	10	150	150
EU – BAT[3]	5	20	50
United States – NSPS/ NESHAP[4]	14.5	100	110
United States (BACT)[5]	14.5	22	70

Note: [1]Large Combustion Plant; [2]Industrial Emissions; [3]Best Available Technique; [4]New Source Performance Standards; [5]Best Available Control Technology.

Table 18.4 gives the National Ambient Air Quality Standards as prescribed by the Central Pollution Control Board of India, under section 16 (2) (h) of the Air (Prevention and Control of Pollution) Act, 1981 (14 of 1981).

Table 18.4 *The National Ambient Air Quality Standards*

Pollutant	Time-weighted average	Concentration in ambient air			Method of measurement
–	–	Industrial area	Residential. rural and other areas,	Sensitive area	–
Sulphur dioxide (SO_2)	Annual average	80 µg/m³	60 µg/m ³	15 µg/m³	1. Improved west and Gacke method
–	24 hours**	120 µg/m³	80 µg/m³	30 µg/m³	2. Ultraviolet fluorescence
Oxides of nitrogen as NO_2	Annual average*	80 µg/m³	60 µg/m³	15 µg/m³	1. Jacob & Hochheiser modified (Na-Arsenite) method

(*Continued*)

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Table 18.4 (*Continued*)

Pollutant	Time-weighted average	Concentration in ambient air			Method of measurement
–	24 hours**	120 μg/m³	80 μg/m³	30 μg/m³	2. Gas phase chemiluminescence
Suspended particulate matter (SPM)	Annual average*	360 μg/m³	140 μg/m³	70 μg/m³	(Average flow rate not less than 1.1 m ³/ minute)
–	24 hours**	500 μg/m³	200 μg/m³	100 μg/ m³	–
Respirable particulate matter (size less than10 μm)(RPM)	Annual average*	120 μg/m³	60 μg/m³	50 μg/m³	–
–	24 hours**	150 μg/m³	100 μg/m³	75 μg/m³	–
Lead (Pb)	Annual average*	1.0 μg/m³	0.75 μg/m³	0.50 μg/ m³	–
–	24 hours**	1.5 μg/m³	1.00 μg/m³	0.75 μg/ m³	AAS method after sampling using EPM 2000 or equivalent filter paper
Carbon monoxide (CO)	8 hours	5.0 μg/m³	2.0 μg/m³	1.0 μg/ m³	–
–	1 hour	10.0 μg/m³	4.0 μg/m³	2.0 μg/ m³	Non-dispersive infrared spectroscopy

*Annual arithmetic mean of minimum 104 measurements in a year taken twice a week 24 hourly at uniform interval.

**24 hourly/8 hourly values should be met 98 per cent of the time in a year. However, 2 per cent of the time, it may exceed but not on two consecutive days.

18.5.3 Methods of Pollution Control

With stringent air quality restriction by the governments, majority of the power plants in the world use ESPs for dust removal and FGD technology for sulphur removal. An ESP can handle large volumes of flue gases accompanied low pressure drop and high collection efficiency (99.5%). It facilitates easy removal of dust particles.

18.5.3.1 Electrostatic Precipitator

Figure 18.5 shows a general arrangement of an ESP. It consists of two sets of electrodes the *emitting or discharging electrode* and *collecting electrode*. In the case of a *tubular-type*

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precipitator emitting electrodes are placed in the centre of the pipe; whereas in the case of *plate-type* precipitator, emitting electrodes are placed midway between the two plates. The emitting electrodes are connected to negative polarity of high voltage (20–100 kV) DC source. The collecting electrodes are connected to the positive polarity of the source and earthed.

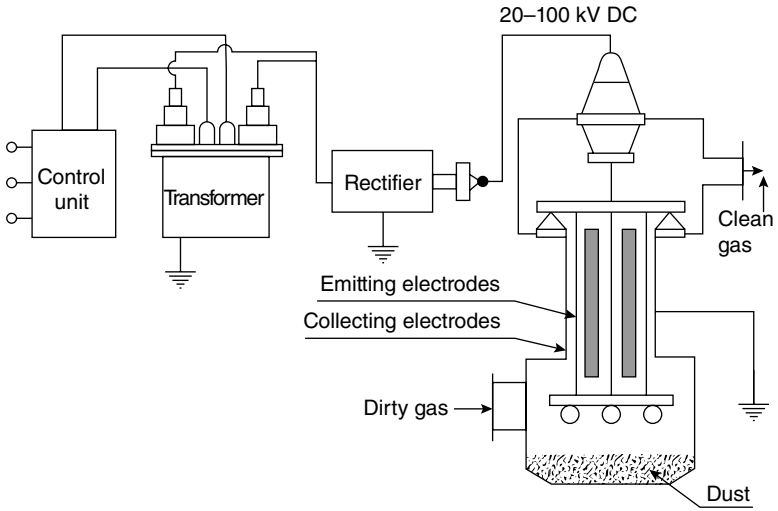


Fig. 18.5 *A General Arrangement of an ESP*

When high voltage is applied, it generates a unidirectional non-uniform electric field having greater magnitude at the discharge electrodes. This results in a blue luminous glow, called a *corona* around them. This corona is an indication of negatively charged ionized gas molecules that travel from discharge electrodes to grounded collection electrodes. The dust particles thus get deposited on the collector electrodes and lose their charge. Remaining dust particles cling to the electrode surface due to electrical resistivity, are removed by rapping the electrodes using rapping motors. The detailed working principle of an ESP is already explained in chapter Section 3.12.2.

18.5.3.2 FGD Technology

FGD is a technology that extracts sulphur dioxides from flue gases produced in coal-based thermal power plants, where sulphur content in coal is more than 0.5 per cent. The coal produced from Indian mines contains only 0.4 per cent sulphur contents; therefore in India, this technology was not required much. Now, a lot of companies in India are importing coal from other countries such as Indonesia, South Africa, which contains sulphur contents 0.6–0.9 per cent. Therefore, it is made mandatory to install the FGD plant to maintain the ambient air quality standards.

Sulphur dioxide is extracted from flue gases in wet scrubber, slurry of alkaline sorbent; lime stone reacts with the sulphur dioxide. More than 90 per cent of the limestone particles in the

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limestone powder (85% purity) are made to pass through a screen (325 mesh) to ensure that maximum amount of limestone particles comes in contact with the sulphur dioxide molecules in the flue gas.

To have a chemical reaction, the limestone powder is mixed into 15–30 per cent slurry introduced into the FGD vessel, re-circulated, and sprayed into the flue gas stream. The following chemical reactions happen in the FGD system:

- Flue gases containing SO_2 enter the absorber and come in contact with limestone slurry ($CaCO_3$) in the wet scrubber producing calcium sulphite.

$$CaCO_3 + SO_2 \rightarrow CaSO_3 + CO_2$$

- Further, $Ca(OH)_2$ in lime, when combined with SO_2 gas produces calcium sulphite and water

$$Ca\,(OH)_2 + SO_2 \rightarrow CaSO_3 + H_2O$$

- Calcium sulphite is further oxidized by forced oxidation utilizing blowers to produce marketable $CaSO_4\ 2H_2O$, popularly known as gypsum.

$$CaSO_3 + H_2O + \tfrac{1}{2}O_2 \rightarrow CaSO_4.2H_2O$$

18.5.3.3 Handling Fly Ash and Bottom Ash of a Thermal Power Plant

With pulverized coal, all the ash particles are formed in suspension and tend to remain in suspension in the furnace. In slag-type furnace, a portion of ash (20–30%) is drained to the furnace bottom in a molten state. The portion of ash that tends to escape towards the stack is called *flyash* and a portion that is collected at the bottom of the furnace and is removed either in dry or wet state is called *bottom ash*.

A 500 MW power plant boiler consumes 300 tons of coal per hour and produces 120 tons of total ash per hour in India. Out of this, 20 per cent (24 tons/hr) comes out in the form of clinkers and remaining 80 per cent (96 tons/hr) is carried in the gas stream and is collected in hoppers beneath economizer, air preheater, mechanical collector and electronic precipitator. The chemical composition of ash varies as per the source of coal and its origin.

The broad composition of ash is given below in Table 18.5:

Table 18.5 *Composition of Ash*

	Content in ash (%)	Content in ash (%)	Content in ash (%)
Silica (SiO_2)	50	38	48
Alumina (Al_2O_3)	40	20	26
Iron oxide (Fe_3O_3)	16	6	10
Calcium (CaO)	10	2	4
Magnesium	3.5	1.0	2.0
Sulphate (SO_3)	2.5	0.5	1.2
Alkalies (Na_2O, K_2O)	5.5	2.0	4.5

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Uses of ash and dust

Coal source, quality, degree of grinding, combustion characteristics and boiler design are some of the factors that affect chemical and physical quality of the ash and its potential use. Combustion conditions as percentage of air and furnace temperature are important for the ash quality as they influence the degree of oxidation of the metals. Ash quality is also affected by the use of additives to control boiler corrosion or to aid in collecting ash in ESP.

Ash is widely used in the production of cement. Fly ash is pozzolanic and develops self-hardening characteristics. Concrete made of ash can yield improved workability and strength greater than all cement-based concrete. The pozzolanic quality of ash–lime mixture creates a healing of cracks in the pavement themselves.

Due to their better alkali values, they are used for treating acidic soils. Its use for agricultural purposes is recently experimented in Japan, and it is found that it supplies essential nutrients as sulphur, boron, calcium and zinc. It also adjusts pH to optimum levels for plant growth. It is also found that if the ash in used in limited quantity in soil, it increases the yield of corn, turnip, white clover and asfalfa.

The use of ash in the production of concrete is technically established. 20 per cent of fly ash and 30 per cent bottom ash are presently used constructively in the United States.

Utilization of fly ash

The disposal of ash using low-grade coal (30–50% ash content) in thermal power plants in India has created lot of problems as large land area and huge quantity of water are required for the same. The fly ash generation 1998 was around 60 million tons per year from 75 coal-fired thermal plants that had increased to 90 million tons per year by 2,000 and 110 million tons in 2010.

About 20 per cent of coal ash in the coal is converted into bottom ash and 80 per cent fly ash. India has a current annual generation of approximately 112 million tons (2011 data) and its proven suitability for variety of applications as admixture in cement/concrete/mortar, lime pozzolana mixture (bricks/blocks), etc. Cement and concrete industry accounts for 50 per cent fly ash utilization; the total utilization of which at present stands at 30MT (28%). The other areas of application are low-lying area fill (17%), roads and embankments (15%), dyke raising (4%) and brick manufacturing (2%); other new areas for safe disposal of fly ash is in paint industry, agriculture, etc.[4] Government of India is not fully prepared to appreciate the magnitude of the fly ash problem. Disposal of fly ash is extremely water- and land-intensive, leading to diversion of fertile land and unsustainable water usage. It also causes displacement of people since large tracts of land are acquired for fly ash disposal, besides leading to air, water and soil contamination. Disposal of fly ash is equally costly as 2 per cent of the cost of the plant goes into disposal of fly ash from the site of generation to the disposal site. It addition to this, the water required to remove and transport the fly ash in the form of slurry is considerably high. Approximately 1 m^3/hr is required per MW generating capacity.

Fly ash can be used for making a variety of building products as discussed below.[5]

(a) *Mixing with cement:* 10–25 per cent dry fly ash can be used as during manufacture of cement or blended with finished portland cement to produce portland pozzolana cement whose strength is higher. The major drawback of this cement is its high cost, and it requires more setting time.

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(b) *Sintered light-weight aggregates:* This is produced by pelletization or nodulization of fly ash and sintering them at 1,000–1,300°C. Unburnt fuel in the fly ash modules supports ignition. Sintered weight aggregate substitutes chips in concrete reducing dead weight. The process know-how developed by CBRI, Roorkee has yet to go in for commercial production.

(c) *Fly ash–stone powder–cement bricks:* These types of bricks are manufactured by mixing weighted fly ash, cement and stone powder in a mixture and moulded and pressed in brick making machine. The compacted bricks are water cured for 28 days. The compressive strength of these bricks varies from 70 to 110 bar depending on the cement content.

CFRI-Dhanbad has developed a process to manufacture bricks using dry fly ash. This process is chemically bounded. The brick material contains 83 per cent fly ash, 10 per cent sand, 7 per cent limestone and 0.2 per cent accelerator. The mixture of the above constituents is subjected to formation as per required shape in the mould and then naturally cured for 2–3 days. These bricks are then steam cured at 2–5 bar pressure for giving the strength. The brick can bear 125 bar compressive load.

(d) *Cast-in-situ fly ash walls:* Using high fly ash mix comprising of cement, lime fly ash and sand in appropriate proportions depending on the quality of fly ash with pre-measured water cement ratio, cast-in-situ can be built. These can be cast to any thickness using steel shuttering. By using this system, we can achieve 20 per cent economy, quicker construction, good finish on both sides of wall and more carpet area.

(e) *Cellular light-weight concrete:* This can be manufactured by a process involving the mixing of fly ash, cement, coarses and fines and a foaming agent. The slurry formed is poured in moulds and allowed to set. The blocks are then removed and are cured by spraying water on the stack. The bulk density of the products varies from 400 to 1,800 kg/m^3 against ordinary brick bulk density of 1,600–1,920 kg/m^3. The blocks are especially useful in high rise construction reducing the dead weight of the structure. DLF Universal Ltd., New Delhi, is using these blocks in their construction projects for the last 8 years. In agriculture: It improves permeability status 1, fertility status (soil health)/crop yield of soil and soil texture.

Fly ash reduces bulk density of soil, improves water holding capacity/porosity and optimizes pH value and soil aeration. It provides micro-nutrients such as Fe, Zn, Cu, Mo, B, Mn, etc., and macro-nutrients such as K, P, Ca, Mg, S, etc.[4] It is reported that the agricultural increase of grains is around 15 per cent, green vegetables 35 per cent and root vegetables 50 per cent, when fly ash is mixed with soil.[5]

18.6 ENVIRONMENTAL IMPACT OF POWER PLANT: SOCIAL AND ECONOMICAL ISSUES OF THE POWER PLANTS

A power plant can affect the environment by its construction and its operation with could be either temporary or permanent. Some of the environmental impacts of power plant with strong connection to social and economic issues could be as follows.

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18.6.1 Land and Air Space

(a) A power plant and its auxiliary components such as natural gas pipelines, water lines coal-handling and storage systems, transmission lines and waste disposal sites occupy considerable space on the ground and in the air, emitting pollutants into the ground and air. This may eliminate opportunities for others to purchase or use the land. It may also affect the existing or future uses of adjoining and nearby land parcels.

(b) The plant's height viz. tall buildings, chimney and towers may result in safety concerns for aircraft or visual impacts for local landowners.

(c) If the land to be used for the power plant is agri-based, growing mostly vegetation (crops, pasture, or old-field vegetation), a power plant could pose serious threat on land use, soils and wildlife.

18.6.2 Water

Fossil fuel-fired and biomass-fired plants burn fuels to generate power. The burning of fuel creates exhaust gases and other by-products, including air pollutants. In addition, large amount of cooling water requirements in these plants is met from nearby rivers or lakes, or from local underground water. The effluents and used water discharged back to land or river, the concentration of pollutants in the water and ground are bound to increase.

18.6.3 Solid Waste

Power plants produce a variety of solid wastes such as ash in thermal plants spent nuclear fuel rods and low-level radioactive wastes in nuclear plants. The filtered solids from the effluent treatment plants are a by-product that must be disposed appropriately.

18.6.4 Construction and Operation

Construction and operation of a power plant can have unsettling effects on the community in which the power plant is built. Construction of the power plant could create nuisance to the surrounding landowners and other citizens. This in turn may affect community aesthetics or business. Costs for community services such as police, fire protection, emergency medical service, schools, hospitals and traffic control can increase. Additional demand requirements might be placed on the municipal water supply or wastewater treatment capacity, or on solid-waste management systems.

Construction of new transportation facilities such as rail, road or barge may make road or rail traffic complicated or burdened by delivery of materials, particularly large items.

Water and natural gas are transported by using pipelines that often require the digging of trenches that may impact residential yards, roads, farm fields, forest or agriculture lands.

18.6.5 Noise

Noise and vibration levels in neighbourhoods might increase during construction and power plant operation. The cooling towers of an operating power plant can also create fog and rime ice. Air space issues and compatibility with local land use must be considered in the light of the space the power plant occupies and the way in which it operates.

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18.6.6 Economical Aspects

The commissioning of power plants can create positive effects on the community such as employment for local residents and purchases of locally produced goods and services creating additional income to the localities. Local tax revenue or state shared revenue for the local governing bodies such as municipalities would also increase.

The electricity produced by the plant could be utilized for more reliable power supply to the local residents by stabilized the local electric transmission grid.

18.7 QUESTIONS

18.7.1 Objective Questions

1. As per the government of India, regulations limitation of particulate material in thermal power plants below 210 MW generating capacity is
   (a) 350 mg/normal m^3
   (b) 150 mg/normal m^3
   (c) 250 mg/normal m^3
   (d) 100 mg/normal m^3
2. As per the government of India, regulations limitation of particulate material in thermal power plants more than 210 MW generating capacity is
   (a) 350 mg/normal m^3
   (b) 150 mg/normal m^3
   (c) 250 mg/normal m^3
   (d) 100 mg/normal m^3
3. Percentage by volume of nitrogen in the atmosphere is
   (a) 78 (b) 21
   (c) 2 (d) 0.93
4. Percentage by volume of oxygen in the atmosphere is
   (a) 78 (b) 21
   (c) 2 (d) 0.93
5. Percentage by volume of argon in the atmosphere is
   (a) 78 (b) 21
   (c) 2 (d) 0.93
6. Layers of atmosphere in correct sequence from earth's surface is
   (a) Troposphere, stratosphere, mesosphere, thermosphere
   (b) Stratosphere, mesosphere, thermosphere, troposphere
   (c) Mesosphere, stratosphere, thermosphere, troposphere
   (d) Thermosphere stratosphere, mesosphere, troposphere

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Answers

1. b 2. a 3. a 4. b 5. d 6. a

18.7.2 Review Questions

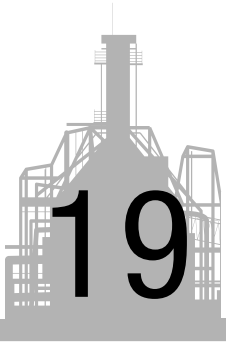
1. Discuss the environmental aspects of power generation.
2. What are the constituents of the atmosphere? Mention different pollutants due to thermal power plant and their effect on human health.
3. Explain environmental control of different pollutants such as particulate matter.
4. Write short notes on the following and mention their contribution in pollution:
   (a) Oxides of sulphur
   (b) Oxides of nitrogen
   (c) Oxides of carbon
5. Explain global warming and green house effect.
6. Define thermal pollution of water and its control.
7. Explain how acid precipitation is responsible for acid rain and acid snow.
8. What are the effluents from power plants, and how do they impact the environment?
9. Discuss the contribution of radiation from nuclear power plant effluents.
10. What is the environmental impact of power plant in relation to social and economical issues?

18.7.3 References

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Instrumentation and Equipments in Power Station 19



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19.1 ELEMENTS OF INSTRUMENTATION

For optimum operation of the power stations, it is essential to depend on automation and control utilizing very sophisticated controlling and monitoring techniques for power systems. Unlike the olden-day plants where control and monitoring systems were done operators utilizing simple circuits, the modern-day power plants are controlled almost entirely on computerized equipment, which has considerably simplified the control of processes involving multiple variables. In order to maximize the efficiency and availability, control and instrumentation or automation in power plants is inevitable.

In a typical power station, the major systems for power plant monitoring and control are as follows.

19.1.1 Process Control System

This is a closed-loop control system that automatically collects plant data by reading instruments, and monitors the physical and electrical parameters associated with the boiler, turbine and generator continuously. The system is used to control pumps, valves and switches for routine functions and during start-up or shutdown periods. Alarms and events are logged for better monitoring of the system.

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19.1.2 Operational Monitoring System

This system collects the operational data that could be utilized by the plant operators for record keeping, report writing and analysis.

19.1.3 Automatic Generation Control System

This system adjusts the generation capacity against load, thereby maintaining the quality and minimizing the cost of energy production and transmission.

19.1.4 Load Frequency Control System

This system is used to monitor generation load by maintaining frequency at the scheduled value. It also monitors net power interchanges with neighbouring control areas at the scheduled values and power allocation among generating units at economical rates.

19.1.5 Power Plant Maintenance

This system stores the information required for analysis of maintenance costs and evaluation of equipment performance. The system being an interactive one could be used by the plant personnel to enter problem data, planning data and work execution data, apart from entry and maintenance of an equipment database and for easy access to equipment history.

19.1.6 Plant Monitoring System

This system collects data for the purpose of fuel monitoring and performance calculations without performing any control actions. Data stored in the system may be retrieved to prepare reports and performance analysis.

19.2 IMPORTANT INSTRUMENTS ON STEAM GENERATOR AND TURBINE

Instrumentation on steam generator and turbines is very essential and vital for better control and problem rectification. Major parameters that need to be monitored are pressure, temperature, water level, frequency and feed rate. In this section, some of the important instruments used for control and installed on turbines and generators are discussed.

19.2.1 Drum Water-Level Control

To overcome the problems associated with water carryover, drum-level water control is very much essential. Higher level of water means water particles being carried into the super heater and turbine section, damaging the turbine blades and boiler tubes. This will result in additional maintenance cost. On the contrary, if the water level is too low, tubes may be overheated, ruptured or melted, thereby causing serious damage to the system and the injury or death to working personnel. There are three different methods of controlling the drum level, namely using single element, two elements or three elements.

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19.2.1.1 Single-Element Feed Water Control

It is a simple feedback loop controlling the drum level. In a single-element control system, the level transmitter (LT) sends a signal to the level controller. The process variable that is the input signal in this case is compared with the set value or set point (SP). The output signal is modified and adjusts to the set value of final control device, which could be a control valve, pump speed control or both as shown in Figure 19.1.

Figure 19.2 shows P&ID control of a single-element feedwater control.

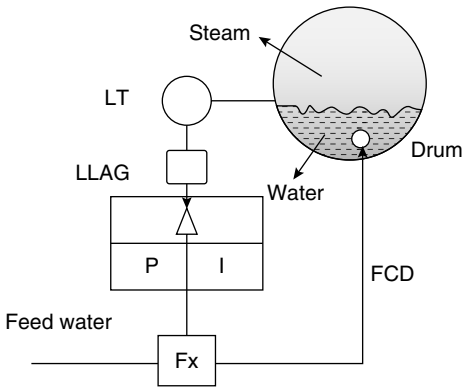


Fig. 19.1 *Single-Element Feedwater Control*

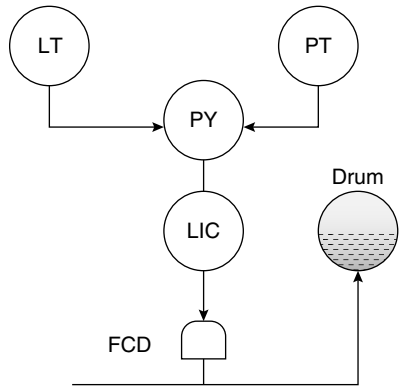


Fig. 19.2 *P&ID Diagram for Single-Element Feedwater Control*

19.2.1.2 Two-Element Feedwater Control

In a two-element system (Figure 19.3), steam flow is also taken into consideration (secondary variable). This system operates on simple feedback and feed-forward control principle. In this system, the secondary variable is related to a manipulated variable, which causes a change in the primary variable. On receiving the steam flow and drum-level controller signal, the steam flow adjusts the feedwater control valve

19.2.1.3 Three-Element Feedwater Control

The three-element system shown in Figure 19.4 combines or relates three parameters, namely steam flow, water flow in addition to drum-level control using a simple feedback, feed-forward and cascade control loop. Based on the steam flow and drum-level controller signal, the steam flow adjusts the feedwater control valve. By adding feedwater flow, the measured value is fed back to the controller.

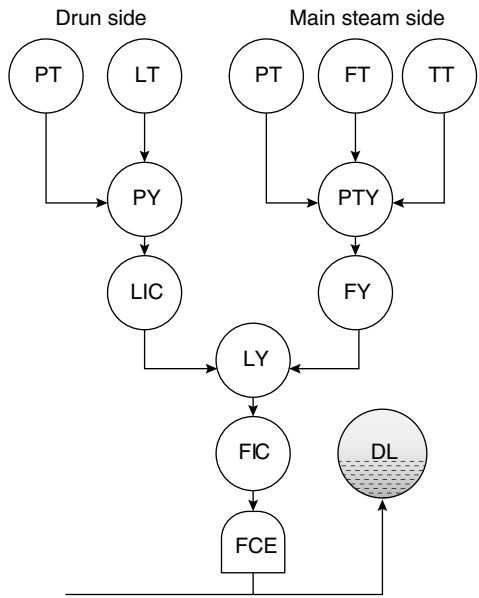


Fig. 19.3 *Two-Element Feedwater Control*

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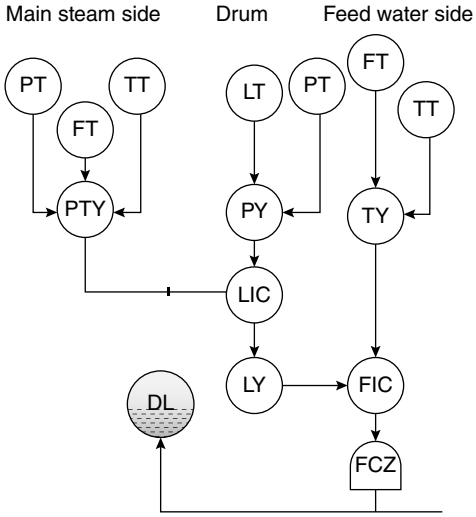


Fig. 19.4 *The Three-Element System*

19.3 COMBUSTION CONTROL

The phenomenon of combustion is far more complex than expected. When fuel and air supply to the furnace at any given instance of time is chemically ideal for combustion, it is known as stoichiometric fuel air ratio. However, given the practical conditions, it is always essential to supply excess air than the theoretically correct quantity. Any oversupply beyond this excess amount results in drop in efficiency of combustion and undesirable excess flow through the chimney.

To ensure sufficient amount of steam generation at the requisite pressure and to maintain correct proportion of fuel and air in a furnace for complete combustion, control of combustion in the furnace is very much essential. Control of combustion may be by manual or by automatic means. Automatic control of combustion is used to distribute the load evenly on the boiler as per the changing demand. Combustion can be effectively controlled by regulating the quantities of air and fuel or both. To ensure that products of combustion flow into the chimney at proper speed, it is essential to regulate differential gas pressure in the furnace. In case of automatic control, the controlling devices operate based on the change in pressure of steam.

Automatic combustion control works based on steam demand and results in efficient control of combustion. It is safer compared to manual control and saves manual labour. It comes into operation when any one of combination of the following fluctuations in the system occur: (i) steam pressure, (ii) steam flow rate and (iii) furnace draft. The action of primary forces is magnified through the master controller that in turn operates the concerned elements through electric, hydraulic or pneumatic forces.

Figure 19.5 shows a typical automatic combustion control system. The master controller *M* operates when the steam pressure falls below the required level. This activates the servo-motor coupled with the vanes of the ID fan that opens the air damper slightly. Simultaneously the secondary fan air damper also opens proportionately. As induced draft changes in the furnace, relay 1 (R_1) operates and adjusts the FD fan vanes until stability is restored in the combustion chamber. This results in an increased flow rate of air through the passage *P*. Due to increased air flow rate, relay 2 (R_2) is actuated, which speeds up the stoker motor to supply extra fuel. Hand adjustments are provided to the servo-motor and to the master controller to advance or retard the entire working system.

When the steam pressure increases, all the above actions are reversed.

19.3.1 Super Heat Temperature Control

The super heaters are used to increase the temperature of steam to the required turbine inlet condition and hence form part of a multivariable system. Figure 19.6 shows a simple super heater steam temperature control system. It is a cascade control based on fixed PID controllers

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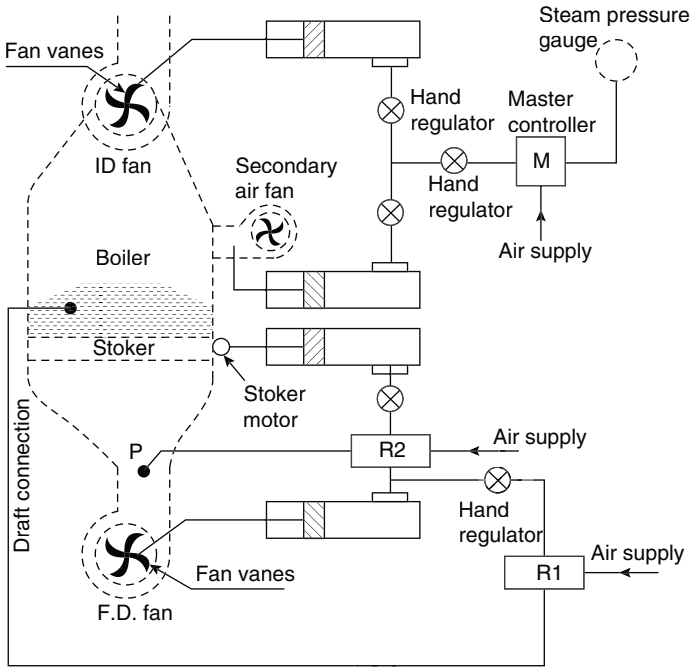


Fig. 19.5 *A Typical Automatic Combustion Control System*

in which the controlled variable is the outlet temperature. The inner loop that is faster than the outer loop has to reject temperature disturbances, originating upstream for this purpose. The system performance can be improved further by introducing load-dependent gain scheduling in the inner control loop.

19.4 INTRODUCTION TO GENERATOR AND EXCITERS

According to Faraday's law, when an electric conductor is moved through a magnetic field, an electrical voltage is induced in the conductor, the magnitude of which is directly proportional to the strength of the magnetic field and the rate at which the conductor crosses the magnetic field. According to Lenz's law, the induced voltage has a polarity that will oppose the change causing the induction. This natural phenomenon is known as generator action and is given by Faraday's law of electro-magnetic induction.

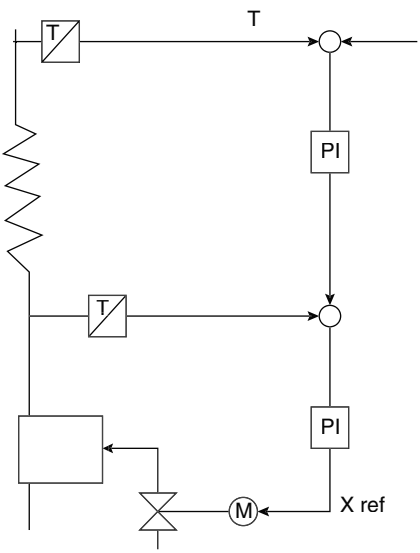


Fig. 19.6 *Superheater Temperature Control*

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If V_{ind} is the induced voltage, $\Delta Ø$ is the change in flux density, Δt is the change in time, then induced voltage is given by the following relation:

$$V_{ind} = \Delta Ø/\Delta t$$

All rotary generators built today use the basic principles of generator action as shown in Figure 19.7. When a magnetic field moves across a stationary conductor, a voltage is induced in the conductor. Reversing the direction of movement of the magnetic field will cause the direction of the induced voltage and the current also to reverse.

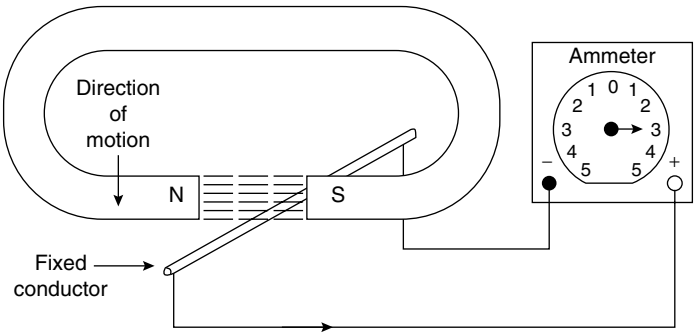


Fig. 19.7 *Working Principle of a Generator*

Figure 19.8 shows the construction of a basic generator.

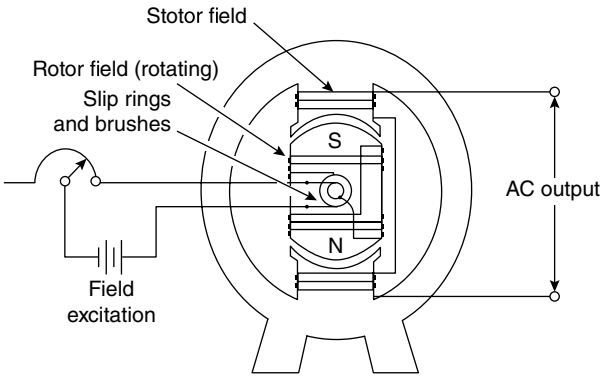


Fig. 19.8 *Construction of a Basic Generator*

19.4.1 The Elementary AC Generator

An elementary ac generator is shown in Figure 19.9. It consists of a conductor or loop of wire in a magnetic field produced using an electromagnet. The two ends of the loop are connected to slip rings and are in contact with two brushes as shown. When the loop is rotated, it cuts magnetic lines of force, first in one direction and then in the other direction.

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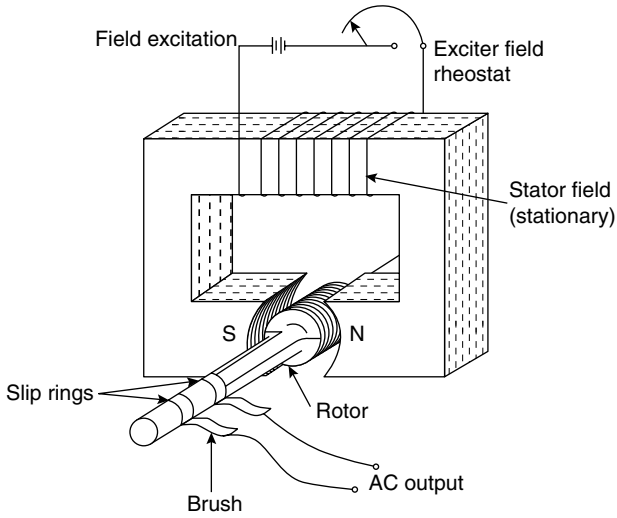


Fig. 19.9 *An Elementary AC Generator*

During the first half turn of rotation, a positive current is produced, whereas in the second half, a negative current is generated, completing one cycle of ac generation, as shown in Figure 19.10.

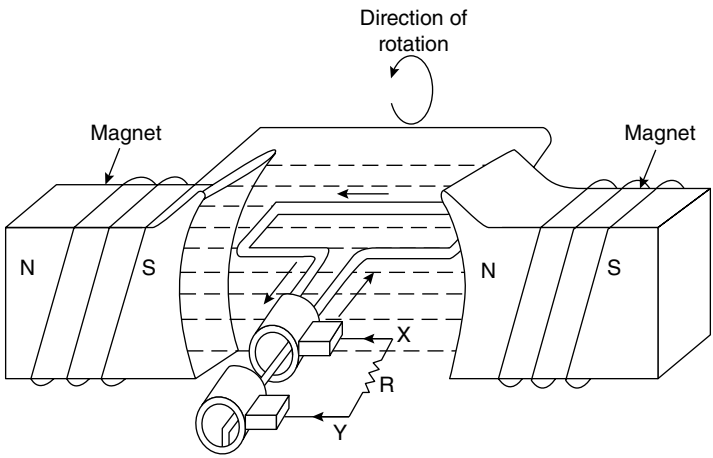


Fig. 19.10 *An AC Generator Working Principle*

19.4.2 Development of the Sine Wave

Referring to Figure 19.11, the instant the loop is in the vertical position, the loop sides move parallel to the field and do not cut the magnetic lines of force. Hence, there is no voltage induced in the loop. As the loop rotates further, the sides will cut the magnetic lines of force inducing voltage in the loop. When the loop is in the horizontal position (90^o), maximum voltage is induced.

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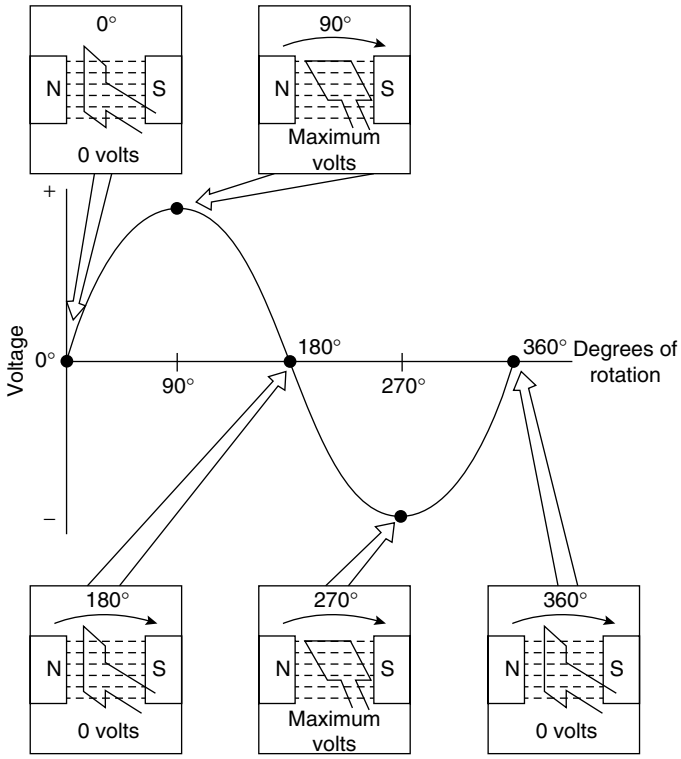


Fig. 19.11 *Development of Sine Wave*

The rotation of the coil through 360° results in an ac sine wave output as depicted in Figure 19.12.

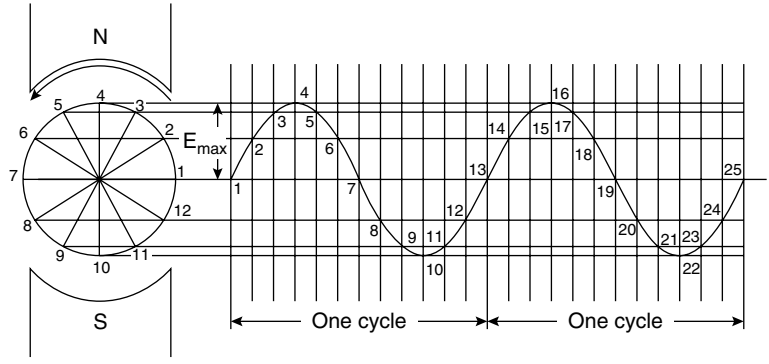


Fig. 19.12 *Sine Wave*

Three-phase voltage is developed utilizes the same principle of development of single-phase voltage. To develop three-phase voltage, three coils are required to be positioned at 120 electrical

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degrees apart as shown in Figure 19.13. A rotating magnetic field induces voltage in the coils that when aggregated produces the three-phase voltage pattern.

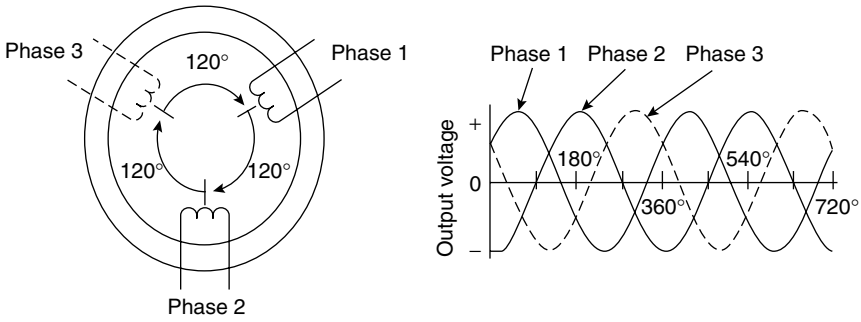


Fig. 19.13 *Three-Phase Voltage Development*

19.4.3 Types of Generators

Essentially, there are two basic types of generators, namely, DC and AC generators. AC generators are further classified as follows:

- Asynchronous (induction) generators
- Synchronous generators

19.4.3.1 Induction Generators

The induction generator is basically an induction motor driven above its synchronous speed by an amount not exceeding the full load slip the unit would have as a motor. For example for a motor with a synchronous speed of 1,200 rpm, assuming a full load slip of 3 per cent, the full load speed would be 1,164 rpm. This unit could also be driven by an external prime mover at 1,236 rpm for use as an induction generator. However, the induction generator requires VARs for excitation. The VARs may be supplied either by capacitors or from the utility grid. The former requires complex control. Induction generators are simple in construction and inexpensive, but they offer little control over their output. The induction generator does not require a separate DC excitation, regulator controls, frequency control or governor.

19.4.3.2 Synchronous Generators

Synchronous generators offer precise control of voltage, frequency, VARs and WATTs, etc., by utilizing voltage regulators and governors. A synchronous generator has a stationary armature winding, known as stator. To obtain the desired terminal voltage, the wires are connected in series or parallel. The armature winding is placed into a slotted laminated steel core. The rotor forms the other major part of the synchronous machine. A magnetic flux developed across the air gap between the rotor and stator generates an EMF. This magnetic flux developed by the DC field poles crosses the air gap of the stator windings, developing a sinusoidal voltage at the generator output terminals. This is known as electromagnetic induction.

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By controlling the amount of DC exciting current supplied to the field, the magnitude of the AC voltage generated can be controlled. The frequency of the voltage developed by the generator depends on the speed of the rotor and the number of field poles. For example, for a 60 Hz system,

$$\text{Frequency} = \text{speed (rpm)} * \text{pole pairs}/60$$

19.4.4 Generator Parts and Function

Major parts and functions of generator are as follows:

Generator frame gives the structural strength and rigidity and serves as a housing to guide cooling air flow.
Collector rings provide a connection and path for DC power into the rotating field windings.
Main coupling connects to the drive shaft.
Generator fan provides continuous circulation of cooling air.
Generator coolers remove heat from the generator cooling air.
Stator core houses the stationary windings forming a magnetic path necessary for induced voltages.
Inner end shield forms a path for cooled air.
Rotating field induces AC voltage in the stator windings.
Stator coil end turns get formed when coils leave one slot in a stator core and are returned to a different slot.
Terminal leads conduct the three-phase voltage and current flow from the generator stator to the external system.
Air gap is the radial clearance between the rotating field and the stator core.

19.4.5 The Exciter

The exciter is the power source used to supply the DC magnetizing current to the field windings of a synchronous generator, inducing AC voltage and current in the generator armature.

There are two basic kinds of exciters, namely rotating and static exciters:

1. *Rotating (brush and brushless)*

Brushless exciters do not require slip-rings, commutators, brushes and are practically maintenance free. Brush-type exciters require slip-rings, commutators and brushes and require periodic maintenance

2. *Static exciters (shunt and series)*

Static excitation is with no moving parts. It provides faster transient response than rotary exciters. They may be of two types as follows:

1. Shunt type: operating field power from generator output voltage
2. Series type: operating field power from generator output voltage and current

Figure 19.14 show conventional, static and brushless exciters.

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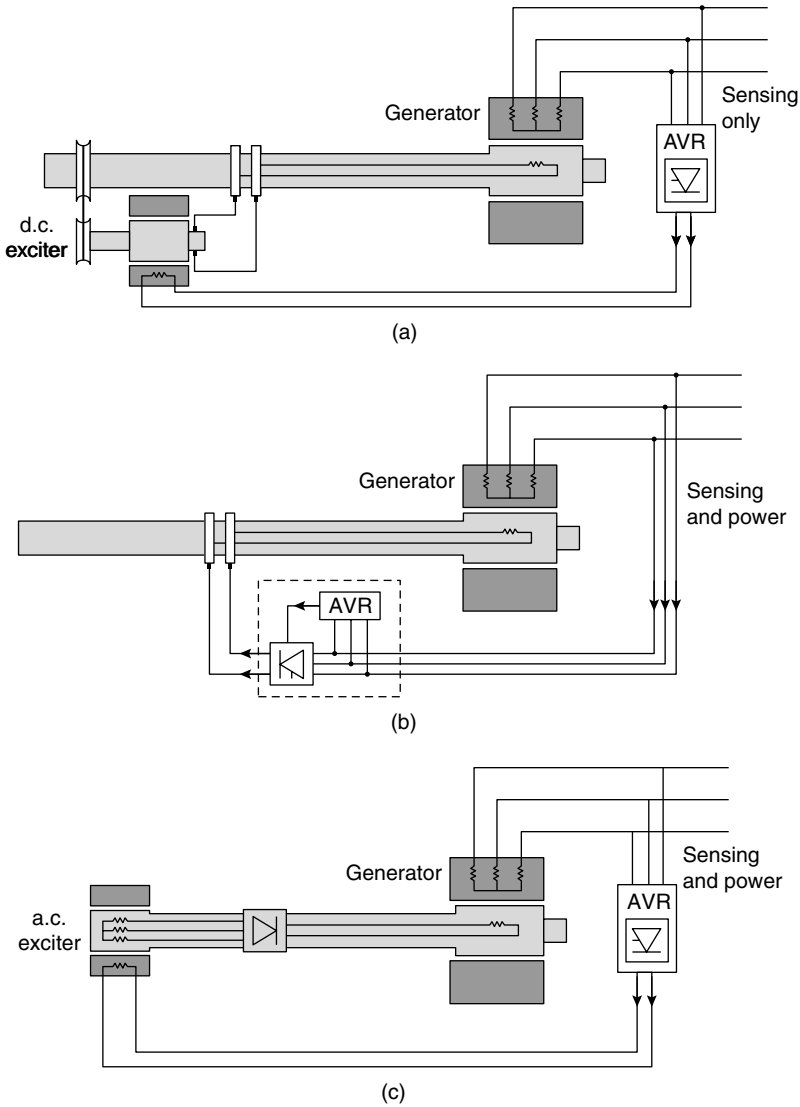


Fig. 19.14 *(a) Conventional, (b) static and (c) brushless exciters*

The amount of excitation required to maintain the output voltage constant is a function of the generator load. As the generator load increases, the amount of excitation increases. The amount of excitation required by a generator for a particular load is defined by the 'generator saturation curve' as shown in Figure 19.15.

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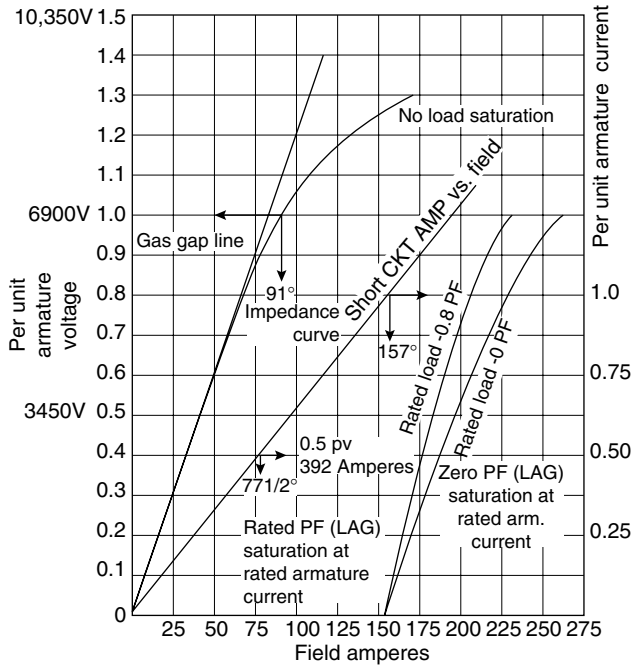


Fig. 19.15 *Three-phase voltage development*

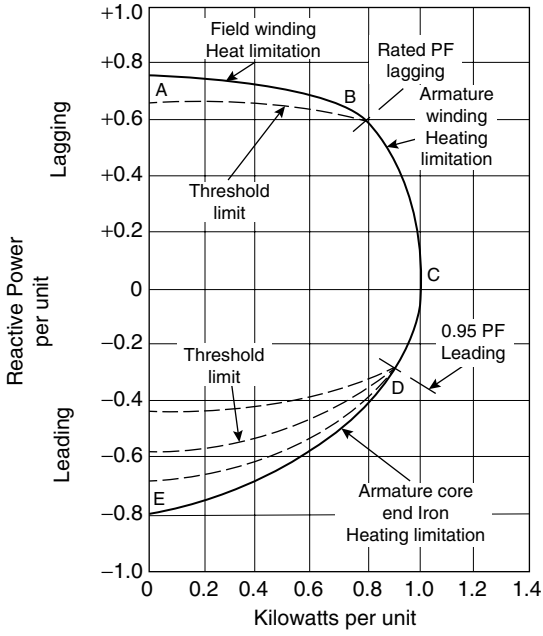


Fig. 19.16 *Three-phase voltage development*

The amount of power that a generator can deliver is defined by the 'generator capability curve' as shown in Figure 19.16.

19.5 EARTHING OF POWER SYSTEM

The main intention of grounding/earthing systems in power station is to provide a common ground reference for normal operation of electrical equipments including automation and measurement systems. In addition, a grounding system ensures the safety of the working personnel and prevents the insulation and other devices from possible damage owing to faulty power supply, lightning, etc. An ideal grounding system should

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- have appropriate ground impedance at both low- and high-level frequencies.
- minimize the maximum values of voltages between different points at conductors and the earth surface.

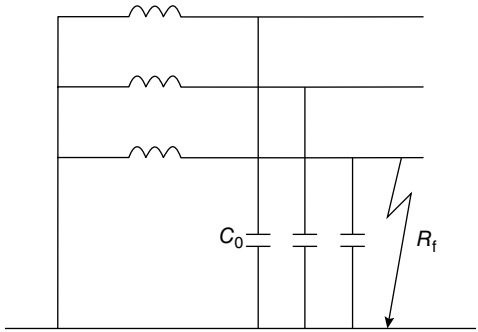


Fig. 19.17 *Earth Fault in a Solidly Earthed Network*

A power system can have more than one neutral point, and it is not necessary that all neutral points of one system be connected to earth, using the same earthing method. Two important functions of neutral earthing are to detect earth faults and to control the fault current. This is because large fault currents can lead to potential rise of exposed parts of the power system reaching dangerous levels. Power systems could be solidly earthed or non-solidly earthed. The latter category includes isolated neutral, resistance and resonant earthing.

In a solidly earthed system, a number of transformer neutrals are directly earthed. A single-phase earth fault current in a solidly earthed system as shown in Figure 19.17 may exceed three-phase fault current. To reduce the magnitude of the current, some of the neutrals are left unearthed.

19.6 POWER AND UNIT TRANSFORMER

As per ANSI/IEEE definition, a transformer is a static electrical device involving no continuously moving parts, used in electric power systems to transfer power between circuits through the use of electromagnetic induction.

Since generation of electrical power in low voltage level is very much cost effective, electrical power generation is done at low voltage level. If the voltage level of a power is increased, the electric current of the power is decreased resulting in lower ohmic or I^2R losses in the system, improved the voltage regulation, reduction in cross-sectional area of the conductor, and hence the capital cost. Due to these reasons, low-level power is stepped up using a step-up transformer for efficient electrical power transmission. As this high-voltage power may not be distributed to the consumers directly, it must subsequently be stepped down at the receiving end with help of step-down transformer.

A power transformer is the one rated at 500 kVA and above, used between the generating and distributing systems, which includes generating locations, distribution points and interconnection points. Based on the customized applications, power transformers are selected. Step-up transformers are used at the generator end (known as GSU transformers), whereas step-down transformers are used at feed distribution circuits. Both single-phase and three-phase transformers are available.

Based on the application, power transformers are classified as dry type (generally for indoor applications) or liquid type (generally for outdoor applications).

Efficiency of power transformers is given by the following relation:

$$\eta = \frac{kVA}{kVA + Losses}$$

where kVA is the rating.

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A unit auxiliary transformer (UAT) is a power transformer that supplies power to the auxiliary equipment of a power-generating station during its normal working condition. A UAT is connected directly to the generator output by using a tap-off of the isolated phase bus duct. This is the cheapest source of power to the generating station, coming in three-windings. One winding is primary and two separate windings constitute secondary. While the primary winding of UAT is equal to the main generator voltage rating, the secondary windings can have same or different voltages, that is, generally 11 kV and or 6.9 kV as per plant requirement.

19.7 CIRCUIT BREAKERS

The circuit breaker is a mechanical switching device that under normal circuit conditions is capable of

- protecting the circuit wiring
- making, carrying and breaking currents

It also makes and carries and breaks current for a specified time under specified abnormal circuit conditions, namely short circuit instances. Depending on their breaking capability, construction type and capability to limit short-circuit currents, circuit breakers are classified as follows:

1. Current–zero interrupting type
2. Current limiting type

Depending on over-current characteristics, circuit breakers are further classified as follows:

- Circuit breakers for motor protection
- Circuit breakers for the protection of connecting circuits and installations.

Some commonly used abbreviations for the designations of circuit breakers are as follows:

1. *Air circuit breaker or ACB*

   These are large open-type circuit breakers used for the protection of installations in the current range of >100 A approximately.

2. *Miniature circuit breaker or MCB*

   These are small circuit breakers used for protection of single or multiple pole wiring, especially in buildings.

3. *Moulded case circuit breaker or MCCB*

   These are compact circuit breakers and have a supporting housing of moulded insulating material.

19.8 PROTECTIVE EQUIPMENTS

When it comes to safety, it is essential to keep workers, plant machineries and the environment safe and in compliance with government norms. The power plant requires specialized attention in this regard. Some of the safety equipments include the following:

- Fall arrest protection
- Industrial communications

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- Respiratory protection and breathing air
- Industrial fire systems
- Gas detection and monitoring
- Industrial hygiene services
- Medical care
- Confined space entry, including atmospheric testing and ventilation
- Personal protective equipment or PPE

19.9 CONTROL BOARD EQUIPMENT

19.9.1 Control Room Instrumentation

A centralized operation of the on-site utilities is essential for large power plants with a provision for control room (CR).

(i) Instrumentation in the control room is microprocessor-based that helps the power plant operator to monitor the operating status of all equipment, and to operate them in addition to evaluate the system conditions.

(ii) The control room instrumentation is also required for monitoring and control of auxiliary equipment and all equipment required regulating the environment of the site.

(iii) It also enables the operators to measure operating conditions that could be used for thermodynamic and performance analysis of the power plant on a continuous basis.

19.9.1.1 A Master Frequency Control System

This system is provided for the primary switchgear main bus for sensing and controlling any or all of the buses:

(i) when they are electrically connected to each other

(ii) when they are in service but electrically separated

(iii) when only one is in service

This system includes the following:

(a) Frequency recorders and frequency deviation transducers

(b) Master frequency standard

(c) Governor–actuator devices

19.9.1.2 A Master Voltmeter Recorder System

This system is used to indicate and record the line-to-line voltages on the main buses and on each medium-voltage power supply bus. While a voltmeter measures true RMS voltage, a multi-channel recorder is used to precisely record all bus voltages.

19.9.1.3 Recording Watt-Hour Metres and Demand Metres

These instruments are installed for the commercial power connection, satisfying the serving utility and local regulating agencies requirements. They provide the data needed to confirm billing by the utility company. Voltage and current measurements are recorded as true RMS.

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Table 19.1 *CR Control Panel Instruments and Controls for Prime Movers*

S. no.	Instrumentation	Description of location
1.	Pressure indicator	Fuel oil supply to engine
2.	Pressure indicator	Fuel from main storage tank
3.	Pressure indicator	Lube oil, supply to engine
4.	Pressure indicator	Lube oil, supply to turbo charger
5.	Pressure indicator	Combustion air, filter downstream
6.	Pressure indicator	Combustion air, turbo charger
7.	Pressure indicator	Starting air–air receiver
8.	Pressure indicator	Cooling water pump discharge
9.	Temperature indicator	High engine oil temperature
10.	Level indicator	Fuel, main storage tank
11.	Level indicator	Fuel, day storage tank
12.	Level indicator	Lube oil of sump tank
13.	Level indicator	Jacket water, surge tank
14.	Temperature indicator	Cooling water supply to engine
15.	Temperature indicator	Cooling water return from engine
16.	Temperature indicator	Each cylinder and combined exhaust
17.	Motor control switches	Jacket water pumps, fuel oil transfer pumps, cooling tower fans, centrifuges and other auxiliaries
18.	Alarms	High, low temperature, oil pressure, levels

Table 19.1 lists the minimum requirement for the CR control panel instruments and controls for prime movers.

19.10 SWITCH GEAR FOR POWER STATION AUXILIARIES

Switch gears are line of equipments used for housing the circuit breakers, protective relays and control wiring. The switchgear is enclosed in metallic containers that prevent the individuals from possible electric hazards. Switchgear is made of a series of cubicles bolted together in a row. In the rear section of each cubicle, bus bars are located while the lower and upper compartments house circuit breakers and protective relays and breaker control wiring, respectively.

19.11 TESTING OF POWER PLANTS AND HEAT BALANCE

Some of the key parameters that influence the performance of the boiler are the following:

(i) Efficiency and evaporation ratio reduces with time

(ii) Poor combustion

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(iii) Heat transfer fouling and poor operation and maintenance

(iv) Deterioration of fuel quality and water quality

In order to find out the reasons for poor performance of the plant generally boiler efficiency and evaporation ratio are determined and any abnormal deviations could be investigated to identify the specific problem for necessary corrective action.

Boiler efficiency is evaluated using the following relation:

$$\eta = \frac{Q_{\text{output}}}{Q_{\text{input}}} \times 100$$

Boiler evaporation ratio is defined as the ratio of amount of steam generated to quantity of fuel consumed.

$$E = \frac{m_{\text{steam}}}{m_{\text{fuel}}} \times 100$$

Different standards are used or followed to evaluate the efficiency and heat balance. Some of the standards are discussed below.

19.11.1 British Standards, BS845: 1987

The British Standard BS 845: 1987 describes the methods and conditions under which a boiler should be tested to determine its efficiency. For the testing purpose, boiler should be operated under steady load conditions (generally full load) for a period of one hour after which readings would be taken during the next hour of steady operation. Efficiency calculations are done based on these values obtained.

According to this standard, the boiler efficiency is defined as the per cent of useful heat available, expressed as a percentage of the total energy potentially available by burning the fuel based on its gross calorific value (GCV). Further, heat balance is done which can be categorized into two parts:

(i) Part one deals with standard boilers, and uses indirect method (heat loss method) for heat balance calculations.

(ii) Part two deals with complex plant where there are many channels of heat flow. In this case, both the direct (input–output method) and indirect methods are applicable, in whole or in part.

19.11.2 ASME Standard: PTC-4-1 Power Test Code for Steam Generating Units

According to this standard,

(i) **Part one** uses direct method. In this method, the energy gain of the working fluid (water and steam) is compared with the energy content of the boiler fuel.

(ii) **Part two** uses indirect method. In this method, the efficiency is calculated based on the difference between the losses and the energy input.

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19.11.3 IS 8753: Indian Standard for Boiler Efficiency Testing

Most standards for computation of boiler efficiency, including IS 8753 and BS 845 are designed for spot measurement of boiler efficiency.

It may be noted that all the above standards do not include blowdown as a loss in the efficiency determination process.

19.11.3.1 The Direct Method Testing

This is also known as input–output method, because the data needed for performance evaluation are steam generated as output, and the amount of fuel burnt as heat input, as depicted in Figure 19.18.

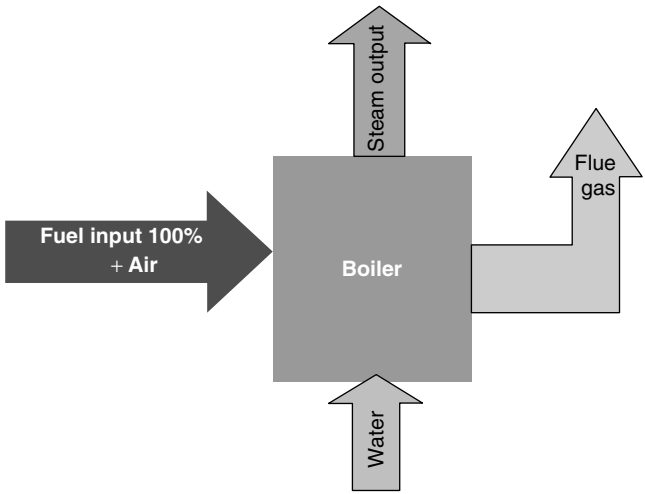


Fig. 19.18 *Direct Method Testing*

19.11.3.2 Parameters Required for Direct Method Testing

(i) *Heat input*

For heat input measurement, calorific value of the fuel and its flow rate in terms of mass or volume must be known.

(ii) *Heat output*

There are several methods that can be used for measuring heat output:

–With steam boilers, an installed steam metre can be used to measure flow rate.

–For small boilers, the alternative is to measure feed water by previously calibrating the feed tank and noting down the levels of water during the beginning and end of the trial. Heat output is the heat addition for conversion of feed water at inlet temperature to steam.

–For boilers with intermittent blowdown, blowdown should be avoided during the trial period.

–In case of boilers with continuous blowdown, the heat loss due to blowdown should be calculated and added to the heat in steam.

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19.11.3.3 Merits and Demerits of Direct Method

(i) *Merits*

(a) It is useful to quickly evaluate the efficiency of boilers.

(b) It requires few parameters for computation.

(c) Fewer instruments are required for monitoring.

(ii) *Demerits*

(a) It is not possible to analyse the reason for lower efficiency of system.

(b) It does not include various losses accountable for various efficiency levels.

19.11.4 The Indirect Method Testing

The efficiency is determined by deducting all the possible losses occurring in the boilers from 100 (see Figure 19.19).

i. e.,

$$\text{Efficiency} = 100 - \text{items } (L1 + L2 + L3 + L4 + L5 + L6 + L7 + L8)$$

where

L_1 – loss due to dry flue gas (sensible heat)
L_2 – loss due to hydrogen in fuel (H_2)
L_3 – loss due to moisture in fuel (H_2O)
L_4 – loss due to moisture in air (H_2O)
L_5 – loss due to carbon monoxide (CO)
L_6 – loss due to surface radiation, convection and other unaccounted

The following losses are applicable to solid fuel fired boiler in addition to the above:

L_7 – unburnt losses in fly ash (carbon)
L_8 – unburnt losses in bottom ash (carbon)

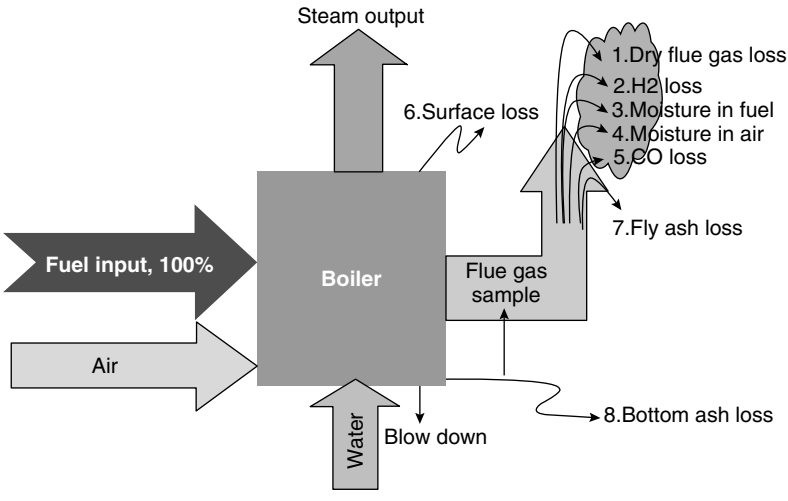


Fig. 19.19 *Indirect Method Testing*

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19.11.5 Energy Balance Sheet

After determining all the losses as mentioned earlier, a simple energy balance sheet is prepared to depict the efficiency of the boiler as shown below.

Boiler energy balance Input–output parameter	kJ/kg of fuel	%
Heat input in fuel	=	100
L_1 – loss due to dry flue gas (sensible heat)		
L_2 – loss due to hydrogen in fuel (H_2)		
L_3 – loss due to moisture in fuel (H_2O)		
L_4 – loss due to moisture in air (H_2O)		
L_5 – loss due to carbon monoxide (CO)		
L_6 – loss due to surface radiation, convection and other unaccounted		
L_7 – unburnt losses in fly ash (carbon)		
L_8 – unburnt losses in bottom ash (carbon)		
Total losses		
Boiler efficiency = 100 – total losses		

19.12 QUESTIONS

19.12.1 Objective Questions

1. The greatest heat loss in an oil-fired boiler is from
   (a) uncontrolled escape of combustion gases up the stack
   (b) radiation in the furnace casing
   (c) blowdown
   (d) incomplete combustion
2. Temperature measurement is an indication of the
   (a) total heat contained in any closed energy system
   (b) level of heat intensity
   (c) total heat of a substance
   (d) rate of heat transfer from one substance to another

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3. A two-element feedwater control system uses
   (a) only drum level as control parameter
   (b) only steam flow rate as control parameter
   (c) both steam flow rate and drum level as control parameters
   (d) temperature as control parameter
4. A single-element feedwater control system uses
   (a) only drum level as control parameter
   (b) only steam flow rate as control parameter
   (c) both steam flow rate and drum level as control parameters
   (d) temperature as control parameter
5. A three-element feedwater control system uses
   (a) only drum level as control parameter
   (b) only steam flow rate as control parameter
   (c) only water flow rate control parameters
   (d) all of the above
6. In an AC generator, maximum voltage is induced
   (a) when the loop is in the horizontal position
   (b) when the loop is in the vertical position
   (c) cannot say
   (d) not dependent on loop position
7. In an AC generator, zero voltage is induced
   (a) when the loop is in the horizontal position
   (b) when the loop is in the vertical position
   (c) cannot say
   (d) not dependent on loop position

Answers

1. a 2. b 3. c 4. a 5. d 6. a 7. b

19.12.2 Review Questions

1. Explain different methods of controlling the drum level of a steam generator.
2. Sketch and explain the working of an AC generator.
3. How sine waves are generated in an AC generator? Explain with a neat sketch.
4. What is an exciter? Explain its function.
5. Discuss on different methods used for performance testing of a power plant.
6. Differentiate between direct and indirect method of evaluating power plant performance.
7. Write a short note on protective equipments and control panel instrumentation.
8. Discuss the method of automatic combustion control.

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19.12.3 References

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