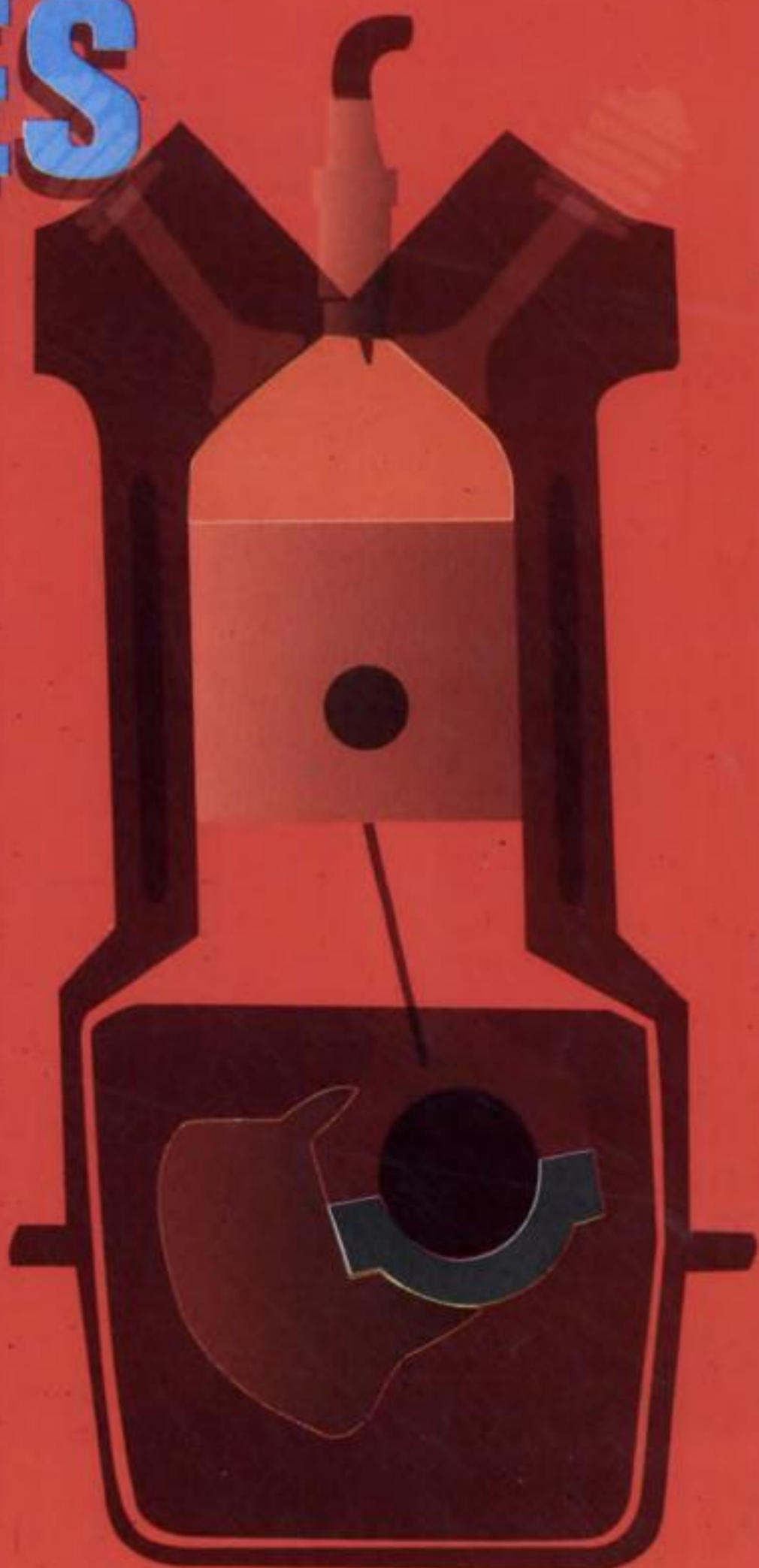


INTERNAL COMBUSTION ENGINES

Second Edition

V GANESAN



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NOMENCLATURE

A

a_1	constant
$amep$	mean effective pressure required to drive the auxiliary components
A	piston area [Chp.1]
A	area of heat transfer [Chp.14]
A	average projected area of each particles [Chp.15]
A	TEL in ml/gal of fuel [Chp.6]
A_1	cross-sectional area at inlet of the carburettor
A_2	cross-sectional area at venturi of the carburettor
A_{act}	actual amount of air in kg for combustion per kg of fuel
A_f	area of cross-section of the fuel nozzle [Chp.8]
A_f	area of fin [Chp.14]
A_e	effective area
A_{th}	theoretical amount of air in kg per kg of fuel
A/F	air-fuel ratio

B

b_1	constant
bp	brake power
bhp	brake horsepower
$bmep$	brake mean effective pressure
$bsfc$	brake specific fuel consumption
BDC	Bottom Dead Centre

C

$cmep$	mean effective pressure required to drive the compressor or scavenging pump
C	velocity [Chp.6]

C_d	coefficient of discharge for the orifice [Chp.9]
C_{da}	coefficient of discharge for the venturi
C_{df}	coefficient of discharge for fuel nozzle
C_f	fuel velocity at the nozzle exit
C_p	specific heat of gas at constant pressure
C_{rel}	relative charge
C_v	specific heat at constant volume
CV	calorific value of the fuel

D

d	cylinder bore diameter [Chp.1]
d	diameter of orifice [Chp.9]
D	brake drum diameter

E

ϵ	expansion ratio
E	stored energy [Chp.2]
E	enrichment [Chp.8]
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening

F

f	coefficient of friction
f_{mep}	frictional mean effective pressure
f_p	frictional power
F	force
F/A	fuel-air ratio
F_R	relative fuel-air ratio

G

g	acceleration due to gravity
g_c	gravitational constant
gp	gross power

H

h	specific enthalpy
h	pressure difference [Chp.9]
h	convective heat transfer coefficient [Chp.14]
H	enthalpy

I

ip	indicated power
$imep$	indicated mean effective pressure
$isfc$	indicated specific fuel consumption
I	intensity
IDC	Inner Dead Centre
IVC	Inlet Valve Closing
IVO	Inlet Valve Opening

K

k	thermal conductivity of gases
k_1	constant [Chp.4]
K	number of cylinders
K_{ac}	optical absorption coefficient

L

l	characteristic length
l	distance [Chp.16]
L	stroke
L_ℓ	length of the light path

M

m	mass
m	exponent [Chp.14]
mep	mean effective pressure
$mmep$	mechanical mean effective pressure
$\dot{m}_a$	mass flow rate of air
$\dot{m}_{act}$	actual mass flow rate of air
$\dot{m}_{th}$	theoretical mass flow rate of air
M_{del}	mass of fresh air delivered
M_f	molecular weight of the fuel
M	molecular weight
M_{ref}	reference mass

N

n	number of power strokes
n	polytropic index [Chp.2]
n	number of soot particles per unit volume [Chp.16]
N	speed in revolutions per minute
N_i	number of injections per minute [Chp.9]

O

<i>ODC</i>	Outer Dead Centre
<i>ON</i>	Octane Numbers

P

<i>p</i>	pressure
<i>pmep</i>	charging mean effective pressure
<i>pp</i>	pumping power
<i>p_{ar}</i>	pure air ratio
<i>p_{bm}</i>	brake mean effective pressure
<i>p_e</i>	exhaust pressure
<i>p_i</i>	inlet pressure
<i>p_{im}</i>	indicated mean effective pressure
<i>p_m</i>	mean effective pressure
<i>P_{cyl}</i>	pressure of charge inside the cylinder
<i>P_{inj}</i>	fuel pressure at the inlet to injector
<i>P_l</i>	pressure loss coefficient
<i>P_s</i>	specific power output
<i>PN</i>	performance number

Q

<i>q</i>	heat transfer
<i>q̇</i>	rate of heat transfer
<i>Q_R</i>	heat rejected
<i>Q_S</i>	heat supplied

R

<i>r</i>	compression ratio
<i>rpn</i>	relative performance number
<i>r_c</i>	cut-off ratio
<i>r_p</i>	pressure ratio
<i>R</i>	length of the moment arm
<i>R</i>	delivery ratio [Chp.20]
<i>R̄</i>	universal gas constant
<i>R_{del}</i>	delivery ratio

S

<i>s̄_p</i>	mean piston speed
<i>sfc</i>	specific fuel consumption
<i>S</i>	spring scale reading

T

t	time
T	absolute temperature
T	torque [Chp.17]
TDC	Top Dead Centre
T_b	black body temperature
T_f	friction torque
T_g	mean gas temperature
T_l	load torque

U

u	specific internal energy
U	internal energy
U_c	chemical energy
U_s	stored energy

V

v	specific volume
V	volume
V_{ch}	volume of cylinder charge
V_{cp}	volume of combustion products
V_{del}	volume of air delivered
V_f	fuel jet velocity
V_{pure}	volume of pure air
V_{ref}	reference volume
V_{res}	volume of residual gas
V_{ret}	volume of retained air or mixture
V_s	displacement volume
V_s	swept volume
V_{short}	short circuiting air
V_{tot}	total volume
V_C	clearance volume
V_T	volume at bottom dead centre

W

w	specific weight
w	work transfer [Chp.8]
W	net work
W	weight [Chp.16]
W	number of quartz windows [Chp.16]
W	load [Chp.13]

WOT	Wide Open Throttle
W_C	compressor work
W_T	turbine work
W_x	external work

Z

z	height of the nozzle exit [Chp.8]
z_2	datum height [Chp.2]
Z	constant

GREEK

α	air coefficient
γ	ratio of specific heats
Δp	pressure difference
ΔT	temperature difference between the gas and the wall
ϵ	heat exchanger efficiency
η	efficiency
$\eta_{air\ std}$	air standard efficiency
η_{bth}	brake thermal efficiency
η_c	compressor efficiency
η_{ch}	charging efficiency
η_{ith}	indicated thermal efficiency
η_m	mechanical efficiency
η_{rel}	relative efficiency
η_{sc}	scavenging efficiency
η_t	turbine efficiency
η_{th}	thermal efficiency
η_{trap}	trapping efficiency
η_v	volumetric efficiency
θ	crank angle [Chp.12]
θ	specific absorbance per particle [Chp.16]
λ	wave length [Chp.16]
λ	excess air factor [Chp.20]
μ	kinematic viscosity of gases
ν	dynamic viscosity
ρ	density
ρ_f	density of fuel
ϕ	equivalence ratio
ψ	magnetic field strength
ω	angular velocity

1

INTRODUCTION

1.1 ENERGY CONVERSION

The distinctive feature of our civilization today, one that makes it different from all others, is the wide use of mechanical power. At one time, the primary source of power for the work of peace or war was chiefly man's muscles. Later, animals were trained to help and afterwards the wind and the running stream were harnessed. But, the great step was taken in this direction when man learned the art of energy conversion from one form to another. The machine which does this job of energy conversion is called an engine.

1.1.1 Definition of 'Engine'

An engine is a device which transforms one form of energy into another form. However, while transforming energy from one form to another, the efficiency of conversion plays an important role. Normally, most of the engines convert thermal energy into mechanical work and therefore they are called 'heat engines'.

1.1.2 Definition of 'Heat Engine'

Heat engine is a device which transforms the chemical energy of a fuel into thermal energy and utilizes this thermal energy to perform useful work. Thus, thermal energy is converted to mechanical energy in a heat engine.

Heat engines can be broadly classified into two categories:

- (i) Internal Combustion Engines (IC Engines)
- (ii) External Combustion Engines (EC Engines)

1.1.3 Classification and Some Basic Details of Heat Engines

Engines whether Internal Combustion or External Combustion are of two types, viz.,

- (i) Rotary engines
- (ii) Reciprocating engines

A detailed classification of heat engines is given in Fig.1.1. Of the various types of heat engines, the most widely used ones are the reciprocating internal combustion engine, the gas turbine and the steam turbine. The steam engine is rarely used nowadays. The reciprocating internal combustion engine enjoys some advantages over the steam turbine due to the absence of heat exchangers in the passage of the working fluid (boilers and condensers in steam turbine plant). This results in a considerable mechanical simplicity and improved power plant efficiency of the internal combustion engine.

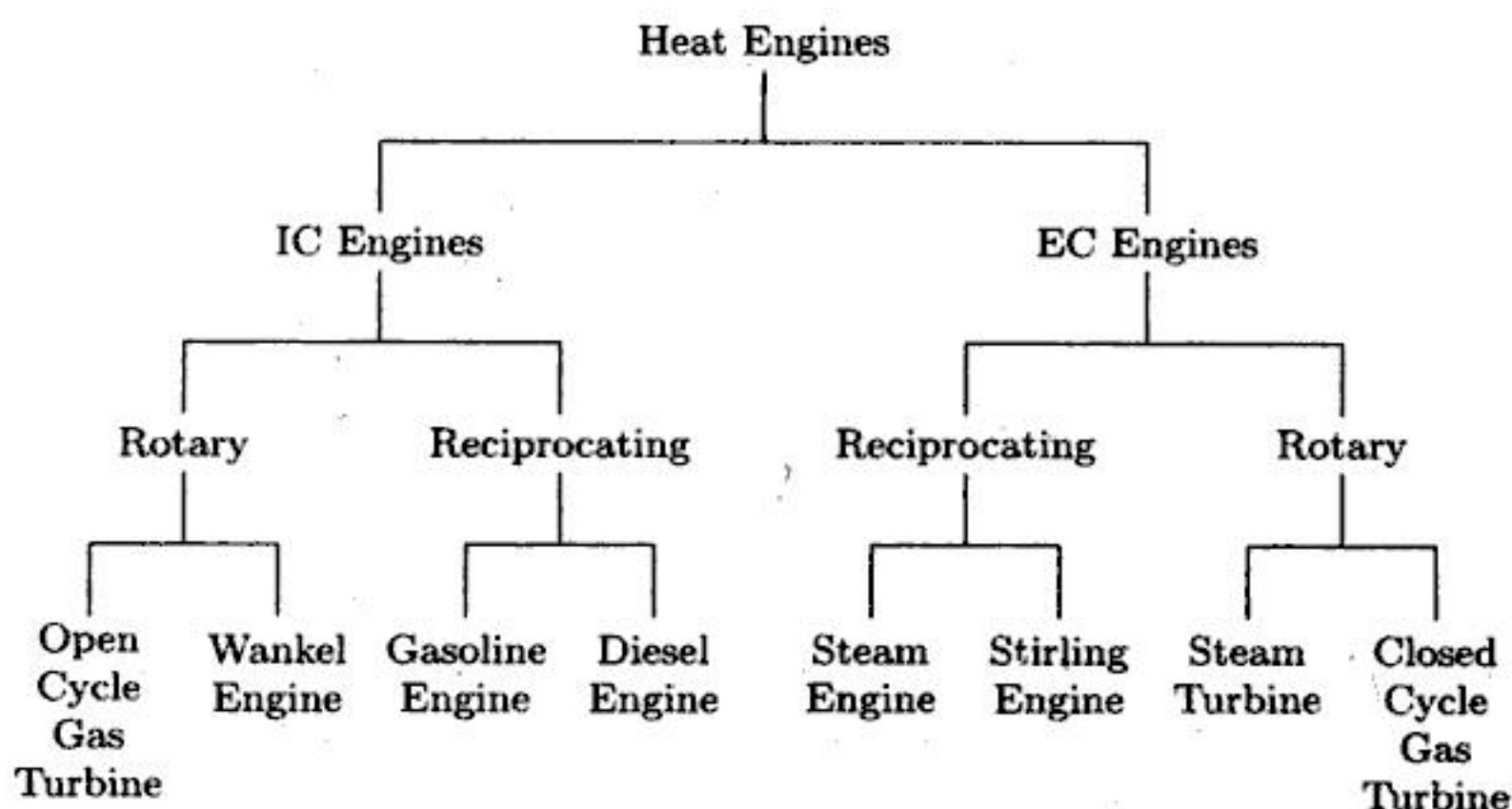


Fig. 1.1 Classification of Heat Engines

Another advantage of the reciprocating internal combustion engine over the other two types is that all its components work at an average temperature which is much below the maximum temperature of the working fluid in the cycle. This is because the high temperature of the working fluid in the cycle persists only for a very small fraction of the cycle time. Therefore, very high working fluid temperatures can be employed resulting in higher thermal efficiency.

Further, in internal combustion engines, higher thermal efficiency can be obtained with moderate maximum working pressure of the fluid in the cycle, and therefore, the weight to power ratio is less

than that of the steam turbine plant. Also, it has been possible to develop reciprocating internal combustion engines of very small power output (power output of even a fraction of a kilowatt) with reasonable thermal efficiency and cost.

The main disadvantage of this type of engine is the problem of vibration caused by the reciprocating components. Also, it is not possible to use a variety of fuels in these engines. Only liquid or gaseous fuels of given specification can be efficiently used. These fuels are relatively more expensive.

Considering all the above factors the reciprocating internal combustion engines have been found suitable for use in automobiles, motor-cycles and scooters, power boats, ships, slow speed aircraft, locomotives and power units of relatively small output.

1.1.4 External Combustion and Internal Combustion Engines

External combustion engines are those in which combustion takes place outside the engine whereas in internal combustion engines combustion takes place within the engine. For example, in a steam engine or a steam turbine, the heat generated due to the combustion of fuel is employed to generate high pressure steam which is used as the working fluid in a reciprocating engine or a turbine.

In case of gasoline or diesel engines, the products of combustion generated by the combustion of fuel and air within the cylinder form the working fluid.

1.2 BASIC ENGINE COMPONENTS AND NOMENCLATURE

Even though reciprocating internal combustion engines look quite simple, they are highly complex machines. There are hundreds of components which have to perform their functions satisfactorily to produce output power. There are two types of engines, viz., spark-ignition (SI) and compression-ignition (CI) engine. Let us now go through the important engine components and the nomenclature associated with an engine.

1.2.1 Engine Components

A cross section of a single cylinder spark-ignition engine with overhead valves is shown in Fig.1.2. The major components of the engine and their functions are briefly described below.

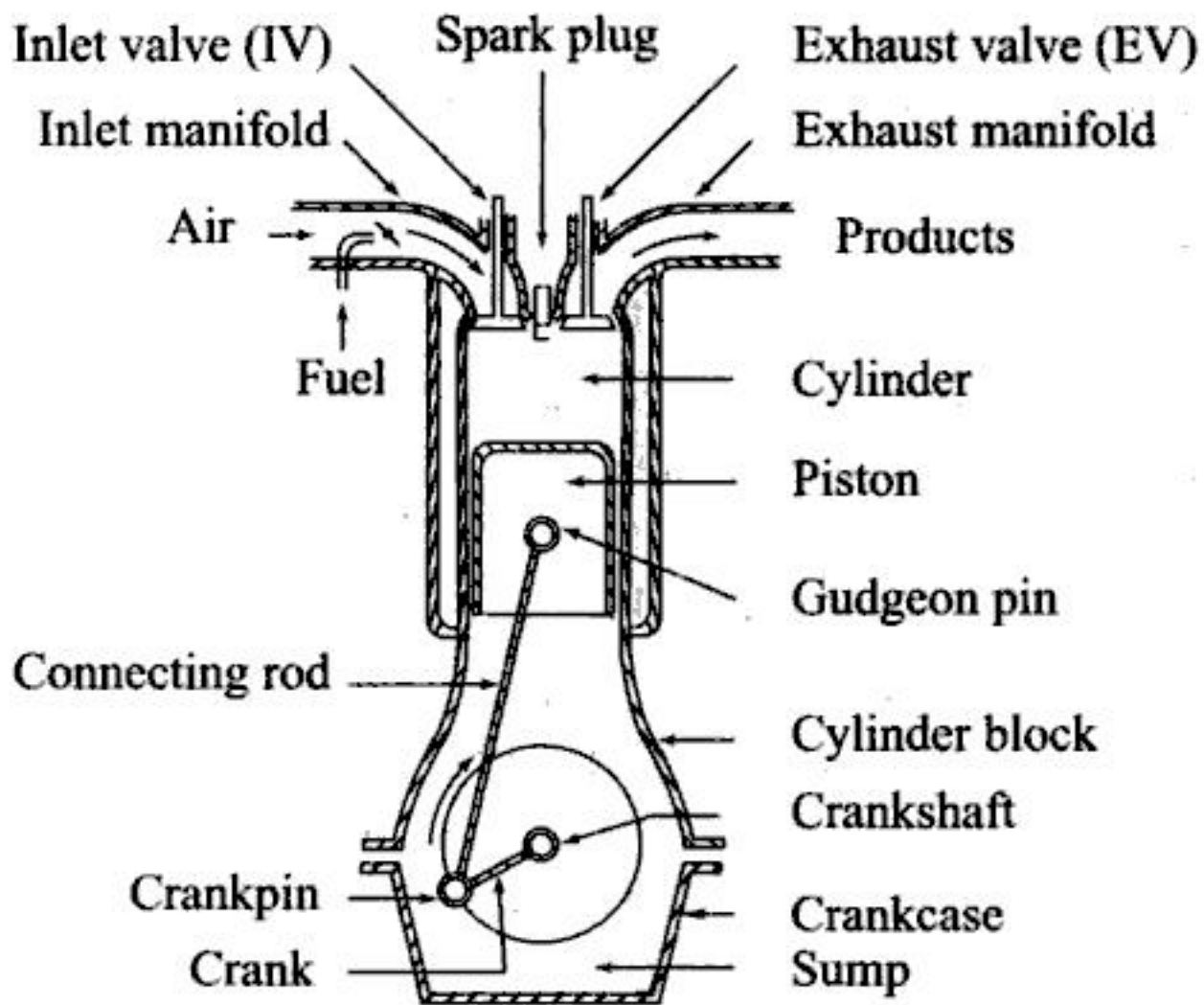


Fig. 1.2 Cross-section of a Spark-Ignition Engine

Cylinder Block : The cylinder block is the main supporting structure for the various components. The cylinder of a multicylinder engine are cast as a single unit, called cylinder block. The cylinder head is mounted on the cylinder block. The cylinder head and cylinder block are provided with water jackets in the case of water cooling or with cooling fins in the case of air cooling. Cylinder head gasket is incorporated between the cylinder block and cylinder head. The cylinder head is held tight to the cylinder block by number of bolts or studs. The bottom portion of the cylinder block is called crankcase. A cover called crankcase which becomes a sump for lubricating oil is fastened to the bottom of the crankcase. The inner surface of the cylinder block which is machined and finished accurately to cylindrical shape is called bore or face.

Cylinder : As the name implies it is a cylindrical vessel or space in which the piston makes a reciprocating motion. The varying volume created in the cylinder during the operation of the engine is filled with the working fluid and subjected to different thermodynamic processes. The cylinder is supported in the cylinder block.

Piston : It is a cylindrical component fitted into the cylinder forming the moving boundary of the combustion system. It fits perfectly (snugly) into the cylinder providing a gas-tight space with the piston rings and the lubricant. It forms the first link in transmitting the gas forces to the output shaft.

Combustion Chamber : The space enclosed in the upper part of the cylinder, by the cylinder head and the piston top during the combustion process, is called the combustion chamber. The combustion of fuel and the consequent release of thermal energy results in the building up of pressure in this part of the cylinder.

Inlet Manifold : The pipe which connects the intake system to the inlet valve of the engine and through which air or air-fuel mixture is drawn into the cylinder is called the inlet manifold.

Exhaust Manifold : The pipe which connects the exhaust system to the exhaust valve of the engine and through which the products of combustion escape into the atmosphere is called the exhaust manifold.

Inlet and Exhaust Valves : Valves are commonly mushroom shaped poppet type. They are provided either on the cylinder head or on the side of the cylinder for regulating the charge coming into the cylinder (inlet valve) and for discharging the products of combustion (exhaust valve) from the cylinder.

Spark Plug : It is a component to initiate the combustion process in Spark-Ignition (SI) engines and is usually located on the cylinder head.

Connecting Rod : It interconnects the piston and the crankshaft and transmits the gas forces from the piston to the crankshaft. The two ends of the connecting rod are called as small end and the big end (Fig.1.3). Small end is connected to the piston by gudgeon pin and the big end is connected to the crankshaft by crankpin.

Crankshaft : It converts the reciprocating motion of the piston into useful rotary motion of the output shaft. In the crankshaft of a single cylinder engine there are a pair of crank arms and balance weights. The balance weights are provided for static and dynamic balancing of the rotating system. The crankshaft is enclosed in a crankcase.

Piston Rings : Piston rings, fitted into the slots around the piston, provide a tight seal between the piston and the cylinder wall thus preventing leakage of combustion gases (Fig.1.3).

Gudgeon Pin : It forms the link between the small end of the connecting rod and the piston.

Camshaft : The camshaft and its associated parts control the opening and closing of the two valves. The associated parts are push rods, rocker arms, valve springs and tappets. This shaft also provides the drive to the ignition system. The camshaft is driven by the crankshaft through timing gears.

Cams : These are made as integral parts of the camshaft and are designed in such a way to open the valves at the correct timing and to keep them open for the necessary duration.

Fly Wheel : The net torque imparted to the crankshaft during one complete cycle of operation of the engine fluctuates causing a change in the angular velocity of the shaft. In order to achieve a uniform torque an inertia mass in the form of a wheel is attached to the output shaft and this wheel is called the flywheel.

1.2.2 Nomenclature

Cylinder Bore (d) : The nominal inner diameter of the working cylinder is called the cylinder bore and is designated by the letter d and is usually expressed in millimeter (mm).

Piston Area (A) : The area of a circle of diameter equal to the cylinder bore is called the piston area and is designated by the letter A and is usually expressed in square centimeter (cm^2).

Stroke (L) : The nominal distance through which a working piston moves between two successive reversals of its direction of motion is called the stroke and is designated by the letter L and is expressed usually in millimeter (mm).

Stroke to Bore Ratio : L/d ratio is an important parameter in classifying the size of the engine.

If $d < L$, it is called under-square engine. If $d = L$, it is called square engine. If $d > L$, it is called over-square engine.

An over-square engine can operate at higher speeds because of larger bore and shorter stroke.

Dead Centre : The position of the working piston and the moving parts which are mechanically connected to it, at the moment when the direction of the piston motion is reversed at either end of the stroke is called the dead centre. There are two dead centres in the engine as indicated in Fig.1.3. They are:

- (i) Top Dead Centre
- (ii) Bottom Dead Centre

(i) **Top Dead Centre (TDC) :** It is the dead centre when the piston is farthest from the crankshaft. It is designated as TDC for vertical engines and Inner Dead Centre (IDC) for horizontal engines.

(ii) **Bottom Dead Centre (BDC) :** It is the dead centre when the piston is nearest to the crankshaft. It is designated as BDC for vertical engines and Outer Dead Centre (ODC) for horizontal engines.

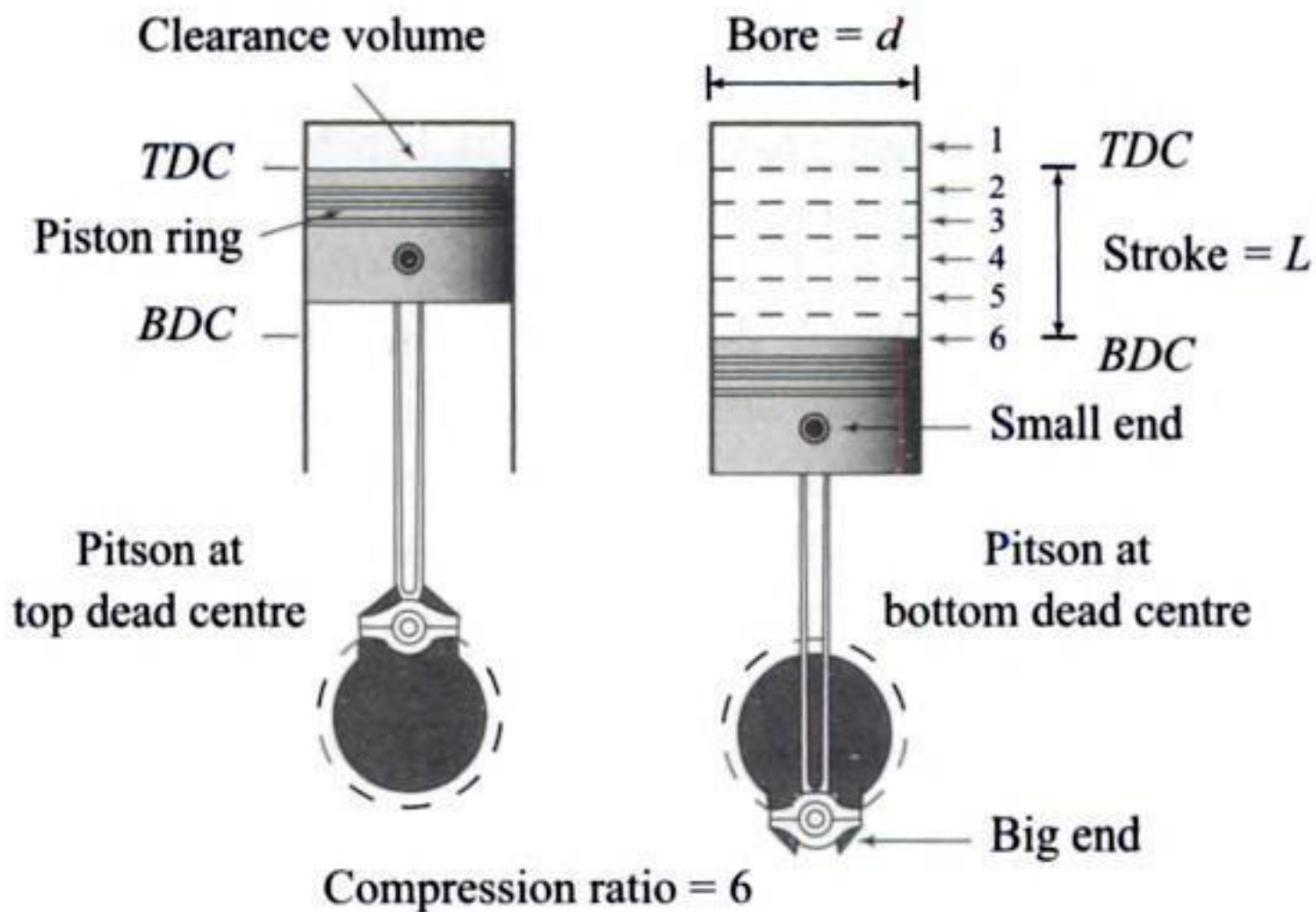


Fig. 1.3 Top and Bottom Dead Centres

Displacement or Swept Volume (V_s) : The nominal volume swept by the working piston when travelling from one dead centre to the other is called the displacement volume. It is expressed in terms of cubic centimeter (cc) and given by

$$V_s = A \times L = \frac{\pi}{4} d^2 L \quad (1.1)$$

Cubic Capacity or Engine Capacity : The displacement volume of a cylinder multiplied by number of cylinders in an engine will give the cubic capacity or the engine capacity. For example, if there are K cylinders in an engine, then

$$\text{Cubic capacity} = V_s \times K$$

Clearance Volume (V_C) : The nominal volume of the combustion chamber above the piston when it is at the top dead centre is the clearance volume. It is designated as V_C and expressed in cubic centimeter (cc).

Compression Ratio (r) : It is the ratio of the total cylinder volume when the piston is at the bottom dead centre, V_T , to the clearance volume, V_C . It is designated by the letter r .

$$r = \frac{V_T}{V_C} = \frac{V_C + V_s}{V_C} = 1 + \frac{V_s}{V_C} \quad (1.2)$$

1.3 THE WORKING PRINCIPLE OF ENGINES

If an engine is to work successfully then it has to follow a cycle of operations in a sequential manner. The sequence is quite rigid and cannot be changed. In the following sections the working principle of both SI and CI engines is described. Even though both engines have much in common there are certain fundamental differences.

The credit of inventing the spark-ignition engine goes to Nicolaus A. Otto (1876) whereas compression-ignition engine was invented by Rudolf Diesel (1892). Therefore, they are often referred to as Otto engine and Diesel engine.

1.3.1 Four-Stroke Spark-Ignition Engine

In a four-stroke engine, the cycle of operations is completed in four strokes of the piston or two revolutions of the crankshaft. During the four strokes, there are five events to be completed, viz., suction, compression, combustion, expansion and exhaust. Each stroke consists of 180° of crankshaft rotation and hence a four-stroke cycle is completed through 720° of crank rotation. The cycle of operation for an ideal four-stroke SI engine consists of the following four strokes : (i) suction or intake stroke; (ii) compression stroke; (iii) expansion or power stroke and (iv) exhaust stroke.

The details of various processes of a four-stroke spark-ignition engine with overhead valves are shown in Fig.1.4 (a-d). When the engine completes all the five events under ideal cycle mode, the p - V diagram will be as shown in Fig.1.5.

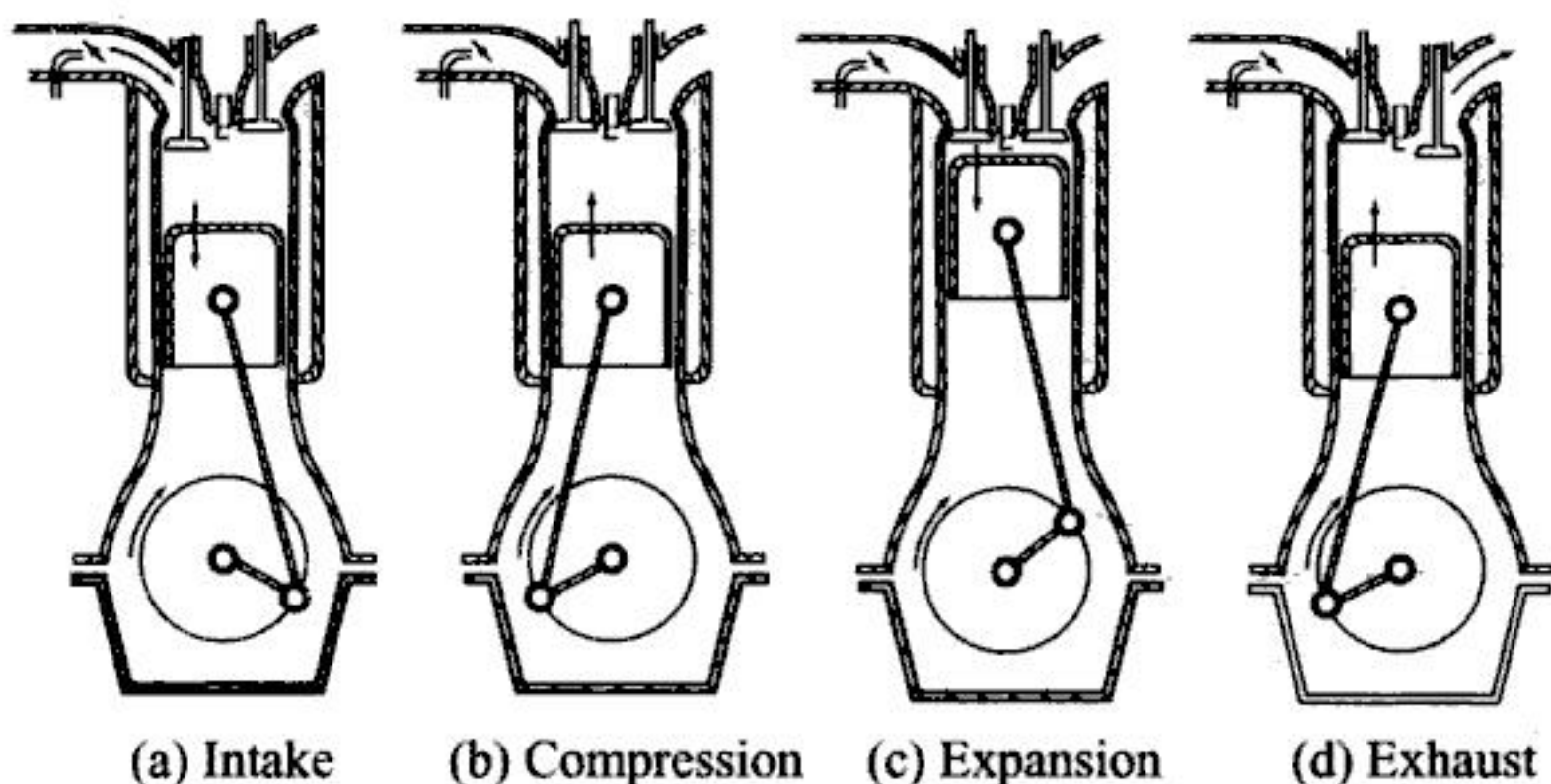


Fig. 1.4 Working Principle of a Four-Stroke SI Engine

- (i) *Suction or Intake Stroke* : Suction stroke $0 \rightarrow 1$ (Fig.1.5) starts when the piston is at the top dead centre and about to move downwards. The inlet valve is open at this time and the exhaust valve is closed, Fig.1.4(a). Due to the suction created by the motion of the piston towards the bottom dead centre, the charge consisting of fuel-air mixture is drawn into the cylinder. When the piston reaches the bottom dead centre the suction stroke ends and the inlet valve closes.
- (ii) *Compression Stroke* : The charge taken into the cylinder during the suction stroke is compressed by the return stroke of the piston $1 \rightarrow 2$, (Fig.1.5). During this stroke both inlet and exhaust valves are in closed position, Fig.1.4(b). The mixture which fills the entire cylinder volume is now compressed into the clearance volume. At the end of the compression stroke the mixture is ignited with the help of a spark plug located on the cylinder head. In ideal engines it is assumed that burning takes place instantaneously when the piston is at the top dead centre and hence the burning process can be approximated as heat addition at constant volume. During the burning process the chemical energy of the fuel is converted into heat energy producing a temperature rise of about 2000°C (process $2 \rightarrow 3$), Fig.1.5. The pressure at the end of the combustion process is considerably increased due to the heat release from the fuel.

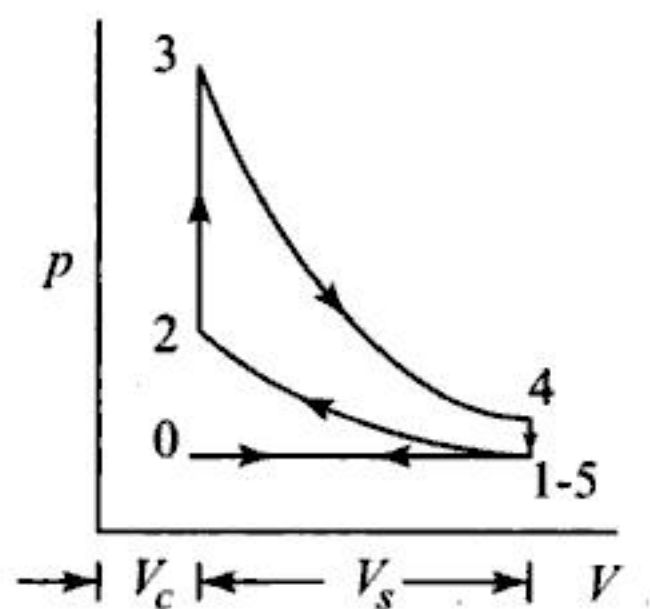


Fig. 1.5 Ideal p - V Diagram of a Four-Stroke SI Engine

- (iii) *Expansion or Power Stroke* : The high pressure of the burnt gases forces the piston towards the BDC, (stroke $3 \rightarrow 4$) Fig.1.5. Both the valves are in closed position, Fig.1.4(c). Of the four-strokes only during this stroke power is produced. Both pressure and temperature decrease during expansion.

- (iv) *Exhaust Stroke* : At the end of the expansion stroke the exhaust valve opens and the inlet valve remains closed, Fig.1.4(d). The pressure falls to atmospheric level a part of the burnt gases escape. The piston starts moving from the bottom dead centre to top dead centre (stroke 5→0), Fig.1.5 and sweeps the burnt gases out from the cylinder almost at atmospheric pressure. The exhaust valve closes when the piston reaches *TDC*. at the end of the exhaust stroke and some residual gases trapped in the clearance volume remain in the cylinder.

These residual gases mix with the fresh charge coming in during the following cycle, forming its working fluid. Each cylinder of a four-stroke engine completes the above four operations in two engine revolutions, one revolution of the crankshaft occurs during the suction and compression strokes and the second revolution during the power and exhaust strokes. Thus for one complete cycle there is only one power stroke while the crankshaft turns by two revolutions. For getting higher output from the engine the heat release (process 2→3) should be as high as possible and the heat rejection (process 3→4) should be as small as possible. So one should be careful in drawing the ideal p - V diagram (Fig.1.5).

1.3.2 Four-Stroke Compression-Ignition Engine

The four-stroke CI engine is similar to the four-stroke SI engine but it operates at a much higher compression ratio. The compression ratio of an SI engine is between 6 and 10 while for a CI engine it is from 16 to 20. In the CI engine during suction stroke, air, instead of a fuel-air mixture, is inducted. Due to the high compression ratio employed, the temperature at the end of the compression stroke is sufficiently high to self ignite the fuel which is injected into the combustion chamber. In CI engines, a high pressure fuel pump and an injector are provided to inject the fuel into the combustion chamber. The carburettor and ignition system necessary in the SI engine are not required in the CI engine.

The ideal sequence of operations for the four-stroke CI engine is as follows:

- (i) *Suction Stroke* : Air alone is inducted during the suction stroke. During this stroke intake valve is open and exhaust valve is closed, Fig.1.6(a).

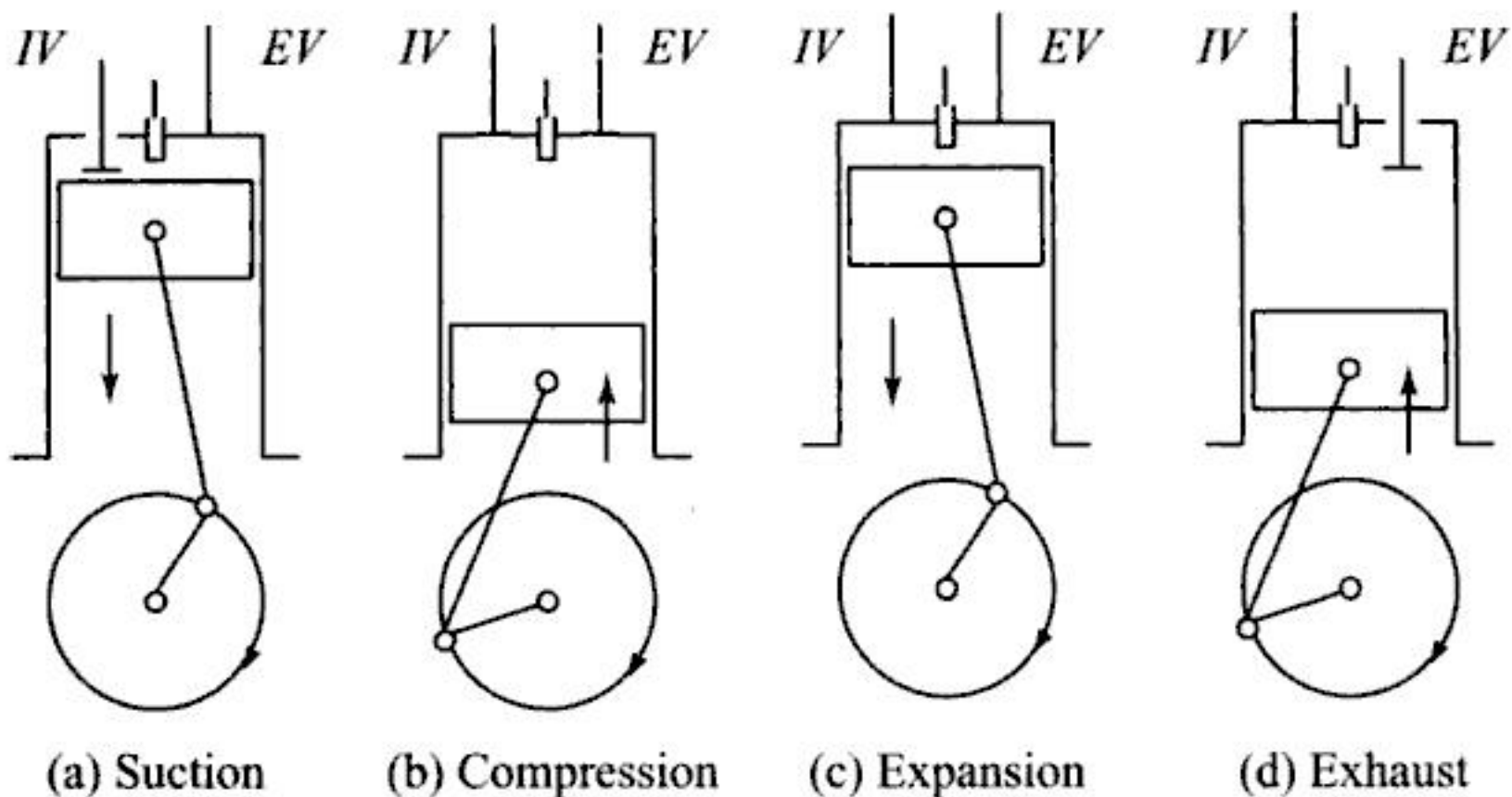


Fig. 1.6 Cycle of Operation of a CI Engine

- (ii) *Compression Stroke* : Air inducted during the suction stroke is compressed into the clearance volume. Both valves remain closed during this stroke, Fig.1.6(b).
- (iii) *Expansion Stroke* : Fuel injection starts nearly at the end of the compression stroke. The rate of injection is such that combustion maintains the pressure constant in spite of the piston movement on its expansion stroke increasing the volume. Heat is assumed to have been added at constant pressure. After the injection of fuel is completed (i.e. after cut-off) the products of combustion expand. Both the valves remain closed during the expansion stroke, Fig.1.6(c).
- (iv) *Exhaust Stroke* : The piston travelling from *BDC* to *TDC* pushes out the products of combustion. The exhaust valve is open and the intake valve is closed during this stroke, Fig.1.6(d). The ideal p - V diagram is shown in Fig.1.7.

Due to higher pressures in the cycle of operations the CI engine has to be more sturdy than a SI engine for the same output. This results in a CI engine being heavier than the SI engine. However, it has a higher thermal efficiency on account of the high compression ratio (of about 18 as against about 8 in SI engines) used.

1.3.3 Comparison of SI and CI Engines

In four-stroke engines, there is one power stroke for every two revolutions of the crankshaft. There are two non-productive strokes of

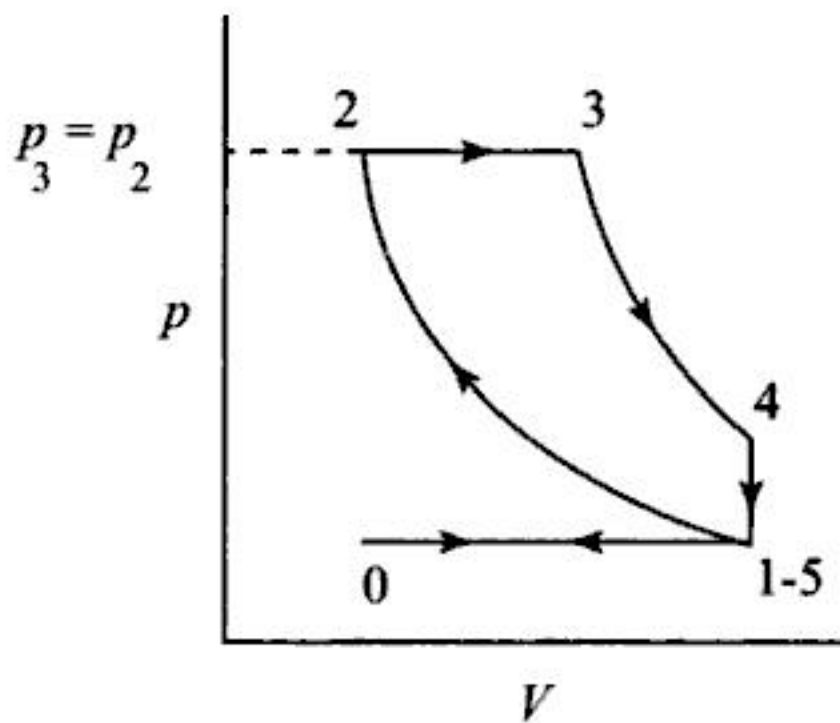


Fig. 1.7 Ideal p - V Diagram for a Four-Stroke CI Engine

exhaust and suction which are necessary for flushing the products of combustion from the cylinder and filling it with the fresh charge. If this purpose could be served by an alternative arrangement, without the movement of the piston, it is possible to obtain a power stroke for every revolution of the crankshaft increasing the output of the engine. However, in both SI and CI engines operating on four-stroke cycle, power can be obtained only in every two revolution of the crankshaft.

Since both SI and CI engines have much in common, it is worthwhile to compare them based on important parameters like basic cycle of operation, fuel induction, compression ratio etc. The detailed comparison is given in Table 1.1.

1.3.4 Two-Stroke Engine

As already mentioned, if the two unproductive strokes, viz., the suction and exhaust could be served by an alternative arrangement, especially without the movement of the piston then there will be a power stroke for each revolution of the crankshaft. In such an arrangement, theoretically the power output of the engine can be doubled for the same speed compared to a four-stroke engine. Based on this concept, Dugald Clark (1878) invented the two-stroke engine.

In two-stroke engines the cycle is completed in one revolution of the crankshaft. The main difference between two-stroke and four-stroke engines is in the method of filling the fresh charge and removing the burnt gases from the cylinder. In the four-stroke engine these operations are performed by the engine piston during the suction and exhaust strokes respectively. In a two-stroke engine, the filling pro-

Table 1.1 Comparison of SI and CI Engines

Description	SI Engine	CI Engine
Basic cycle	Works on Otto cycle or constant volume heat addition cycle.	Works on Diesel cycle or constant pressure heat addition cycle.
Fuel	Gasoline, a highly volatile fuel. Self-ignition temperature is high.	Diesel oil, a non-volatile fuel. Self-ignition temperature is comparatively low.
Introduction of fuel	A gaseous mixture of fuel-air is introduced during the suction stroke. A carburettor and an ignition system are necessary. Modern engines have gasoline injection.	Fuel is injected directly into the combustion chamber at high pressure at the end of the compression stroke. A fuel pump and injector are necessary.
Load control	Throttle controls the quantity of fuel-air mixture introduced.	The quantity of fuel is regulated. Air quantity is not controlled.
Ignition	Requires an ignition system with spark plug in the combustion chamber. Primary voltage is provided by either a battery or a magneto.	Self-ignition occurs due to high temperature of air because of the high compression. Ignition system and spark plug are not necessary.
Compression ratio	6 to 10. Upper limit is fixed by antiknock quality of the fuel.	16 to 20. Upper limit is limited by weight increase of the engine.
Speed	Due to light weight and also due to homogeneous combustion, they are high speed engines.	Due to heavy weight and also due to heterogeneous combustion, they are low speed engines.
Thermal efficiency	Because of the lower CR , the maximum value of thermal efficiency that can be obtained is lower.	Because of higher CR , the maximum value of thermal efficiency that can be obtained is higher.
Weight	Lighter due to lower peak pressures.	Heavier due to higher peak pressures.

cess is accomplished by the charge compressed in crankcase or by a blower. The induction of the compressed charge moves out the product of combustion through exhaust ports. Therefore, no piston strokes are required for these two operations. Two strokes are sufficient to complete the cycle, one for compressing the fresh charge and the other for expansion or power stroke.

Figure 1.8 shows one of the simplest two-stroke engines, viz., the crankcase scavenged engine. Figure 1.9 shows the ideal indicator diagram of such an engine. The air or charge is inducted into the crankcase through the spring loaded inlet valve when the pressure in the crankcase is reduced due to upward motion of the piston during compression stroke. After the compression and ignition, expansion takes place in the usual way.

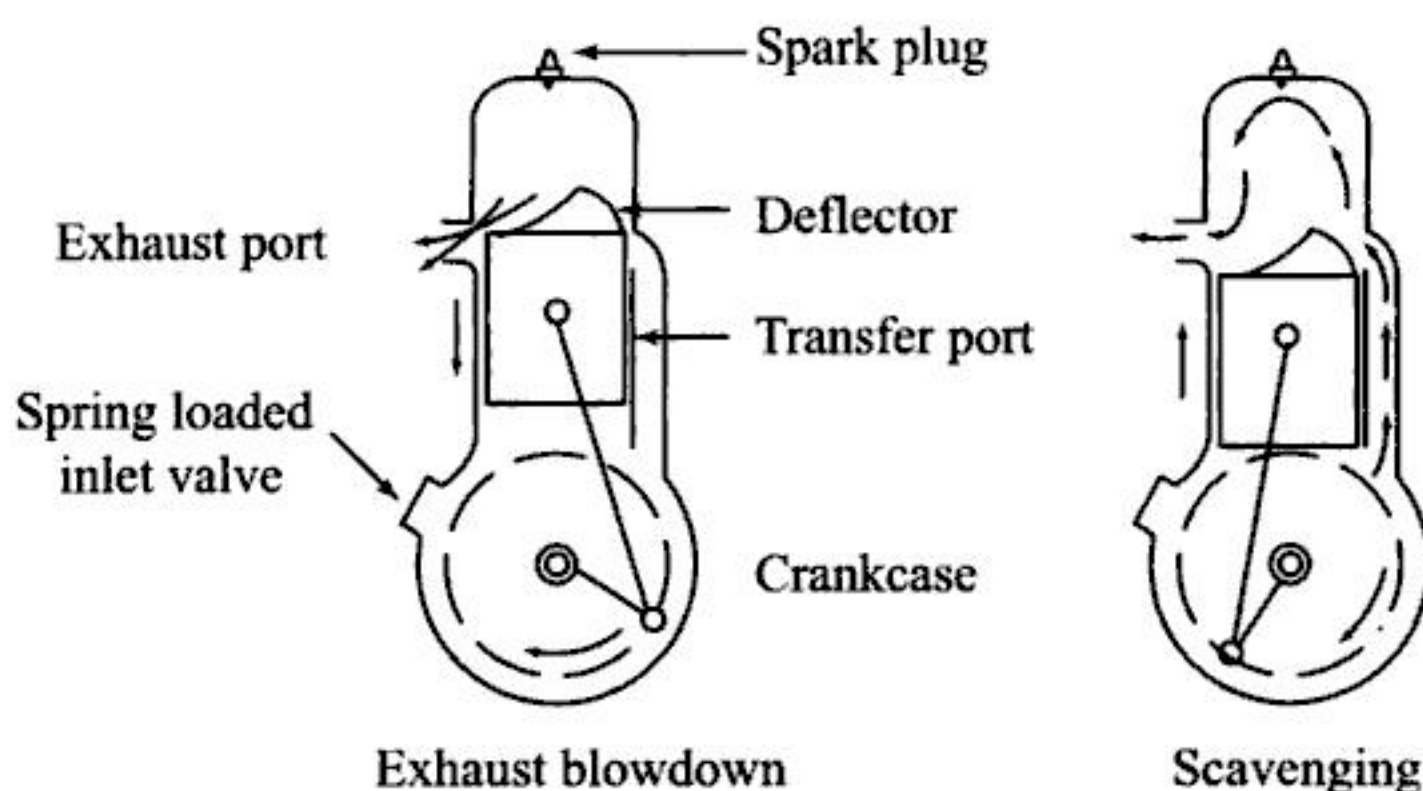


Fig. 1.8 Crankcase Scavenged Two-Stroke Engine

During the expansion stroke the charge in the crankcase is compressed. Near the end of the expansion stroke, the piston uncovers the exhaust ports and the cylinder pressure drops to atmospheric pressure as the combustion products leave the cylinder. Further movement of the piston uncovers the transfer ports, permitting the slightly compressed charge in the crankcase to enter the engine cylinder. The top of the piston has usually a projection to deflect the fresh charge towards the top of the cylinder before flowing to the exhaust ports. This serves the double purpose of scavenging the upper part of the cylinder of the combustion products and preventing the fresh charge from flowing directly to the exhaust ports.

The same objective can be achieved without piston deflector by proper shaping of the transfer port. During the upward motion of the

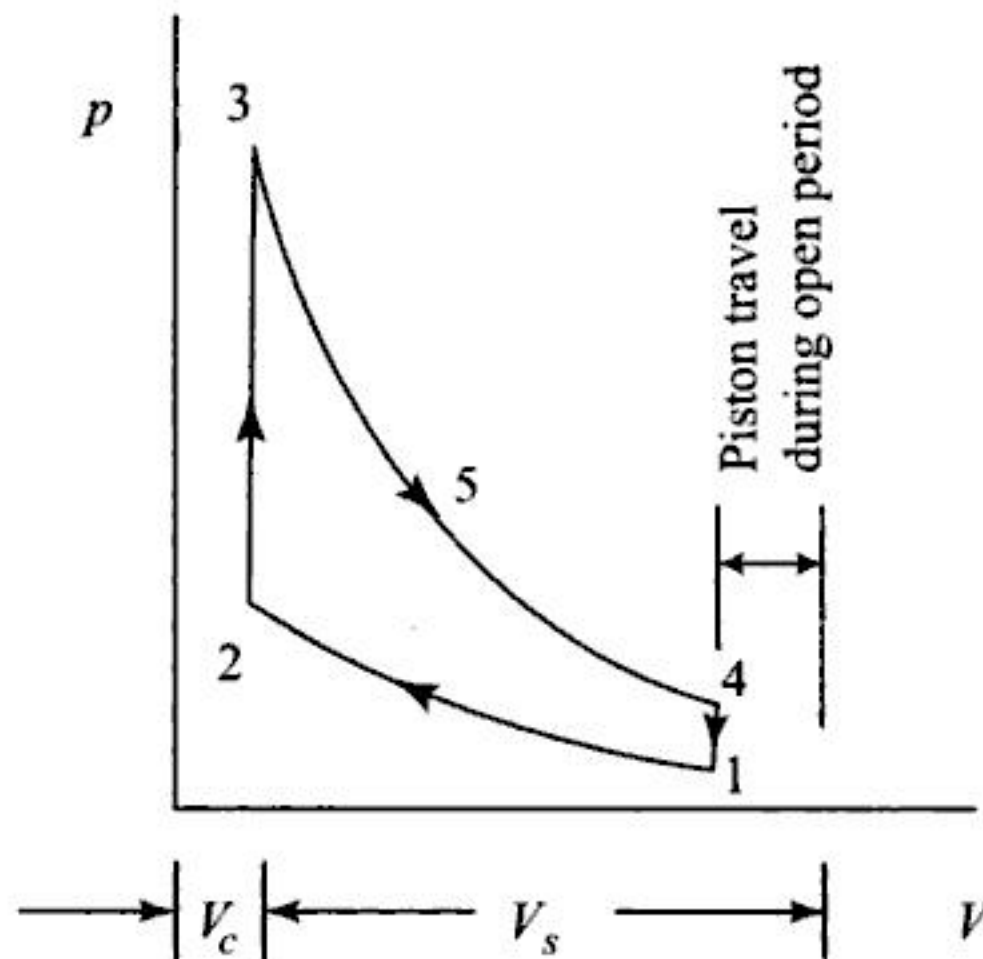


Fig. 1.9 Ideal Indicator Diagram of a Two-Stroke SI Engine

piston from *BDC* the transfer ports close first and then the exhaust ports close when compression of the charge begins and the cycle is repeated.

1.3.5 Comparison of Four-Stroke and Two-Stroke Engines

The two-stroke engine was developed to obtain a greater output from the same size of the engine. The engine mechanism also eliminates the valve arrangement making it mechanically simpler. Almost all two-stroke engines have no conventional valves but only ports (some have an exhaust valve). This simplicity of the two-stroke engine makes it cheaper to produce and easy to maintain. Theoretically a two-stroke engine develops twice the power of a comparable four-stroke engine because of one power stroke every revolution (compared to one power stroke every two revolutions of a four-stroke engine). This makes the two-stroke engine more compact than a comparable four-stroke engine. In actual practice power output is not exactly doubled but increased by only about 30% because of

- (i) reduced effective expansion stroke and
- (ii) increased heating caused by increased number of power strokes which limits the maximum speed.

The other advantages of the two-stroke engine are more uniform torque on crankshaft and comparatively less exhaust gas dilution.

However, when applied to the spark-ignition engine the two-stroke cycle has certain disadvantages which have restricted its application to only small engines suitable for motor cycles, scooters, lawn mowers, outboard engines etc. In the SI engine, the incoming charge consists of fuel and air. During scavenging, as both inlet and exhaust ports are open simultaneously for some time, there is a possibility that some of the fresh charge containing fuel escapes with the exhaust. This results in high fuel consumption and lower thermal efficiency. The other drawback of two-stroke engine is the lack of flexibility, viz., the capacity to operate with the same efficiency at all speeds. At part throttle operating condition, the amount of fresh mixture entering the cylinder is not enough to clear all the exhaust gases and a part of it remains in the cylinder to contaminate the charge. This results in irregular operation of the engine.

The two-stroke diesel engine does not suffer from these defects. There is no loss of fuel with exhaust gases as the intake charge in diesel engine is only air. The two-stroke diesel engine is used quite widely. Many of the high output diesel engines work on this cycle. A disadvantage common to all two-stroke engines, gasoline as well as diesel, is the greater cooling and lubricating oil requirements due to one power stroke in each revolution of the crankshaft. Consumption of lubricating oil is high in two-stroke engines due to higher temperature. A detailed comparison of two-stroke and four-stroke engines is given in Table 1.2.

1.4 ACTUAL ENGINES

Actual engines differ from the ideal engines because of various constraints in their operation. The indicator diagram also differs considerably from the ideal indicator diagrams. Actual indicator diagrams of a two-stroke and a four-stroke SI engines are shown in Figs.1.10(a) and 1.10(b) respectively. The various processes are indicated in the respective figures.

1.5 CLASSIFICATION OF IC ENGINES

Internal combustion engines are usually classified on the basis of the thermodynamic cycle of operation, type of fuel used, method of charging the cylinder, type of ignition, type of cooling and the cylinder arrangement etc. Details are given in Fig.1.11.

Table 1.2 Comparison of Four and Two-Stroke Cycle Engines

Four-Stroke Engine	Two-Stroke Engine
The thermodynamic cycle is completed in four strokes of the piston or in two revolutions of the crankshaft. Thus, one power stroke is obtained in every two revolutions of the crankshaft.	The thermodynamic cycle is completed in two strokes of the piston or in one revolution of the crankshaft. Thus one power stroke is obtained in each revolution of the crankshaft.
Because of the above, turning moment is not so uniform and hence a heavier flywheel is needed.	Because of the above, turning moment is more uniform and hence a lighter flywheel can be used.
Again, because of one power stroke for two revolutions, power produced for same size of engine is less, or for the same power the engine is heavier and bulkier.	Because of one power stroke for every revolution, power produced for same size of engine is twice, or for the same power the engine is lighter and more compact.
Because of one power stroke in two revolutions lesser cooling and lubrication requirements. Lower rate of wear and tear.	Because of one power stroke in one revolution greater cooling and lubrication requirements. Higher rate of wear and tear.
Four-stroke engines have valves and valve actuating mechanisms for opening and closing of the intake and exhaust valves.	Two-stroke engines have no valves but only ports (some two-stroke engines are fitted with conventional exhaust valve or reed valve).
Because of comparatively higher weight and complicated valve mechanism, the initial cost of the engine is more.	Because of light weight and simplicity due to the absence of valve actuating mechanism, initial cost of the engine is less.
Volumetric efficiency is more due to more time for induction.	Volumetric efficiency is low due to lesser time for induction.
Thermal efficiency is higher; part load efficiency is better.	Thermal efficiency is lower; part load efficiency is poor.
Used where efficiency is important, viz., in cars, buses, trucks, tractors, industrial engines, aeroplanes, power generation etc.	Used where low cost, compactness and light weight are important, viz., in mopeds, scooters, motorcycles, hand sprayers etc.

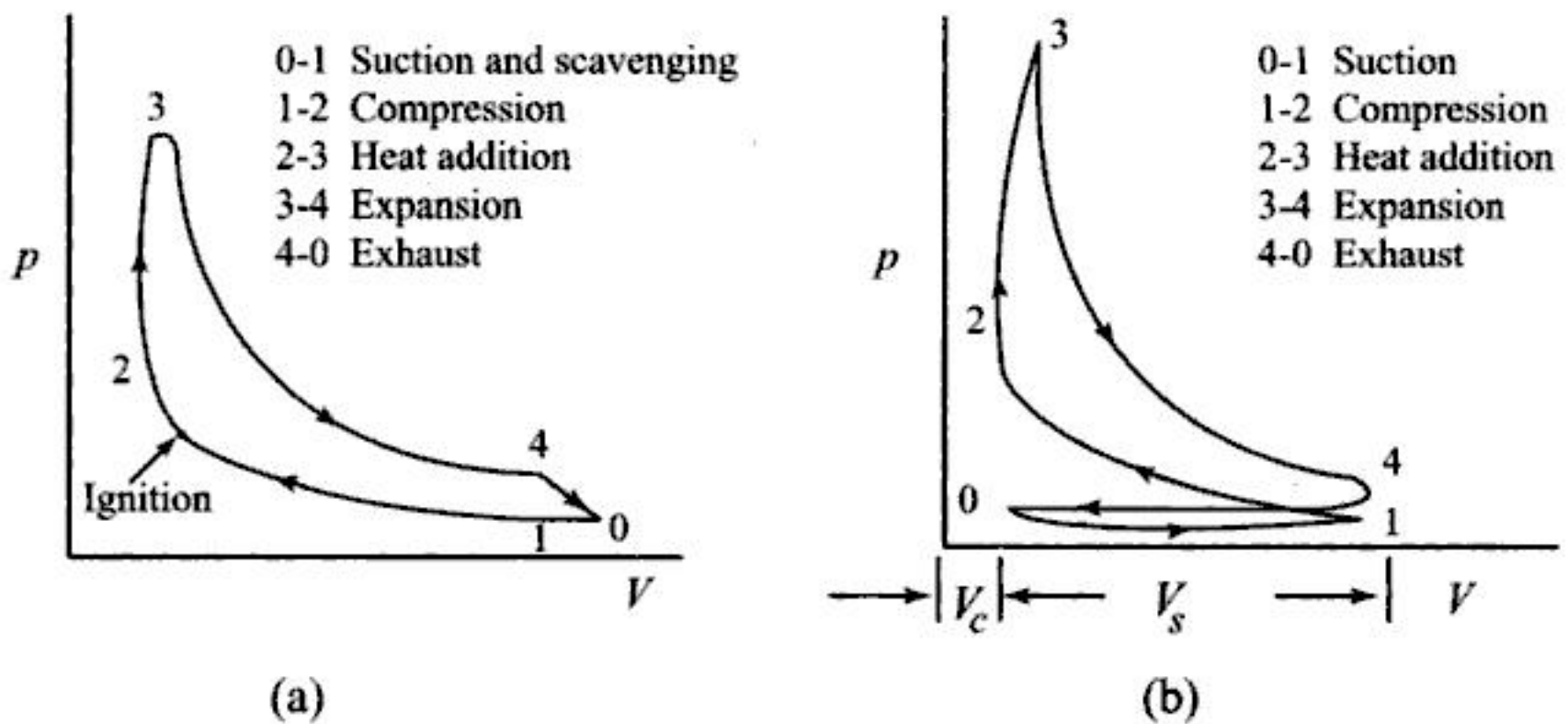


Fig. 1.10 Actual Indicator Diagrams of a Two-Stroke and Four-Stroke SI Engine

1.5.1 Cycle of Operation

According to the cycle of operation, IC engines are basically classified into two categories

- (i) Constant volume heat addition cycle engine or Otto cycle engine. It is also called a Spark-Ignition engine, SI engine or Gasoline engine .
- (ii) Constant-pressure heat addition cycle engine or Diesel cycle engine. It is also called a compression-ignition engine, CI engine or Diesel engine .

1.5.2 Type of Fuel Used

Based on the type of fuel used engines are classified as

- (i) Engines using volatile liquid fuels like gasoline, alcohol, kerosene, benzene etc.
The fuel is generally mixed with air to form a homogeneous charge in a carburettor outside the cylinder and drawn into the cylinder in its suction stroke. The charge is ignited near the end of the compression stroke by an externally applied spark and therefore these engines are called spark-ignition engines.
- (ii) Engines using gaseous fuels like natural gas, Liquefied Petroleum Gas (LPG), blast furnace gas and biogas.

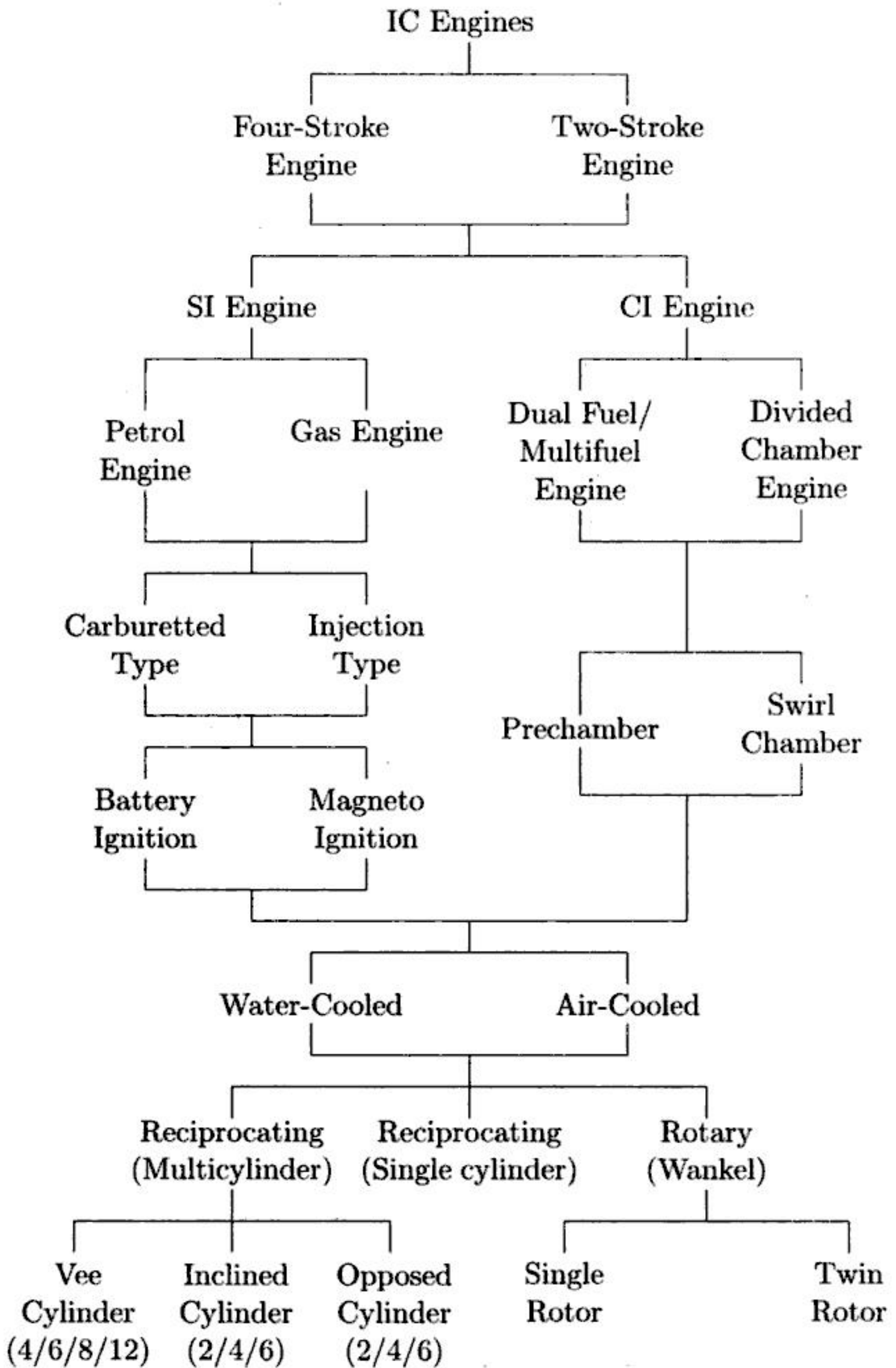


Fig. 1.11 Classification of Internal Combustion Engines

The gas is mixed with air and the mixture is introduced into the cylinder during the suction process. Working of this type of engine is similar to that of the engines using volatile liquid fuels (SI gas engine).

- (iii) Engine using solid fuels like charcoal, powdered coal etc.
Solid fuels are generally converted into gaseous fuels outside the engine in a separate gas producer and the engine works as a gas engine.
- (iv) Engines using viscous (low volatility at normal atmospheric temperatures) liquid fuels like heavy and light diesel oils.
The fuel is generally introduced into the cylinder in the form of minute droplets by a fuel injection system near the end of the compression process. Combustion of the fuel takes place due to its coming into contact with the high temperature compressed air in the cylinder. Therefore, these engines are called compression-ignition engines.
- (v) Engines using two fuels (dual-fuel engines)
A gaseous fuel or a highly volatile liquid fuel is supplied along with air during the suction stroke or during the initial part of compression through a gas valve in the cylinder head and the other fuel (a viscous liquid fuel) is injected into the combustion space near the end of the compression stroke (dual-fuel engines).

1.5.3 Method of Charging

According to the method of charging, the engines are classified as

- (i) Naturally aspirated engines : Admission of air or fuel-air mixture at near atmospheric pressure.
- (ii) Supercharged Engines : Admission of air or fuel-air mixture under pressure, i.e., above atmospheric pressure.

1.5.4 Type of Ignition

Spark-ignition engines require an external source of energy for the initiation of spark and thereby the combustion process. A high voltage spark is made to jump across the spark plug electrodes. In order to produce the required high voltage there are two types of ignition systems which are normally used. They are :

- (i) battery ignition system (ii) magneto ignition system.

They derive their name based on whether a battery or a magneto is used as the primary source of energy for producing the spark.

In the case of CI engines there is no need for an external means to produce the ignition. Because of high compression ratio employed, the resulting temperature at the end of the compression process is high enough to self-ignite the fuel when injected. However, the fuel should be atomized into very fine particles. For this purpose a fuel injection system is normally used.

1.5.5 Type of Cooling

Cooling is very essential for the satisfactory running of an engine. There are two types of cooling systems in use and accordingly, the engines is classified as

- (i) air-cooled engine (ii) water-cooled engine

1.5.6 Cylinder Arrangements

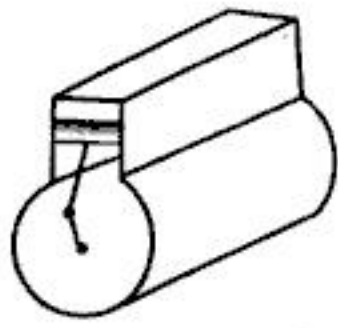
Another common method of classifying reciprocating engines is by the cylinder arrangement. The cylinder arrangement is only applicable to multicylinder engines. Two terms used in connection with cylinder arrangements must be defined first.

- (i) *Cylinder Row* : An arrangement of cylinders in which the centre-line of the crankshaft journals is perpendicular to the plane containing the centrelines of the engine cylinders.
- (ii) *Cylinder Bank* : An arrangement of cylinders in which the centre-line of the crankshaft journals is parallel to the plane containing the centrelines of the engine cylinders.

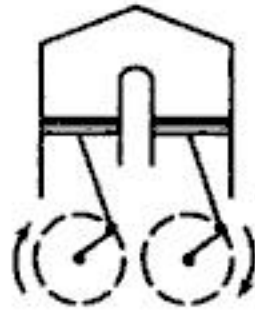
A number of cylinder arrangements popular with designers are described below. The details of various cylinder arrangements are shown in Fig.1.12.

In-line Engine : The in-line engine is an engine with one cylinder bank, i.e. all cylinders are arranged linearly, and transmit power to a single crankshaft. This type is quite common with automobile engines. Four and six cylinder in-line engines are popular in automotive applications.

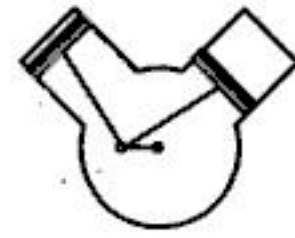
'V' Engine : In this engine there are two banks of cylinders (i.e., two in line engines) inclined at an angle to each other and with one crankshaft. Most of the high powered automobiles use the 8 cylinder 'V' engine, four in-line on each side of the 'V'. Engines with more than six cylinders generally employ this configuration.



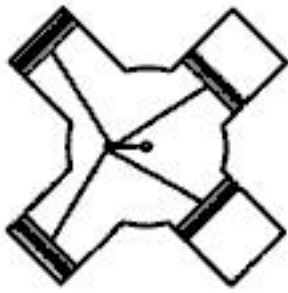
In-line



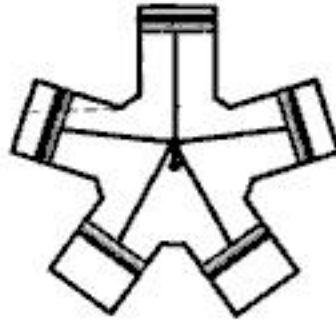
U-cylinder



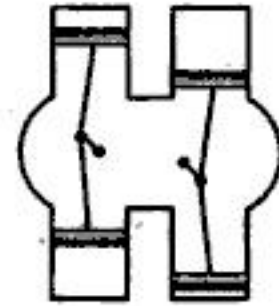
V-type



X-type



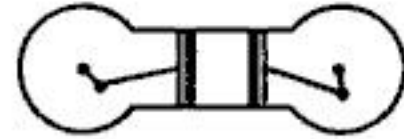
Radial



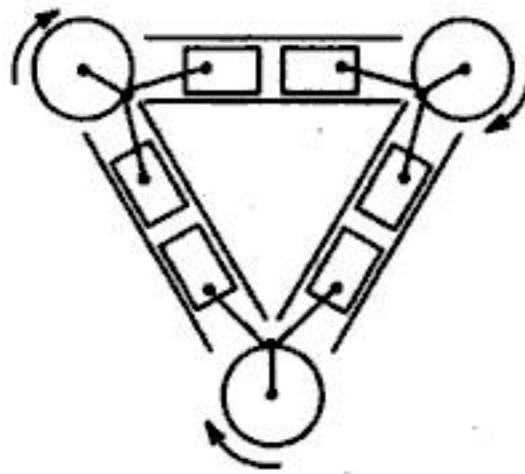
H-type



Opposed cylinder



Opposed piston



Delta type

Fig. 1.12 Engine Classification by Cylinder Arrangements

Opposed Cylinder Engine : This engine has two cylinder banks located in the same plane on opposite sides of the crankshaft. It can be visualized as two 'in-line' arrangements 180 degrees apart. It is inherently a well balanced engine and has the advantages of a single crankshaft. This design is used in small aircrafts.

Opposed Piston Engine : When a single cylinder houses two pistons, each of which driving a separate crankshaft, it is called an opposed piston engine. The movement of the pistons is synchronized by coupling the two crankshafts. Opposed piston arrangement, like opposed cylinder arrangement, is inherently well balanced. Further, it has the advantage of requiring no cylinder head. By its inherent features, this engine usually functions on the principle of two-stroke engines.

Radial Engine : Radial engine is one where more than two cylinders in each row are equally spaced around the crankshaft. The radial arrangement of cylinders is most commonly used in conventional air-cooled aircraft engines where 3, 5, 7 or 9 cylinders may be used in one bank and two to four banks of cylinders may be used. The odd number of cylinders is employed from the point of view of balancing. Pistons of all the cylinders are coupled to the same crankshaft.

‘X’ Type Engine : This design is a variation of ‘V’ type. It has four banks of cylinders attached to a single crankshaft.

‘H’ Type Engine : The ‘H’ type is essentially two ‘Opposed cylinder’ type utilizing two separate but interconnected crankshafts.

‘U’ Type Engine : The ‘U’ type is a variation of opposed piston arrangement.

Delta Type Engine : The delta type is essentially a combination of three opposed piston engine with three crankshafts interlinked to one another.

In general, automobile engines and general purpose engines utilize the ‘in-line’ and ‘V’ type configuration or arrangement. The ‘radial’ engine was used widely in medium and large aircrafts till it was replaced by the gas turbine. Small aircrafts continue to use either the ‘opposed cylinder’ type or ‘in-line’ or ‘V’ type engines. The ‘opposed piston’ type engine is widely used in large diesel installations. The ‘H’ and ‘X’ types do not presently find wide application, except in some diesel installations. A variation of the ‘X’ type is referred to as the ‘pancake’ engine.

1.6 APPLICATION OF IC ENGINES

The most important application of IC engines is in transport on land, sea and air. Other applications include industrial power plants and as prime movers for electric generators. Table 1.3 gives, in a nutshell, the applications of both IC and EC engines.

1.6.1 Two-Stroke Gasoline Engines

Small two-stroke gasoline engines are used where simplicity and low cost of the prime mover are the main considerations. In such applications a little higher fuel consumption is acceptable. The smallest engines are used in mopeds (50 cc engine) and lawn mowers. Scooters and motor cycles, the commonly used two wheeler transport, have generally 100-150 cc, two-stroke gasoline engines developing a maximum brake power of about 5 kW at 5500 rpm. High powered motor

Table 1.3 Application of Engines

IC Engine		EC Engine	
Type	Application	Type	Application
Gasoline engines	Automotive, Marine, Aircraft	Steam Engines	Locomotives, Marine
Gas engines	Industrial power	Stirling Engines	Experimental Space Vehicles
Diesel engines	Automotive, Locomotive, Power, Marine	Steam Turbines	Power, Large Marine
Gas turbines	Power, Aircraft, Industrial, Marine	Close Cycle Gas Turbine	Power, Marine

cycles have generally 250 cc two-stroke gasoline engines developing a maximum brake power of about 10 kW at 5000 rpm. Two-stroke gasoline engines may also be used in very small electric generating sets, pumping sets, and outboard motor boats. However, their specific fuel consumption is higher due to the loss of fuel-air charge in the process of scavenging and because of high speed of operation for which such small engines are designed.

1.6.2 Two-Stroke Diesel Engines

Very high power diesel engines used for ship propulsion are commonly two-stroke diesel engines. In fact, all engines between 400 to 900 mm bore are loop scavenged or uniflow type with exhaust valves (see Figs.20.8 and 20.9). The brake power on a single crankshaft can be upto 37000 kW. Nordberg, 12 cylinder 800 mm bore and 1550 mm stroke, two-stroke diesel engine develops 20000 kW at 120 rpm. This speed allows the engine to be directly coupled to the propeller of a ship without the necessity of gear reducers.

1.6.3 Four-Stroke Gasoline Engines

The most important application of small four-stroke gasoline engines is in automobiles. A typical automobile is powered by a four-stroke four cylinder engine developing an output in the range of 30-60 kW at a speed of about 4500 rpm. American automobile engines are much bigger and have 6 or 8 cylinder engines with a power output upto

185 kW. However, the oil crisis and air pollution from automobile engines have reversed this trend towards smaller capacity cars.

Four-stroke gasoline engines were also used for buses and trucks. They were generally 4000 cc, 6 cylinder engines with maximum brake power of about 90 kW. However, in this application gasoline engines have been practically replaced by diesel engines. The four-stroke gasoline engines have also been used in big motor cycles with side cars. Another application of four-stroke gasoline engine is in small pumping sets and mobile electric generating sets.

Small aircraft generally use radial four-stroke gasoline engines. Engines having maximum power output from 400 kW to 4000 kW have been used in aircraft. An example is the Bristol Contours 57, 18 cylinder two row, sleeve valve, air-cooled radial engine developing, a maximum brake power of about 2100 kW.

1.6.4 Four-Stroke Diesel Engines

The four-stroke diesel engine is one of the most efficient and versatile prime movers. It is manufactured in sizes from 50 mm to more than 1000 mm of cylinder diameter and with engine speeds ranging from 100 to 4500 rpm while delivering outputs from 1 to 35000 kW.

Small diesel engines are used in pump sets, construction machinery, air compressors, drilling rigs and many miscellaneous applications. Tractors for agricultural application use about 30 kW diesel engines whereas jeeps, buses and trucks use 40 to 100 kW diesel engines. Generally, the diesel engines with higher outputs than about 100 kW are supercharged. Earth moving machines use supercharged diesel engines in the output range of 200 to 400 kW. Locomotive applications require outputs of 600 to 4000 kW. Marine applications, from fishing vessels to ocean going ships use diesel engines from 100 to 35000 kW. Diesel engines are used both for mobile and stationary electric generating plants of varying capacities. Compared to gasoline engines, diesel engines are more efficient and therefore manufacturers have come out with diesel engines in personal transportation. However, the vibrations from the engine and the unpleasant odour in the exhaust are the main drawbacks.

1.7 THE FIRST LAW ANALYSIS OF ENGINE CYCLE

Before a detailed thermodynamic analysis of the engine cycle is done, it is desirable to have a general picture of the energy flow or energy balance of the system so that one becomes familiar with the various

performance parameters. Figure 1.13 shows the energy flow through the reciprocating engine and Fig.1.14 shows its block diagram as an open system.

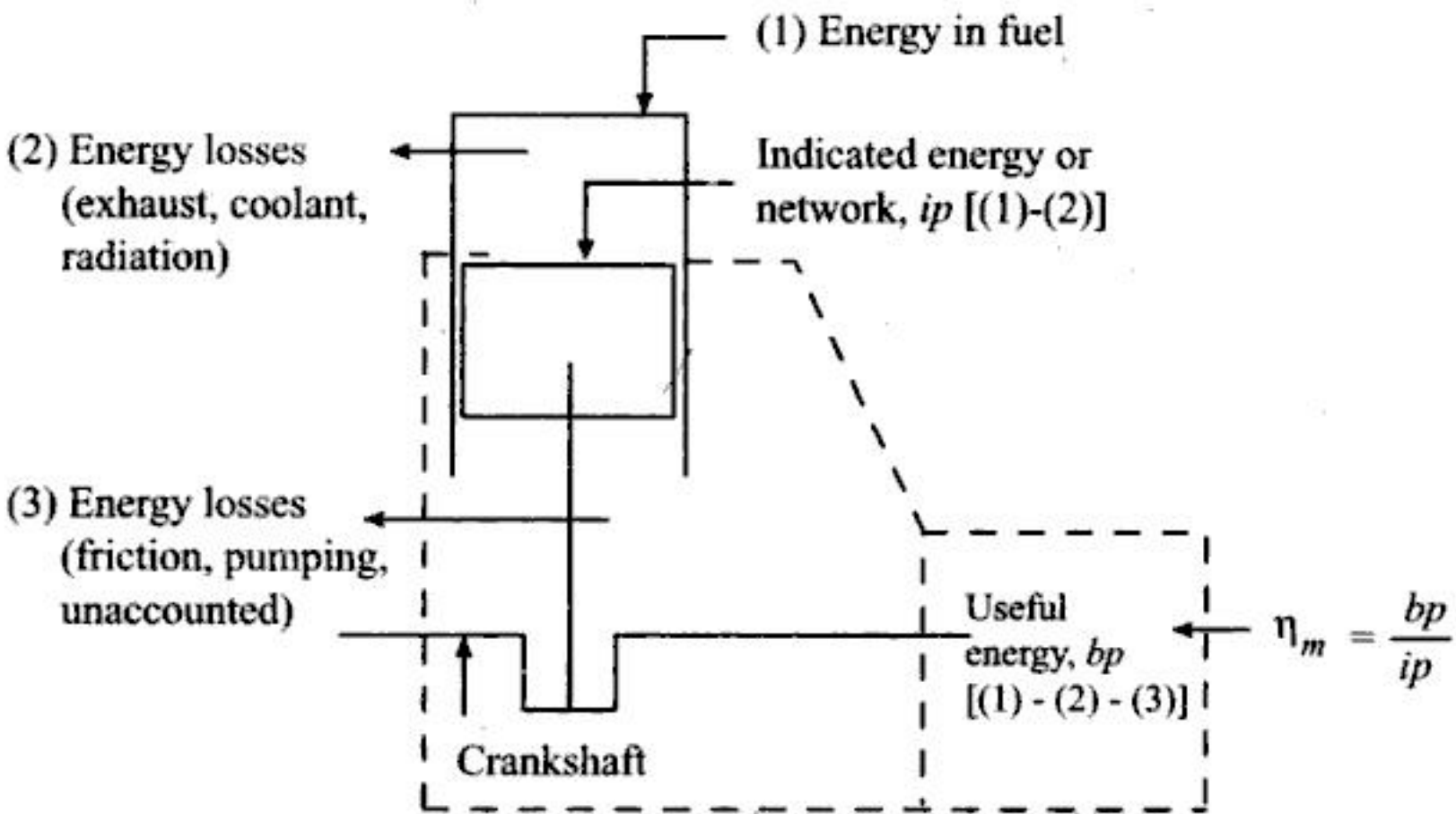


Fig. 1.13 Energy Flow through the Reciprocating Engine

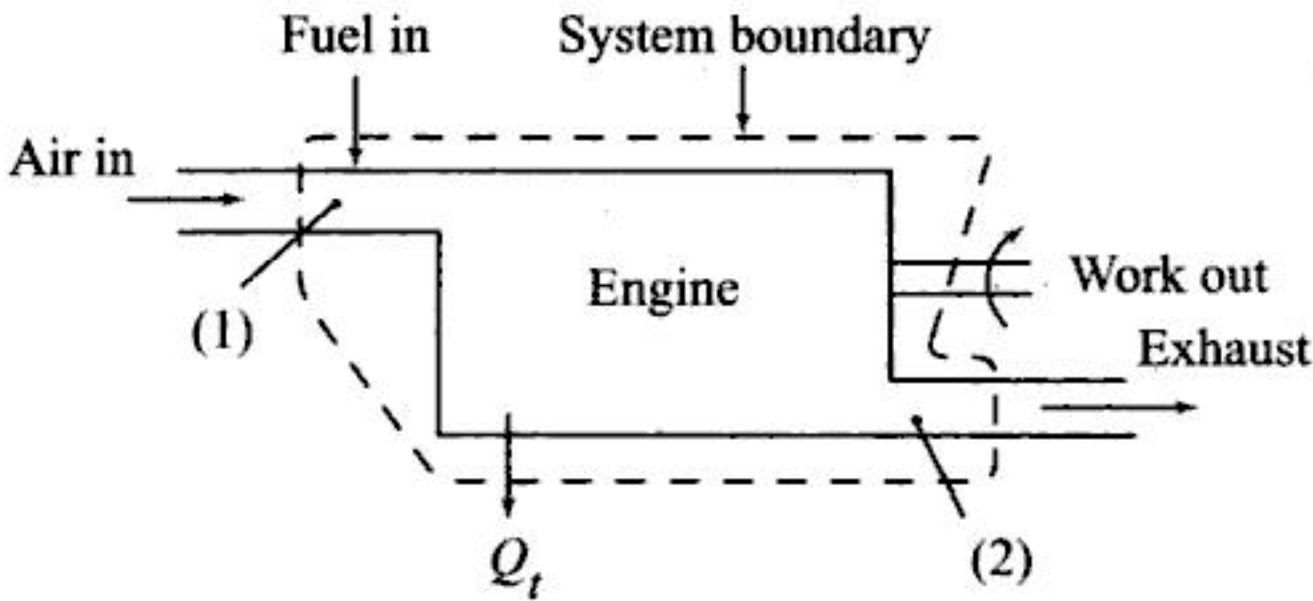


Fig. 1.14 Reciprocating Engine as an Open System

According to the first law of thermodynamics, energy can neither be created nor destroyed. It can only be converted from one form to another. Therefore, there must be an energy balance of input and output to a system. In the reciprocating internal combustion engine the fuel is fed into the combustion chamber where it burns in air converting chemical energy of the fuel into heat. The liberated heat energy cannot be totally utilized for driving the piston as there are losses through the engine exhaust, to the coolant and due to radiation. The heat energy which is converted to power at this stage

is called the indicated power, ip and it is utilized to drive the piston. The energy represented by the gas forces on the piston passes through the connecting rod to the crankshaft. In this transmission there are energy losses due to bearing friction, pumping losses etc. In addition, a part of the energy available is utilized in driving the auxiliary devices like feed pump, valve mechanisms, ignition systems etc. The sum of all these losses, expressed in units of power is termed as frictional power, fp . The remaining energy is the useful mechanical energy and is termed as the brake power, bp . In energy balance, generally, frictional power is not shown separately because ultimately this energy is accounted in exhaust, cooling water, radiation, etc.

1.8 ENGINE PERFORMANCE PARAMETERS

The engine performance is indicated by the term *efficiency*, η . Five important engine efficiencies and other related engine performance parameters are given below:

(i)	Indicated thermal efficiency	(η_{ith})
(ii)	Brake thermal efficiency	(η_{bth})
(iii)	Mechanical efficiency	(η_m)
(iv)	Volumetric efficiency	(η_v)
(v)	Relative efficiency or Efficiency ratio	(η_{rel})
(vi)	Mean effective pressure	(p_m)
(vii)	Mean piston speed	$(\bar{s}_p)$
(viii)	Specific power output	(P_s)
(ix)	Specific fuel consumption	(sfc)
(x)	Inlet-valve Mach Index	(Z)
(x)	Fuel-air or air-fuel ratio	$(F/A \text{ or } A/F)$
(xi)	Calorific value of the fuel	(CV)

Figure 1.15 shows the diagrammatic representation of energy distribution in an IC engine.

1.8.1 Indicated Thermal Efficiency (η_{ith})

Indicated thermal efficiency is the ratio of energy in the indicated power, ip , to the input fuel energy in appropriate units.

$$[ht]\eta_{ith} = \frac{ip \text{ [kJ/s]}}{\text{energy in fuel per second [kJ/s]}} \quad (1.3)$$

$$= \frac{ip}{\text{mass of fuel/s} \times \text{calorific value of fuel}} \quad (1.4)$$

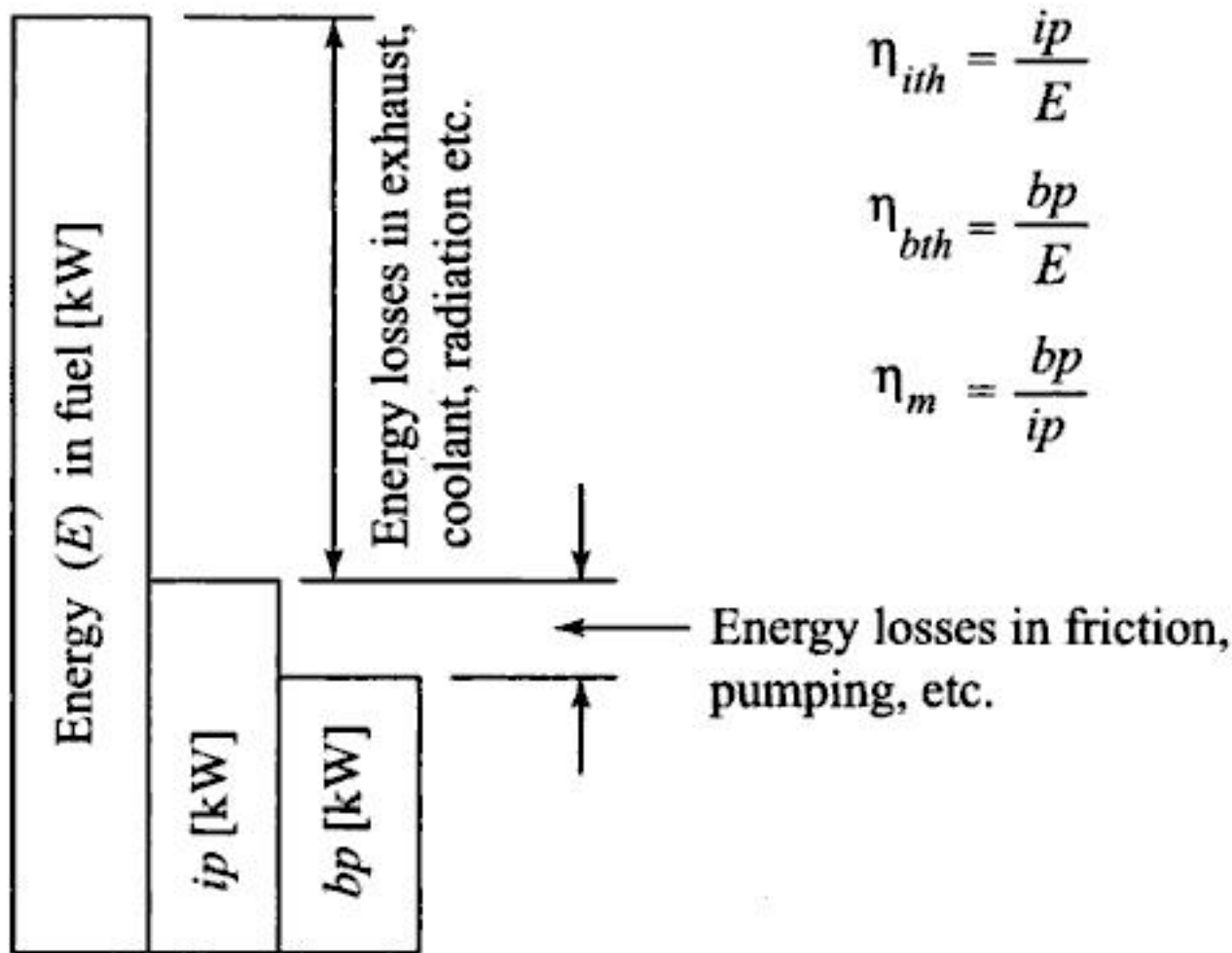


Fig. 1.15 Energy Distribution

1.8.2 Brake Thermal Efficiency (η_{bth})

Brake thermal efficiency is the ratio of energy in the brake power, bp , to the input fuel energy in appropriate units.

$$\eta_{bth} = \frac{bp}{\text{Mass of fuel/s} \times \text{calorific value of fuel}} \quad (1.5)$$

1.8.3 Mechanical Efficiency (η_m)

Mechanical efficiency is defined as the ratio of brake power (delivered power) to the indicated power (power provided to the piston).

$$\eta_m = \frac{bp}{ip} = \frac{bp}{bp + fp} \quad (1.6)$$

$$fp = ip - bp \quad (1.7)$$

It can also be defined as the ratio of the brake thermal efficiency to the indicated thermal efficiency.

1.8.4 Volumetric Efficiency (η_v)

This is one of the very important parameters which decides the performance of four-stroke engines. Four-stroke engines have distinct suction stroke and therefore the volumetric efficiency indicates the

breathing ability of the engine. It is to be noted that the utilization of the air is what going to determine the power output of the engine. Hence, an engine must be able to take in as much air as possible.

Volumetric efficiency is defined as the volume flow rate of air into the intake system divided by the rate at which the volume is displaced by the system.

$$\eta_v = \frac{\dot{m}_a}{\rho_a V_{disp} N/2} \quad (1.8)$$

where ρ_a is the inlet density

An alternative equivalent definition for volumetric efficiency is

$$\eta_v = \frac{\dot{m}_a}{\rho_a V_d} \quad (1.9)$$

It is to be noted that irrespective of the engine whether SI, CI or gas engine, *volumetric rate of air flow is what to be taken into account* and not the mixture flow.

If ρ_a is taken as the atmospheric air density, then η_v represents the pumping performance of the entire inlet system. If it is taken as the air density in the inlet manifold, then η_v represents the pumping performance of the inlet port and valve only.

The normal range of volumetric efficiency at full throttle for SI engines is between 80 to 85% where as for CI engines it is between 85 to 90%. Gas engines have much lower volumetric efficiency since gaseous fuel displaces air and therefore the breathing capacity of the engine is reduced.

1.8.5 Relative Efficiency or Efficiency Ratio (η_{rel})

Relative efficiency or efficiency ratio is the ratio of thermal efficiency of an actual cycle to that of the ideal cycle. The efficiency ratio is a very useful criterion which indicates the degree of development of the engine.

$$\eta_{rel} = \frac{\text{Actual thermal efficiency}}{\text{Air-standard efficiency}} \quad (1.10)$$

1.8.6 Mean Effective Pressure (p_m)

Mean effective pressure is the average pressure inside the cylinders of an internal combustion engine based on the calculated or measured power output. It increases as manifold pressure increases. For any particular engine, operating at a given speed and power output, there

will be a specific indicated mean effective pressure, $imep$, and a corresponding brake mean effective pressure, $bmep$. They are derived from the indicated and brake power respectively. For derivation see Chapter 17. Indicated power can be shown to be

$$ip = \frac{p_{im} L A n K}{60 \times 1000} \quad (1.11)$$

then, the indicated mean effective pressure can be written as

$$p_{im} = \frac{60000 \times ip}{L A n K} \quad (1.12)$$

Similarly, the brake mean effective pressure is given by

$$p_{bm} = \frac{60000 \times bp}{L A n K} \quad (1.13)$$

where ip = indicated power (kW)
 p_{im} = indicated mean effective pressure (N/m²)
 L = length of the stroke (m)
 A = area of the piston (m²)
 N = speed in revolutions per minute (rpm)
 n = Number of power strokes
 $N/2$ for 4-stroke and N for 2-stroke engines
 K = number of cylinders

Another way of specifying the indicated mean effective pressure p_{im} is from the knowledge of engine indicator diagram (p - V diagram). In this case, p_{im} , may be defined as

$$p_{im} = \frac{\text{Area of the indicator diagram}}{\text{Length of the indicator diagram}}$$

where the length of the indicator diagram is given by the difference between the total volume and the clearance volume.

1.8.7 Mean Piston Speed ($\bar{s}_p$)

An important parameter in engine applications is the mean piston speed, $\bar{s}_p$. It is defined as

$$\bar{s}_p = 2LN$$

where L is the stroke and N is the rotational speed of the crankshaft in rpm. It may be noted that $\bar{s}_p$ is often a more appropriate parameter than crank rotational speed for correlating engine behaviour as a function of speed.

Resistance to gas flow into the engine or stresses due to the inertia of the moving parts limit the maximum value of $\bar{s}_p$ to within 8 to 15 m/s. Automobile engines operate at the higher end and large marine diesel engines at the lower end of this range of piston speeds.

1.8.8 Specific Power Output (P_s)

Specific power output of an engine is defined as the power output per unit piston area and is a measure of the engine designer's success in using the available piston area regardless of cylinder size. The specific power can be shown to be proportional to the product of the mean effective pressure and mean piston speed.

$$\text{Specific power output, } P_s = bp/A \quad (1.14)$$

$$= \text{constant} \times p_{bm} \times \bar{s}_p \quad (1.15)$$

As can be seen the specific power output consists of two elements, viz., the force available to work and the speed with which it is working. Thus, for the same piston displacement and bme_p , an engine running at a higher speed will give a higher specific output. It is clear that the output of an engine can be increased by increasing either the speed or the bme_p . Increasing the speed involves increase in the mechanical stresses of various engine components. For increasing the bme_p better heat release from the fuel is required and this will involve more thermal load on engine cylinder.

1.8.9 Specific Fuel Consumption (sfc)

The fuel consumption characteristics of an engine are generally expressed in terms of specific fuel consumption in kilograms of fuel per kilowatt-hour. It is an important parameter that reflects how good the engine performance is. It is inversely proportional to the thermal efficiency of the engine.

$$sfc = \frac{\text{Fuel consumption per unit time}}{\text{Power}} \quad (1.16)$$

Brake specific fuel consumption and indicated specific fuel consumption, abbreviated as $bsfc$ and $isfc$, are the specific fuel consumptions on the basis of bp and ip respectively.

1.8.10 Inlet-Valve Mach Index (Z)

In a reciprocating engine the flow of intake charge takes place through the intake valve opening which is varying during the induction oper-

ation. Also, the maximum gas velocity through this area is limited by the local sonic velocity. Thus gas velocity is finally chosen by the following equation,

$$u = \frac{A_p}{C_i A_i} V_p \quad (1.17)$$

where u = gas velocity through the inlet valve
at smallest flow area

A_p = piston area

A_i = nominal intake valve opening area

C_i = inlet valve flow co-efficient

and

$$\frac{u}{\alpha} = \frac{A_p}{A_i} \frac{V_p}{C_i \alpha} = \left(\frac{b}{D_i} \right)^2 \frac{V_p}{C_i \alpha} = Z \quad (1.18)$$

where b = cylinder diameter

D_i = inlet valve diameter

V_p = mean piston speed

α = inlet sonic velocity

C_i = inlet valve average flow co-efficient

Z = inlet valve Mach index.

Large number of experiments have been conducted on CFR single cylinder engine with gaseous mixtures and short induction pipe lengths, at fixed valve timing and fixed compression ratio, but varying inlet valve diameter and lift. The results are quite revealing as regards the relationship of volumetric efficiency versus Mach index are concerned. From Fig.1.16, it could be seen that the maximum volumetric efficiency is obtainable for an inlet Mach number of 0.55. Therefore, engine designers must take care of this factor to get the maximum volumetric efficiency for their engines.

1.8.11 Fuel-Air (F/A) or Air-Fuel Ratio (A/F)

The relative proportions of the fuel and air in the engine are very important from the standpoint of combustion and the efficiency of the engine. This is expressed either as a ratio of the mass of the fuel to that of the air or vice versa.

In the SI engine the fuel-air ratio practically remains a constant over a wide range of operation. In CI engines at a given speed the air flow does not vary with load; it is the fuel flow that varies directly with load. Therefore, the term fuel-air ratio is generally used instead of air-fuel ratio.

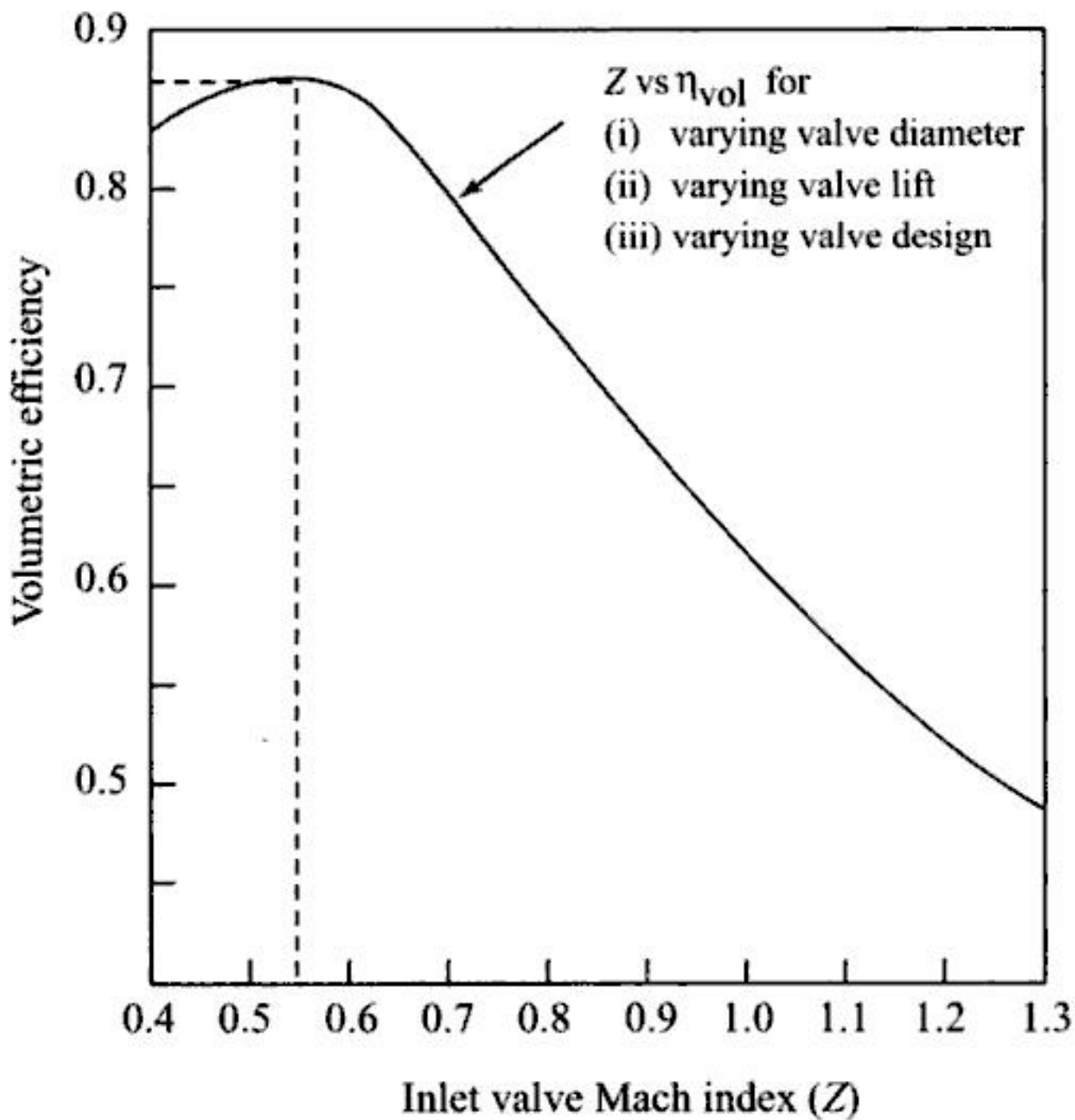


Fig. 1.16 Inlet-Valve Mach Index

A mixture that contains just enough air for complete combustion of all the fuel in the mixture is called a chemically correct or stoichiometric fuel-air ratio. A mixture having more fuel than that in a chemically correct mixture is termed as rich mixture and a mixture that contains less fuel (or excess air) is called a lean mixture. The ratio of actual fuel-air ratio to stoichiometric fuel-air ratio is called equivalence ratio and is denoted by ϕ .

$$\phi = \frac{\text{Actual fuel-air ratio}}{\text{Stoichiometric fuel-air ratio}} \quad (1.19)$$

Accordingly, $\phi = 1$ means stoichiometric (chemically correct) mixture, $\phi < 1$ means lean mixture and $\phi > 1$ means rich mixture.

1.8.12 Calorific Value (CV)

Calorific value of a fuel is the thermal energy released per unit quantity of the fuel when the fuel is burned completely and the products of combustion are cooled back to the initial temperature of the

combustible mixture. Other terms used for the calorific value are heating value and heat of combustion.

When the products of combustion are cooled to 25 °C practically all the water vapour resulting from the combustion process is condensed. The heating value so obtained is called the higher calorific value or gross calorific value of the fuel. The lower or net calorific value is the heat released when water vapour in the products of combustion is not condensed and remains in the vapour form.

1.9 DESIGN AND PERFORMANCE DATA

Engine ratings usually indicate the highest power at which manufacturers expect their products to give satisfactory economy, reliability, and durability under service conditions. Maximum torque, and the speed at which it is achieved, is also usually given. Since both of these quantities depend on displaced volume, for comparative analysis between engines of different displacements in a given engine category normalized performance parameters are more useful.

Typical design and performance data for SI and CI engines used in different applications are summarized in Table 1.4. The four-stroke cycle dominates except in the smallest and largest engines. The larger engines are turbocharged or supercharged. The maximum rated engine speed decreases as engine size increases, maintaining the maximum mean piston speed in the range of about 8 to 15 m/s. The maximum brake mean effective pressure for turbocharged and supercharged engines is higher than for naturally aspirated engines. Because the maximum fuel-air ratio for SI engines is higher than for CI engines, their naturally aspirated maximum *b MEP* levels are higher. As the engine size increases, brake specific fuel consumption decreases and fuel conversion efficiency increases due to the reduced heat losses and friction. For the large CI engines, brake thermal efficiencies of about 40% and indicated thermal efficiencies of about 50% can be obtained in modern engines.

Table 1.4 Typical Design and Performance Data for Modern Internal Combustion Engines

	Operating cycle (Stroke)	Compression ratio	Bore (m)	Stroke/bore ratio	Rated Maximum		Weight/ Power ratio (kg/kW)	Approx. best <i>bsfc</i> (g/kW h)
					Speed (rev/min)	<i>bmeP</i> (atm)		
Spark-ignition engines								
Small (e.g. motorcycles)	2/4	6–10	0.05–0.085	1.2–0.9	4500–7500	4–10	5.5–2.5	350
Passenger cars	4	8–10	0.07–0.1	1.1–0.9	4500–6500	7–10	4–2	270
Trucks	4	7–9	0.09–0.13	1.2–0.7	3600–5000	6.5–7	6.5–2.5	300
Large gas engines	2/4	8–12	0.22–0.45	1.1–1.4	300–900	6.8–12	23–35	200
Wankel engines	4	≈ 9	0.57 dm ³ per chamber		6000–8000	9.5–10.5	1.6–0.9	300
Compression-ignition engines								
Passenger cars	4	16–20	0.075–0.1	1.2–0.9	4000–5000	5–7.5	5–2.5	250
Trucks	4	16–20	0.1–0.15	1.3–0.8	2100–4000	6–9	7–4	210
Locomotive	4/2	16–18	0.15–0.4	1.1–1.3	425–1800	7–23	6–18	190
Large engines	2	10–12	0.4–1	1.2–3.0	110–400	9–17	12–50	180

Worked out Examples

- 1.1 The cubic capacity of a four-stroke over-square spark-ignition engine is 245 cc. The over-square ratio is 1.1. The clearance volume is 27.2 cc. Calculate the bore, stroke and compression ratio of the engine.

Solution

$$\text{Cubic capacity, } V_s = \frac{\pi}{4} d^2 L = \frac{\pi}{4} \frac{d^3}{1.1} = 245$$

$$d^3 = 343$$

$$\text{Bore, } d = 7 \text{ cm} \quad \underline{\underline{\text{Ans}}}$$

$$\text{Stroke, } L = \frac{7}{1.1} = 6.36 \text{ cm} \quad \underline{\underline{\text{Ans}}}$$

$$\begin{aligned} \text{Compression ratio, } r &= \frac{V_s + V_c}{V_c} \\ &= \frac{245 + 27.2}{27.2} = 10 \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

- 1.2 The mechanical efficiency of a single-cylinder four-stroke engine is 80%. The frictional power is estimated to be 25 kW. Calculate the indicated power (*ip*) and brake power (*bp*) developed by the engine.

Solution

$$\frac{bp}{ip} = 0.8$$

$$ip - bp = 25$$

$$ip - 0.8 \times ip = 25$$

$$ip = \frac{25}{0.2} = 125 \text{ kW} \quad \underline{\underline{\text{Ans}}}$$

$$bp = ip - fp = 125 - 25 = 100 \text{ kW} \quad \underline{\underline{\text{Ans}}}$$

- 1.3 A 42.5 kW engine has a mechanical efficiency of 85%. Find the indicated power and frictional power. If the frictional power is assumed to be constant with load, what will be the mechanical efficiency at 60% of the load?

Solution

$$\begin{aligned}
 \text{Indicated power, } ip &= \frac{bp}{\eta_m} = \frac{42.5}{0.85} = \mathbf{50 \text{ kW}} \quad \underline{\underline{\text{Ans}}} \\
 \text{Frictional power, } fp &= ip - bp = 50 - 42.5 \\
 &= \mathbf{7.5 \text{ kW}} \quad \underline{\underline{\text{Ans}}} \\
 \text{Brake power at 60\% load} &= 42.5 \times 0.6 = 25.5 \text{ kW} \\
 \text{Mechanical efficiency } \eta_m &= \frac{bp}{bp + fp} = \frac{25.5}{25.5 + 7.5} \\
 &= 0.773 = \mathbf{77.3\%} \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

- 1.4 Find out the speed at which a four-cylinder engine using natural gas can develop a brake power of 50 kW working under following conditions. Air-gas ratio 9:1, calorific value of the fuel = 34 MJ/m³, Compression ratio 9:1, volumetric efficiency = 70%, indicated thermal efficiency = 35% and the mechanical efficiency = 80% and the total volume of the engine is 2 litres.

Solution

$$\begin{aligned}
 \text{Total volume/cylinder, } V_{tot} &= \frac{2000}{4} = 500 \text{ cc} \\
 \text{Swept volume/cylinder, } V_s &= \frac{9}{10} \times 500 = 450 \text{ cc} \\
 \text{Volume of air taken in/cycle} &= \eta_v \times V_s \\
 &= 0.7 \times 450 = 315 \text{ cc} \\
 \text{Volume of gas taken in/cycle} &= \frac{315}{9} = 35 \text{ cc} \\
 \text{Energy supplied/cylinder, } E &= 35 \times 10^{-6} \times 34 \times 10^3 \\
 &= \mathbf{1.19 \text{ kJ}} \quad (1)
 \end{aligned}$$

$$\text{Indicated thermal efficiency, } \eta_{ith} = \frac{bp/\eta_m}{\text{Energy supplied/cylinder/s}}$$

$$\text{Energy supplied/cylinder/s, } E_1 = \frac{50/0.8}{0.35 \times 4} = 44.64 \text{ kJ}$$

$$\begin{aligned} \text{Now, energy supplied per cylinder in kJ} &= \frac{E_1}{N/120} \\ &= \frac{44.64 \times 120}{N} \\ &= \frac{5356.8}{N} \end{aligned} \quad (2)$$

$$\text{Equating (1) and (2) } \frac{5356.8}{N} = 1.19$$

$$N \approx 4500 \text{ rpm} \quad \underline{\underline{\text{Ans}}}$$

1.5 A four-stroke, four-cylinder diesel engine running at 2000 rpm develops 60 kW. Brake thermal efficiency is 30% and calorific value of fuel (*CV*) is 42 MJ/kg. Engine has a bore of 120 mm and stroke of 100 mm. Take $\rho_a = 1.15 \text{ kg/m}^3$, air-fuel ratio = 15:1 and $\eta_m = 0.8$. Calculate (i) fuel consumption (kg/s); (ii) air consumption (m^3/s); (iii) indicated thermal efficiency; (iv) volumetric efficiency; (v) brake mean effective pressure and (vi) mean piston speed

Solution

$$\begin{aligned} \text{Fuel consumption, } \dot{m}_f &= \frac{bp}{\eta_{bth} \times CV} = \frac{60}{0.3 \times 42000} \\ &= 4.76 \times 10^{-3} \text{ kg/s} \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

$$\begin{aligned} \text{Air consumption} &= \frac{\dot{m}_f}{\rho_a} \frac{A}{F} = \frac{4.76 \times 10^{-3}}{1.15} \times 15 \\ &= 62.09 \times 10^{-3} \text{ m}^3/\text{s} \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

$$\text{Air flow rate/cylinder} = \frac{62.09 \times 10^{-3}}{4} = 15.52 \times 10^{-3} \text{ m}^3/\text{s}$$

$$\text{Indicated power} = \frac{bp}{\eta_m} = \frac{60}{0.8} = 75 \text{ kW}$$

$$\begin{aligned}\eta_{ith} &= \frac{75}{4.76 \times 10^{-3} \times 42000} \\ &= 0.37515 = \mathbf{37.51\%} \quad \underline{\underline{\text{Ans}}}\end{aligned}$$

Volumetric efficiency =

$$\frac{\text{Actual volume flow rate of air}}{\text{Volume flow rate of air corresponding to displacement volume}} \times 100$$

$$\begin{aligned}\eta_v &= \frac{15.52 \times 10^{-3}}{\frac{\pi}{4} \times 0.12^2 \times 0.10 \times \frac{2000}{120}} \times 100 \\ &= \mathbf{82.3\%} \quad \underline{\underline{\text{Ans}}}\end{aligned}$$

Brake mean effective pressure,

$$\begin{aligned}p_{bm} &= \frac{bp}{LANK} \\ &= \frac{60}{0.1 \times \frac{\pi}{4} \times 0.12^2 \times \frac{2000}{2 \times 60} \times 4} \times 10^3 \\ &= 7.96 \times 10^5 \text{ N/m}^2 = \mathbf{7.96 \text{ bar}} \quad \underline{\underline{\text{Ans}}}\end{aligned}$$

- 1.6 A single-cylinder, four-stroke hydrogen fuelled spark-ignition engine delivers a brake power of 20 kW at 6000 rpm. The air-gas ratio is 8:1 and the calorific value of fuel is 11000 kJ/m³. The compression ratio is 8:1. If volumetric efficiency is 70%, indicated thermal efficiency is 33% and the mechanical efficiency is 90%, calculate the cubic capacity of the engine.

Solution

$$\begin{aligned}\text{Energy input} &= \frac{bp/\eta_m}{\eta_{ith}} = \frac{20}{0.8 \times 0.33} \\ &= 75.76 \text{ kJ/s}\end{aligned}$$

$$\text{Number of power strokes/s} = \frac{N}{2 \times 60} = \frac{6000}{120} = 50$$

$$\text{Energy input/power stroke} = \frac{75.76}{50} = 1.52 \text{ kJ}$$

$$\text{Actual volume of H}_2 \times CV = 1.52$$

Actual volume of hydrogen taken in

$$= \frac{1.52 \times 10^6}{11000} = 138.18 \text{ cc}$$

$$\text{Actual volume of air take in} = \frac{A}{F} \times 138.18 = 8 \times 138.18$$

$$= 1105.44 \text{ cc}$$

$$\text{Swept volume, } V_s = \frac{\text{Actual volume of air taken in}}{\eta_v}$$

$$= \frac{1105.44}{0.7} = 1579.2 \text{ cc}$$

$$\text{Cubic capacity of the engine} = V_s \times K = 1579.2 \times 1$$

$$= \mathbf{1579.2 \text{ cc}} \quad \leftarrow \underline{\underline{\text{Ans}}}$$

1.7 Consider two engines with the following details:

Engine I: Four-stroke, four-cylinder, SI engine, indicated power is 40 kW, mean piston speed 10 m/s.

Engine II: Two-stroke, two-cylinder, SI engine, indicated power is 10 kW.

Assume that mean effective pressure of both the engine to be same. Ratio of bore of the engine I:II = 2:1. Show that the mean piston speed of engine II is same as that of engine I.

Solution

$$ip = \frac{P_m L A n K}{60000}$$

$n = \frac{N}{2}$ for four-stroke engine and $n = N$ for two-stroke engine.

$$\bar{s}_p = 2LN$$

$$\text{For engine I: } 40 = \frac{P_{mI} \times A_I \times \frac{\bar{s}_{pI}}{4} \times 4}{60000}$$

$$\text{For engine II: } 10 = \frac{P_{mII} \times A_{II} \times \frac{\bar{s}_{pII}}{2} \times 2}{60000}$$

$$\frac{40}{10} = \frac{A_I}{A_{II}} \times \frac{10}{\bar{s}_{pII}}$$

$$\begin{aligned}\bar{s}_{pII} &= \frac{A_I}{A_{II}} \times \frac{10}{4} = \left(\frac{d_1}{d_2}\right)^2 \times 2.5 \\ &= \left(\frac{2}{1}\right)^2 \times 2.5 = 10 \text{ m/s}\end{aligned}$$

$$\bar{s}_{pII} = \bar{s}_{pI} = 10 \text{ m/s} \quad \underline{\underline{\text{Ans}}}$$

1.8 An one-litre cubic capacity, four-stroke, four-cylinder SI engine has a brake thermal efficiency of 30% and indicated power is 40 kW at full load. At half load, it has a mechanical efficiency of 65%. Assuming constant mechanical losses, calculate: (i) brake power (ii) frictional power (iii) mechanical efficiency at full load (iv) indicated thermal efficiency. If the volume decreases by eight-fold during the compression stroke, calculate the clearance volume.

Solution

Let the brake power at full load be bp and the frictional power be fp .

$$bp + fp = 40 \text{ kW} \quad (1)$$

$$\text{At half load, } bp = 0.5 \times bp \text{ at full load}$$

$$\eta_m = 0.65 = \frac{0.5 bp}{0.5 bp + fp}$$

$$\begin{aligned}0.5 bp &= 0.65 \times (0.5 \times bp + fp) \\ &= 0.325 \times bp + 0.65 \times fp\end{aligned}$$

$$fp = \frac{0.175}{0.65} \times bp = 0.27bp \quad (2)$$

$$\text{Using (2) in (1)} \quad bp = \frac{40}{1.27} = 31.5 \text{ kW} \quad \underline{\underline{\text{Ans}}}$$

$$fp = 31.5 \times 0.27 = 8.5 \text{ kW} \quad \underline{\underline{\text{Ans}}}$$

$$\eta_m \text{ at full load} = \frac{31.5}{40} = 0.788 = 78.8\% \quad \underline{\underline{\text{Ans}}}$$

$$\eta_{ith} = \frac{\eta_{bth}}{\eta_m} = \frac{30}{78.8} \times 100 = 38\% \quad \underline{\underline{\text{Ans}}}$$

$$\text{Swept volume/cylinder} = \frac{1000}{4} = 250 \text{ cc}$$

$$r = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c} = 8$$

$$V_c = \frac{250}{7} = 35.71 \text{ cc} \quad \underline{\underline{\text{Ans}}}$$

1.9 A four-stroke petrol engine at full load delivers 50 kW. It requires 8.5 kW to rotate it without load at the same speed. Find its mechanical efficiency at full load, half load and quarter load?

Also find out the volume of the fuel consumed per second at full load if the brake thermal efficiency is 25%, given that calorific value of the fuel = 42 MJ/kg and specific gravity of petrol is 0.75. Estimate the indicated thermal efficiency.

Solution

$$\begin{aligned} \text{Mechanical efficiency at full load} &= \frac{bp}{bp + fp} \\ &= \frac{50}{50 + 8.5} = 0.8547 = 85.47\% \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

$$\begin{aligned} \text{Mechanical efficiency at half load} \\ &= \frac{25}{25 + 8.5} = 0.7462 = 74.62\% \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

$$\begin{aligned} \text{Mechanical efficiency at quarter load} \\ &= \frac{12.5}{12.5 + 8.5} = 0.5952 = 59.52\% \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

$$\begin{aligned} \text{Mass flow rate of fuel } \dot{m}_f &= \frac{bp}{\eta_{bth} \times CV} \\ &= \frac{50}{0.25 \times 42000} = 4.76 \times 10^{-3} \text{ kg/s} \end{aligned}$$

$$\begin{aligned} \text{Volume flow rate of fuel} \\ &= \frac{4.76 \times 10^{-3}}{750} = 6.34 \times 10^{-6} \text{ m}^3/\text{s} \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

$$\begin{aligned} \text{Indicated thermal efficiency at full load} \\ \eta_{ith} &= \frac{\eta_{bth}}{\eta_m} = \frac{0.25}{0.8547} = 0.2925 = 29.25\% \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

1.10 The indicated thermal efficiency of four-stroke engine is 32% and its mechanical efficiency is 78%. The fuel consumption rate is 20 kg/h running at a fixed speed. The brake mean pressure developed is 6 bar and the mean piston speed is 12 m/s. Assuming it to be a single cylinder square engine, calculate the crank radius and the speed of the engine. Take $CV = 42000$ kJ/kg.

Solution

$$\begin{aligned}\text{Brake thermal efficiency, } \eta_{bth} &= \eta_{ith} \times \eta_m = 0.32 \times 0.78 \\ &= 0.2496 = 24.96\%\end{aligned}$$

$$\begin{aligned}\text{Rate of energy input from fuel} &= \frac{20}{3600} \times 42000 = 233.33 \text{ kW} \\ \text{Brake power } bp &= \eta_{bth} \times 233.33 \\ &= 0.2496 \times 233.33 = 58.24 \text{ kW}\end{aligned}$$

Since it is a square engine, $d = L$.

$$\begin{aligned}p_{bm} &= \frac{bp \times 60000}{LAnK} \\ &= \frac{58.24 \times 60000}{\frac{\pi}{4}L^3 \times n \times 1} = 6 \times 10^5 \\ L^3n &= 7.415 \quad (1)\end{aligned}$$

Note L is in m and N in per minute. Now,

$$\begin{aligned}\bar{s}_p &= 12 = \frac{2LN}{60} \\ LN &= 360 \quad (2)\end{aligned}$$

Dividing (1) by (2), gives,

$$L^2 \frac{n}{N} = 0.0206$$

For a four-stroke engine $n/N = \frac{1}{2}$.

$$\begin{aligned}L &= \sqrt{0.0206 \times 2} = 0.203 \text{ m} \\ &= 203 \text{ mm}\end{aligned}$$

$$\text{Crank radius} = \frac{203}{2} = 101.5 \text{ mm}$$

$$\text{Speed, } N = \frac{360}{L} = \frac{360}{0.203}$$

$$= 1773.4 \text{ rpm} \quad \text{Ans}$$

Review Questions

1.1 Define the following : (i) engine and (ii) heat engine.

1.2 How are heat engines classified?

1.3 Explain the basic difference in their work principle?

1.4 Give examples of EC and IC engines.

1.5 Compare EC and IC engines.

1.6 What are the important basic components of an IC engine? Explain them briefly.

1.7 Draw the cross-section of a single cylinder spark-ignition engine and mark the important parts.

1.8 Define the following :

- | | |
|---------------------------|-----------------------|
| (i) bore | (iv) clearance volume |
| (ii) stroke | (v) compression ratio |
| (iii) displacement volume | (vi) cubic capacity |

Mention the units in which they are normally measured.

1.9 What is meant by TDC and BDC? In a suitable sketch mark the two dead centres.

1.10 What is meant by cylinder row and cylinder bank?

1.11 With neat sketches explain the working principle of four-stroke spark-ignition engine.

1.12 Classify the internal combustion engine with respect to

- | | |
|---------------------------------------|------------------------|
| (i) cycle of operation | (iv) type of ignition |
| (ii) cylinder arrangements | (v) type of fuels used |
| (iii) method of charging the cylinder | (vi) type of cooling |

- 1.13 In what respects four-stroke cycle CI engine differ from that of an SI engine?
- 1.14 What is the main reason for the development of two-stroke engines and what are the two main types of two-stroke engines?
- 1.15 Describe with a neat sketch the working principle of a crankcase scavenged two-stroke engine.
- 1.16 Draw the ideal and actual indicator diagrams of a two-stroke SI engine. How are they different from that of a four-stroke cycle engine?
- 1.17 Compare four-stroke and two-stroke cycle engines. Bring out clearly their relative merits and demerits.
- 1.18 Compare SI and CI engines with respect to
- | | |
|----------------------------|-----------------------|
| (i) basic cycle | (v) compression ratio |
| (ii) fuel used | (vi) speed |
| (iii) introduction of fuel | (vii) efficiency |
| (iv) ignition | (viii) weight |
- 1.19 Discuss in detail the application of various types of internal combustion engines.
- 1.20 Give an account of the first law analysis of an internal combustion engine.
- 1.21 Show by means of a diagram the energy flow in a reciprocating internal combustion engine.
- 1.22 What is meant by mean piston speed? Explain its importance.
- 1.23 Discuss briefly the design performance data of SI and CI engines.
- 1.24 Define the following efficiencies :
- | | |
|----------------------------------|---------------------------|
| (i) indicated thermal efficiency | (iv) relative efficiency |
| (ii) brake thermal efficiency | (v) volumetric efficiency |
| (iii) mechanical efficiency | |
- 1.25 Explain briefly
- | |
|---------------------------------|
| (i) mean effective pressure |
| (ii) specific output |
| (iii) specific fuel consumption |
| (iv) fuel-air ratio |
| (v) heating value of the fuel |

Exercise

- 1.1 A diesel engine has a brake thermal efficiency of 30 per cent. If the calorific value of the fuel is 42000 kJ/kg. Find its brake specific fuel consumption. *Ans:* 0.2857 kg/kW h
- 1.2 A gas engine having a cylinder 250 mm bore and 450 mm stroke has a volumetric efficiency of 80%. Air-gas ratio equals 9:1, calorific value of fuel 21000 kJ per m³ at NTP. Calculate the heat supplied to the engine per working cycle. If the compression ratio is 5:1, what is the heat value of the mixture per working stroke per m³ of total cylinder volume? *Ans:* (i) 36.08 kJ (ii) 1306.8 kJ/m³
- 1.3 A certain engine at full load delivers a brake power of 36.8 kW at certain speed. It requires 7.36 kW to overcome the friction and to rotate the engine without load at the same speed. Calculate its mechanical efficiency. Assuming that the mechanical losses remain constant, what will be the mechanical efficiency at half load and quarter load. *Ans:* (i) 83.33% (ii) 71.42% (iii) 55.55%
- 1.4 An engine used for pumping water develops a brake power of 3.68 kW. Its indicated thermal efficiency is 30%, mechanical efficiency is 80%, calorific value of the fuel is 42,000 kJ/kg and its specific gravity = 0.875. Calculate (i) the fuel consumption of the engine in (a) kg/h (b) litres/h (ii) indicated specific fuel consumption and (iii) brake specific fuel consumption. *Ans:* (i) (a) 1.31 kg/h (b) 1.5017 litres/h (ii) 0.2856 kg/kW h (iii) 0.3571 kg/kW h
- 1.5 A two-stroke CI engine develops a brake power of 368 kW while its frictional power is 73.6 kW. Its fuel consumption is 180 kg/h and works with an air-fuel ratio of 20:1. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) mechanical efficiency (iii) air consumption per hour (iv) indicated thermal efficiency and (v) brake thermal efficiency. *Ans:* (i) 441.6 kW (ii) 83.3% (iii) 3600 kg/h (iv) 21% (v) 17.5%
- 1.6 Compute the brake mean effective pressure of a four-cylinder, four-stroke diesel engine having 150 mm bore and 200 mm stroke which develops a brake power of 73.6 kW at 1200 rpm. *Ans:* 5.206 bar

- 1.7 Compute the brake mean effective pressure of a four-cylinder, two-stroke engine, 100 mm bore 125 mm stroke when it develops a torque of 490 Nm. *Ans:* 7.84 bar
- 1.8 Find the brake thermal efficiency of an engine which consumes 7 kg of fuel in 20 minutes and develops a brake power of 65 kW. The fuel has a heating value of 42000 kJ/kg. *Ans:* 26.53%
- 1.9 Find the mean piston speed of a diesel engine running at 1500 rpm. The engine has a 100 mm bore and L/d ratio is 1.5. *Ans:* 7.5 m/s
- 1.10 An engine is using 5.2 kg of air per minute while operating at 1200 rpm. The engine requires 0.2256 kg of fuel per hour to produce an indicated power of 1 kW. The air-fuel ratio is 15:1. Indicated thermal efficiency is 38% and mechanical efficiency is 80%. Calculate (i) brake power and (ii) heating value of the fuel. *Ans:* (i) 73.7 kW (ii) 41992.89 kJ/kg
- 1.11 A four-cylinder, four-stroke, spark-ignition engine has a bore of 80 mm and stroke of 80 mm. The compression ratio is 8. Calculate the cubic capacity of the engine and the clearance volume of each cylinder. What type of engine is this? *Ans:* (i) 1608.4 cc (ii) 57.4 cc (iii) Square engine
- 1.12 A four-stroke, compression-ignition engine with four cylinders develops an indicated power of 125 kW and delivers a brake power of 100 kW. Calculate (i) frictional power (ii) mechanical efficiency of the engine. *Ans:* (i) 25 kW (ii) 80%
- 1.13 An engine with 80 per cent mechanical efficiency develops a brake power of 30 kW. Find its indicated power and frictional power. If frictional power is assumed to be constant, what will be the mechanical efficiency at half load. *Ans:* (i) 37.5 kW (ii) 7.5 kW (iii) 66.7%
- 1.14 A single-cylinder, compression-ignition engine with a brake thermal efficiency of 30% uses high speed diesel oil having a calorific value of 42000 kJ/kg. If its mechanical efficiency is 80 per cent, calculate (i) *bsfc* in kg/kW h (ii) *isfc* in kg/kW h *Ans:* (i) 0.286 kg/kW h (ii) 0.229 kg/kW h
- 1.15 A petrol engine uses a fuel of calorific value of 42000 kJ/kg and has a specific gravity of 0.75. The brake thermal efficiency is 24 per cent and mechanical efficiency is 80 per cent. If the engine

develops a brake power of 29.44 kW, calculate (i) volume of the fuel consumed per second (ii) indicated thermal efficiency

Ans: (i) 2.92×10^{-3} kg/s (ii) 30%

- 1.16 A single-cylinder, four-stroke diesel engine having a displacement volume of 790 cc is tested at 300 rpm. When a braking torque of 49 Nm is applied, analysis of the indicator diagram gives a mean effective pressure of 980 kPa. Calculate the brake power and mechanical efficiency of the engine.

Ans: (i) 1.54 kW (ii) 79.4%

- 1.17 A four-stroke SI engine delivers a brake power of 441.6 kW with a mechanical efficiency of 85 per cent. The measured fuel consumption is 160 kg of fuel in one hour and air consumption is 410 kg during one sixth of an hour. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) frictional power (iii) air-fuel ratio (iv) indicated thermal efficiency (v) brake thermal efficiency.

Ans: (i) 519.5 kW (ii) 77.9 kW (iii) 15.5
(iv) 28.1% (v) 23.9%

- 1.18 A two-stroke CI engine develops a brake power of 368 kW while 73.6 kW is used to overcome the friction losses. It consumes 180 kg/h of fuel at an air-fuel ratio of 20:1. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) mechanical efficiency; (iii) Air consumption (iv) indicated thermal efficiency (v) brake thermal efficiency.

Ans: (i) 441.6 kW (ii) 83.3% (iii) 1 kg/s
(iv) 21% (v) 17.5%

- 1.19 A four-stroke petrol engine delivers a brake power of 36.8 kW with a mechanical efficiency of 80%. The air-fuel ratio is 15:1 and the fuel consumption is 0.4068 kg/kW h. The heating value of the fuel is 42000 kJ/kg. Calculate (i) indicated power (ii) frictional power (iii) brake thermal efficiency (iv) indicated thermal efficiency (v) total fuel consumption (vi) air consumption/second.

Ans: (i) 46 kW (ii) 9.2 kW (iii) 21% (iv) 26.25%
(v) 0.0042 kg/s (vi) 0.063 kg/s.

- 1.20 A spark-ignition engine has a fuel-air ratio of 0.067. How many kg of air per hour is required for a brake power output of 73.6 kW at an overall brake thermal efficiency of 20%? How many m^3 of air is required per hour if the density of air

is 1.15 kg/m^3 . If the fuel vapour has a density four times that of air, how many m^3 per hour of the mixture is required? The calorific value of the fuel is given as 42000 kJ/kg .

Ans: (i) 470.75 kg/h (ii) $409.35 \text{ m}^3/\text{h}$ (iii) $416.21 \text{ m}^3/\text{h}$

- 1.21 A four-stroke CI engine having a cylinder diameter of 39 cm and stroke of 28 cm has a mechanical efficiency of 80% . Assume the frictional power to be 80 kW . Its fuel consumption is 86 kg/h with an air-fuel ratio of $18:1$. The speed of the engine is 2000 rpm . Calculate (i) indicated power (ii) if η_{ith} is 40% , calculate the calorific value of the fuel used (iii) p_{im} (iv) $\dot{m}_a/\text{hour}$ (v) $\bar{s}_p$.

Ans: (i) 400 kW (ii) 41860 kJ (iii) 12.13 bar
(iv) 1548 kg/h (v) 18.7 m/s

- 1.22 A four-stroke LPG engine having a cylinder 250 mm diameter and stroke of 300 mm has a volumetric efficiency of 70% at atmospheric conditions. Gas to air ratio is $8:1$. Calorific value of the fuel is 100 MJ/m^3 at atmospheric conditions. Find the heat supplied to the engine per working cycle. If the compression ratio is 10 , what is the heat value of the mixture per working stroke per m^3 of the total cylinder volume?

Ans: (i) 128.8 kJ (ii) 7.8 MJ

- 1.23 A four-cylinder spark-ignition engine has the following dimensions: bore = 680 mm and a crank radius = 375 mm . If the compression ratio is $8:1$, determine the (i) stroke length (ii) swept volume (iii) cubic capacity (iv) clearance volume and (v) total volume. If the volumetric efficiency is 80% determine the (vi) actual volume of air aspirated/stroke in each cylinder?

Ans: (i) 750 mm (ii) 1.088 m^3 (iii) 0.272 m^3 (iv) 0.039 m^3
(v) 0.0311 m^3 (vi) 0.2176 m^3

- 1.24 An engine with an indicated thermal efficiency of 25% and mechanical efficiency of 75% consumes 25 kg/h of fuel at a fixed speed. The brake mean effective pressure is 5 bar and the mean piston speed is 15 m/s . Assuming it is a single cylinder square engine determine the crank radius and the speed in rpm. Take CV of the fuel = 42 MJ/kg . *Ans:* (i) 68.2 mm (ii) 3300 rpm

- 1.25 A four-cylinder SI engine running at 1200 rpm gives 18.87 kW as brake power. When one cylinder missed firing the average torque was 100 Nm . Calculate the indicated thermal efficiency if the calorific value of fuel is 42000 kJ/kg . The engine uses 0.335 kg of fuel per kW/h . What is the mechanical efficiency of the engine?

Ans: (i) 34.2% (ii) 74.9%

- 1.26 A certain engine with a bore of 250 mm has an indicated thermal efficiency of 30%. The *bsfc* and specific power output are 0.35 kg/kW h and 90 kW/m². Find the mechanical efficiency and brake thermal efficiency of the engine. Take the calorific value of the fuel as 42 MJ/kg. *Ans:* (i) 81.7% (ii) 24.5%
- 1.27 A single-cylinder, four-stroke engine having a cubic capacity of 0.7 litre was tested at 200 rpm. From the indicator diagram the mean effective pressure was found to be 10⁶ N/m² and the mechanical efficiency is 75%. Find the frictional power of the engine if the engine is an over-square engine with a over-square ratio of 0.8. Calculate the bore and stroke.
Ans: (i) 0.29 kW (ii) 41.5 mm (iii) 89.34 mm
 (iv) 111.68 mm
- 1.28 In a performance test on a four-stroke engine, the indicator diagram area was found to be 5×10^{-4} m² and the length of the indicator diagram was 0.05 m. If the y-axis has a scale of 1 m = 50 MPa, find the *imep* of the engine given that bore = 150 mm, stroke = 200 mm. The measured engine speed was 1200 rpm. Also calculate the *ip* and *isfc* of the engine if the fuel injected per cycle is 0.5 cc with the specific gravity of 0.8.
Ans: (i) 5 bar (ii) 70.68 kW (iii) 203 g/kW h
- 1.29 A four-stroke, four-cylinder automotive engine develops 150 Nm brake torque at 3000 rpm. Assuming *bmep* to be 0.925 bar, find (i) brake power (ii) displacement volume (iii) stroke (iv) bore. Take $\bar{s}_p = 12$ m/s.
Ans: (i) 47.124 kW (ii) 5.1×10^{-3} m³
 (iii) 120 mm (iv) 233 mm
- 1.30 A single-cylinder, four-stroke, engine has a *bsfc* of 1.13×10^{-5} kg/kW h and a fuel consumption rate of 0.4068 kg/h. The specific power output of the engine is 0.33 kW/cm². If the engine runs at 3000 rpm find the displacement volume of the cylinder and if the $\bar{s}_p$ is 15 m/s, find the *bmep*.
Ans: (i) 900 cc (ii) 4.44 bar

2

REVIEW OF BASIC PRINCIPLES

2.1 INTRODUCTION

The main objectives in studying the theory of IC engines can be summarized as

- (i) To have a better understanding of the various processes taking place and the conditions prevailing in the engine cylinder.
- (ii) To predict the changes in power, fuel consumption and reliability resulting from the changes in the operating conditions or changes in the design features of a given engine.
- (iii) To predict the operating characteristics of a new design from test results on a similar engine of a different size.

Basic knowledge and the familiarity of the principles of thermodynamics, physics and chemistry is a must for achieving the above objectives. However, mere familiarity alone will not serve the purpose. The familiarity must be supplemented by the ability to apply these principles correctly. This requires an understanding of fundamentals used to derive the important formulae. The main aim of this chapter is to place before the reader the basic principles of thermodynamics, physics, chemistry and other relevant information for the understanding of the various details discussed in the subsequent chapters. Although a fairly exact treatment has been attempted, the reader is advised to go through other relevant textbooks to get more detailed account of the various basic principles.

2.2 BASIC AND DERIVED QUANTITIES

In this section we will briefly recall the definition of basic quantities like length, time, mass, weight, acceleration due to gravity, force and pressure. These quantities will be used to derive other quantities.

2.2.1 Length

The unit of length is metre. Originally, metal bar standards were used to define length. They have been abandoned now in favour of a standard length which can be reproduced in any laboratory in the world. Length is now defined in terms of the atomic standards. The orange light emitted by the gas krypton under special experimental conditions is used to define length. One metre is defined to be 1,650,763.73 wavelengths of the orange light Krypton-86. However, for convenience in engine applications, millimeter (mm) and centimeter (cm) may also be used to represent length.

2.2.2 Time

The unit of time is second. Time is also defined in terms of atomic standards. The time standard is based on the duration of the periods of the radiation of the atom cesium-133.

$$1 \text{ second} = 9,192,631,770 \text{ periods}$$

The second, minute, hour, day and year are used throughout the world. Their interrelation requires no description here.

2.2.3 Mass and Weight

The mass of a body, m is the amount of matter in the body. The mass of a given body therefore is the same anywhere in the universe. The unit of mass is kilogram (kg). However, for engine applications, gram (gm) may also be used as the unit of mass.

The weight of a body, w is the force with which the earth attracts the body. The weight of a given body will change with the value of acceleration due to gravity, g .

The weight, w is always equal to the number of kg mass, m times the local value of g or

$$w = mg \quad (2.1)$$

2.2.4 Gravitational Constant

The force of attraction by the earth experienced by any body is called the gravitational force. Since, the mass of most of the bodies under consideration are negligible compared to that of earth this force affects only the other bodies. This results in bodies not able to float and a body without support experiences a motion in the direction of the gravitational force. The acceleration of such freely falling body is called acceleration due to gravity. This is found to vary from place to place on the surface of the earth and hence an average value is used for engineering calculations, viz., 9.81 m/s^2 or 32 ft/s^2 .

In the derivation of unit of force it is required to prescribe mass and acceleration. In prescribing the acceleration, two conventions are used, one to prescribe unit acceleration and the other to prescribe the acceleration due to gravity. In the four methods of systems of units so far evolved two use the former and two later.

F.P.S.	:	1 pound	=	1 slug $\times$ 32 ft/s^2
C.G.S.	:	1 dyne	=	1 gm $\times$ 1 m/s^2
M.K.S.	:	1 kgf	=	1 kg $\times$ 9.81 m/s^2
SI	:	1 Newton	=	1 kg $\times$ 1 m/s^2

In using M.K.S. system care should be taken to differentiate between kgf and kg since 1 kgf is actually equal to 9.81 Newton. This is done by defining a gravitational constant, g_c which is numerically equal to the acceleration due to gravity but without dimensions. Since it is proposed to use SI units in this book use of g_c is not elaborated further.

2.2.5 Force

In this book the unit of force will be Newton and is abbreviated as N. A Newton is the force required to give an acceleration of 1 m/s^2 to a mass of one kg of matter. This definition is adopted because it is convenient to metrologists; its adoption is rendered possible by the universal validity of Newton's Second Law of Motion.

2.2.6 Pressure

Fluids, gas or liquid exert forces on its boundaries. These forces are not concentrated at particular points, but are distributed. It is therefore useful to define the quantity pressure as the force exerted on a surface which is normal to the force, divided by the area of the

surface. The units of pressure is N/m^2 . However, it may also be expressed in bar, Pascal and standard atmosphere, (atm).

The inter relation between bar, Pascal and standard atmosphere with N/m^2 is given below.

$$\begin{array}{llll} 1 \text{ Pascal} & (\text{Pa}) & = & 1 \text{ N/m}^2 \\ 1 \text{ bar} & (\text{bar}) & = & 10^5 \text{ Pascal} \\ 1 \text{ standard atmosphere} & (\text{atm}) & \approx & 1.01325 \times 10^5 \text{ N/m}^2 \end{array}$$

2.2.7 Temperature

The unit of temperature is Kelvin. The Kelvin is $1/273.16$ of the thermodynamic temperature of the triple point of water, viz., the point at which liquid water, water vapour and ice are in equilibrium.

2.3 THE GAS LAWS

During the various strokes of the engine different pressures and temperatures exist in the engine cylinder and all gases encountered in engine operation are assumed to obey at all these pressures and temperatures the simple gas law,

$$pV = N\bar{R}T \quad (2.2)$$

where	p	=	Pressure of the gas	(N/m^2)
	V	=	Volume occupied by the gas	(m^3)
	N	=	Number of moles of gas	(kmol)
	$\bar{R}$	=	Universal gas constant	$(8.314 \text{ kJ/kmol K})$
	T	=	Absolute temperature	(K)

The above equation is an approximation sufficiently close for engineering purposes. This law is very simple and 'always' applies to 'any' gas, regardless of the process to which the gas is subjected, as long as the gas is not near its liquefaction.

2.3.1 The Mole

A kilomol of any gas is a quantity of the gas whose weight is equal to the molecular weight in kg. Thus

$$N = \frac{m}{M} \quad (2.3)$$

where	m	=	Mass of gas (kg)
	M	=	Molecular weight of the gas, (g/mol)

Combining Eqs.2.2 and 2.3, the gas law may be written as

$$pV = \frac{m}{M} \bar{R}T = m \left(\frac{\bar{R}}{M} \right) T = mRT \quad (2.4)$$

The quantity $\bar{R}/M$ is called the characteristic gas constant and is usually denoted by R . Since M is different for different gases, the characteristic gas constant will also be different for different gases.

2.3.2 Specific Volume and Density

The volume occupied by unit mass of a substance is known as its specific volume (m^3/kg) and is denoted by v . The inverse of the specific volume is the density (kg/m^3) and is denoted by ρ . The specific volume and density of the gas depend upon the pressure and temperature.

2.3.3 Simplification of the Gas Laws

When the number of moles m/M of gas under consideration is fixed, both N and R are constant. Then from Eqs.2.2 and 2.4,

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} = \dots = \frac{p_n v_n}{T_n} = \text{Constant} \quad (2.5)$$

This simple relation is always true when applied to a constant number of moles of a gas and makes it possible to calculate the effect of changing any two of the variables (pressure, volume or temperature) upon the remaining one.

2.3.4 Mixture of Gases

During expansion and exhaust strokes (also during suction and compression) the cylinder is occupied by two or more gases. Each of these gases diffuses and fills up the entire space, obeying the gas law just as though the other gases were not present. This principle is called Dalton's Law. For example, let three gases 1, 2 and 3 occupy the cylinder at a particular instant. Let the volume of the cylinder at that instant be V . If the temperature of the mixture is T , then the pressure exerted on the walls of the cylinder by gas 1 is, by the gas law,

$$p_1 = \frac{N_1 \bar{R}T}{V}$$

and the pressure exerted by gas 2 is

$$p_2 = \frac{N_2 \bar{R}T}{V}$$

The total pressure within the container is

$$p_t = p_1 + p_2 + p_3 + \dots = \frac{(N_1 + N_2 + N_3 + \dots) \bar{R}T}{V} = \frac{N_t \bar{R}T}{V} \quad (2.6)$$

From the above expression for p_1 and p_2 , it is seen that

$$\frac{p_1}{p_2} = \frac{N_1}{N_2}$$

and from Eq.2.6

$$\frac{p_1}{p_t} = \frac{N_1}{N_t} \quad (2.7)$$

The pressures p_1 , p_2 etc. which go to make up the total pressure are called partial pressure.

2.4 FORMS OF ENERGY

An engine takes in chemical energy in the fuel and changes most of this into sensible energy during the combustion process, and then transforms part of the sensible energy into mechanical work on the piston head.

During the suction stroke the atmosphere does work on the mixture as it forces it into the cylinder. During the exhaust stroke the products of combustion do work on the atmosphere as they leave the cylinder. In order to understand these processes, it is necessary to study the various forms of energy in some detail.

Energy is defined as the capacity to do work. If a machine or body, because of its position, temperature or velocity is capable of doing work on another body, it is said to have energy. Work is done whenever motion takes place against a resistance. The amount of work done is the distance moved, times the magnitude of the resisting force in the direction of motion.

$$\text{Work} = \text{Force} \times \text{distance}$$

Work and energy are expressed in the same units, viz., Nm or Joule.

When energy is being transferred from one body to another by virtue of a temperature difference, it is called heat and is usually measured in Joules.

Work and heat are related by the mechanical equivalent of heat J . The relation is

$$W = JH \quad (2.8)$$

where W is the work, H is the heat and $J = 427 \text{ kgf m/kcal}$. Thus work and heat can be expressed in any units of convenience with the help of J .

2.4.1 Transfer of Energy

Heat and work are both energy in transit. When two bodies at different temperatures are brought in contact, heat flows from a body at higher temperature to a body at lower temperature. Similarly, when work is done, the system doing work loses energy and the system upon which the work is being done gains exactly this amount of energy.

2.4.2 Stored Energy

In the two cases above, when the process is completed the energy gained by the second body is not called heat or work, since, heat is no longer flowing and work is no longer being done. The additional energy which is now stored in the second body may be called stored energy, E , which includes many types of energies like kinetic energy, potential energy, surface tension, electrical energy etc. depending upon the manner in which they are stored.

2.4.3 Potential Energy

It is the energy contained in the system by virtue of its elevation with reference to an arbitrarily chosen datum, usually the sea level. Or alternatively it is equivalent to energy required to raise the system from an arbitrary datum.

2.4.4 Kinetic Energy

Any body of mass, $m \text{ kg}$, moving with a velocity of $C \text{ metres per second}$ will do $\frac{1}{2}mC^2$ Joules of work before coming to rest. In this state of motion, the body is said to have kinetic energy equal to $\frac{1}{2}mC^2$ Joules.

2.4.5 Internal Energy

A body may possess energy due to the motion or position or the attraction between the particles of which it is made. In a permanent gas (i.e. a gas which is far from its liquefaction point), the internal energy, E , will usually be in the form of

- (i) translational motion of the molecules of the gas and
- (ii) motion of the atoms within the molecules

These two kinds of internal energy are known as sensible internal energy, although only that part which is due to molecular translation can actually be felt as warmth. For a given quantity of a particular permanent gas, the amount of sensible internal energy present is fixed by the temperature of the gas alone. The sensible internal energy is denoted by U_s . The gas under consideration may also contain chemical energy which is usually denoted by U_c .

The sum of the sensible internal energy and the chemical energy of the fuel present and is called the total internal energy, U , i.e.,

$$U = U_s + U_c \quad (2.9)$$

If during any combustion process no heat is allowed to escape from the gases and no work is permitted to be done by the gases, then at any time during the process the increase in sensible energy due to combustion is exactly equal to the loss in chemical energy as the fuel is used up. The total internal energy therefore remains unchanged.

2.5 FIRST LAW OF THERMODYNAMICS

Whenever a system undergoes a cyclic change, the algebraic sum of work transfer is proportional to the algebraic sum of heat transfer as work and heat are mutually convertible from one form into the other.

For a closed system, a change in the energy content is the algebraic difference between the heat supply, Q , and the work done, W , during any change in the state. Mathematically,

$$dE = \delta Q - \delta W \quad (2.10)$$

The energy E may include many types of energies like kinetic energy, potential energy, electrical energy, surface tension etc., but from the

thermodynamic point of view these energies are ignored and the energy due to rise in temperature alone is considered. It is denoted by U and the first law is written as:

$$dU = \delta Q - \delta W \quad (2.11)$$

or during a process 1-2, dU can be denoted as

$$U_2 - U_1 = \int_1^2 (\delta Q - \delta W) \quad (2.12)$$

2.6 PROCESS

A change in the condition or state of a substance is called a process. The process may consist of heating, flow from one place to another, expansion etc. In general, a process may be divided into non-flow or flow processes.

2.6.1 Non-flow Processes

If there is no flow of material into or out of a system during a process, it is called a non-flow process. This is the simplest kind of process, and much can be learned about it by applying the principle of conservation of energy.

2.7 ANALYSIS OF NON-FLOW PROCESSES

The purpose of the analysis is to apply the First Law of Thermodynamics to process in which a non-flow system changes from one state to the other and also to develop some useful relations.

2.7.1 Constant Volume or Isochoric Process

This process is usually encountered in the analysis of air-standard Otto, Diesel and Dual cycles. Figure 2.1 shows the constant volume process on a p - V diagram.

As there is no change in volume, the work $\int (pdV)$ is zero. Hence, according to the first law for the constant volume process the change in internal energy is equal to the heat transfer, i.e.,

$$dU = \delta Q = mC_v dT = mC_v(T_2 - T_1) \quad (2.13)$$

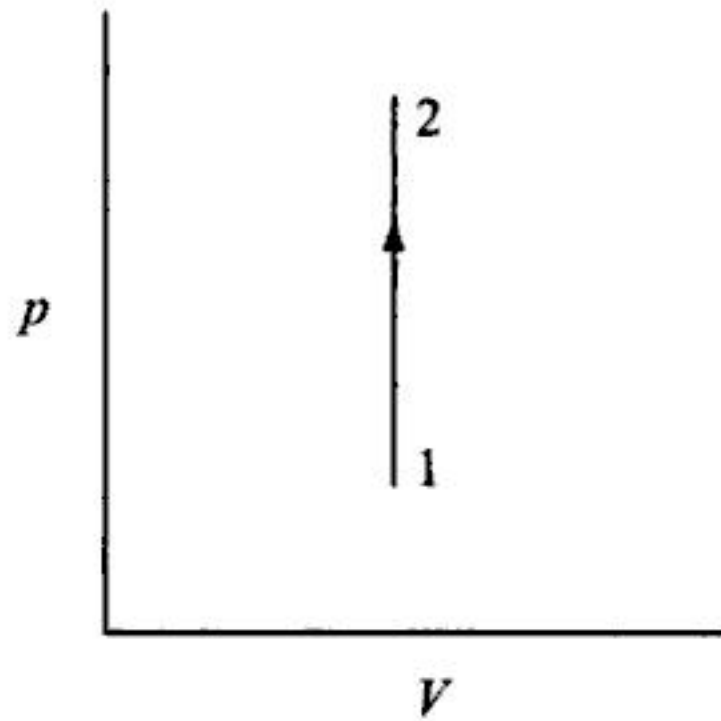


Fig. 2.1 Constant Volume Process

For unit mass,

$$du = \delta q = C_v dT$$

$$C_v = \left(\frac{du}{dT} \right)_v \quad (2.14)$$

i.e., specific heat at constant volume is the rate of change of internal energy with respect to absolute temperature.

2.7.2 Constant Pressure or Isobaric Process

Figure 2.2 shows a system that changes from state 1 to state 2 at constant pressure. Application of first law yields,

$$\delta Q = dU + p dV = d(U + pV) = dH \quad (2.15)$$

where H is known as the enthalpy. Thus, during constant pressure process, heat transfer is equal to change in enthalpy or

$$dH = \delta Q = m C_p dT \quad (2.16)$$

For unit mass,

$$\delta q = dh = C_p dT \quad (2.17)$$

$$C_p = \left(\frac{dh}{dT} \right)_p \quad (2.18)$$

i.e., specific heat at constant pressure is the rate of change of specific enthalpy with respect to absolute temperature.

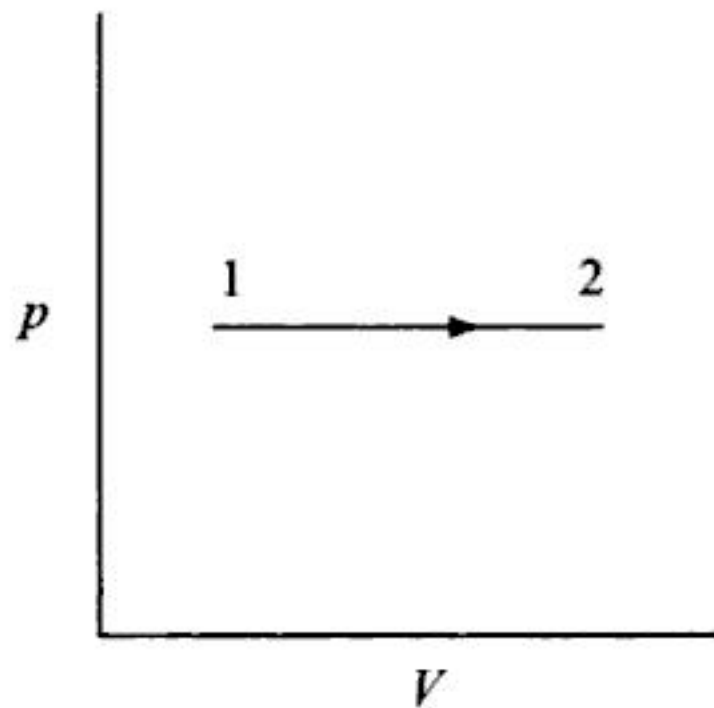


Fig. 2.2 Constant Pressure Process

2.7.3 Constant Temperature or Isothermal Process

The isothermal process on a p - V diagram is illustrated in Fig.2.3. As there is no temperature change during this process, there will not be any change in internal energy i.e., $dU = 0$, then according to the first law

$$\delta Q = \delta W \quad (2.19)$$

or

$$Q_{1-2} = \int_1^2 p dV = p_1 V_1 \log_e \left(\frac{V_2}{V_1} \right) \quad (2.20)$$

2.7.4 Reversible Adiabatic or Isentropic Process

If a process occurs in such a way that there is no heat transfer between the surroundings and the system, but the boundary of the system moves giving displacement work, the process is said to be adiabatic. Such a process is possible if the system is thermally insulated from the surroundings. Hence, $\delta Q = 0$, therefore,

$$\delta W = -\delta U = -mC_v dT \quad (2.21)$$

Reversible adiabatic process is also known as isentropic process. Let $pV^\gamma = C$ be the law of the isentropic process. For unit mass flow,

$$q_{1-2} = 0 = w_{1-2} + u_2 - u_1 \quad (2.22)$$

or

$$w_{1-2} = -(u_2 - u_1)$$

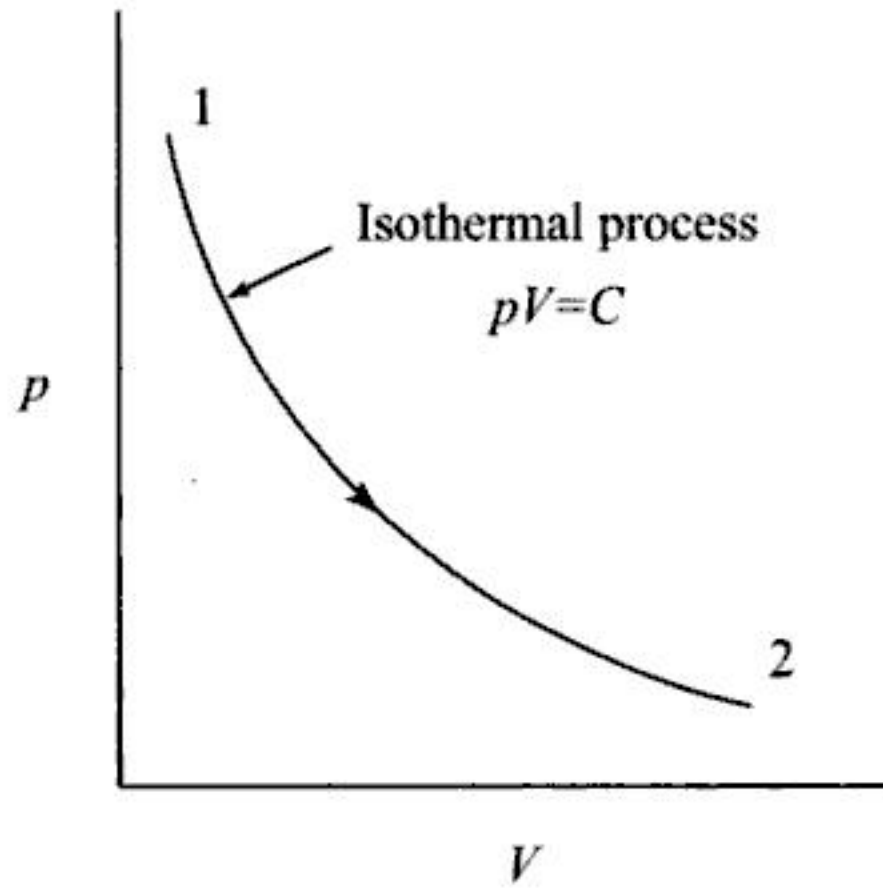


Fig. 2.3 Constant Temperature Process

In other words, work is done at the expense of internal energy

$$\begin{aligned}
 W_{1-2} &= \int_1^2 p dV = \int_1^2 \frac{C}{V^\gamma} dV \\
 &= \frac{[CV^{1-\gamma}]_{V_1}^{V_2}}{1-\gamma} = \frac{CV_2^{1-\gamma} - CV_1^{1-\gamma}}{1-\gamma}
 \end{aligned} \quad (2.23)$$

when $C = p_1 V_1^\gamma = p_2 V_2^\gamma$,

$$\begin{aligned}
 W_{1-2} &= \frac{p_2 V_2^\gamma V_2^{1-\gamma} - p_1 V_2^\gamma V_2^{1-\gamma}}{1-\gamma} \\
 &= \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}
 \end{aligned}$$

Using $pV = RT$ for unit mass flow, we have,

$$\begin{aligned}
 w_{1-2} &= \frac{R(T_1 - T_2)}{\gamma - 1} \\
 &= C_v(T_1 - T_2) = -(u_2 - u_1) \\
 \frac{p_1 V_1}{T_1} &= \frac{p_2 V_2}{T_2}
 \end{aligned}$$

therefore,

$$\frac{T_2}{T_1} = \frac{p_2 V_2}{p_1 V_1} = \left(\frac{V_1}{V_2}\right)^\gamma \left(\frac{V_2}{V_1}\right)$$

$$= \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} \quad (2.24)$$

$$\begin{aligned} \frac{T_2}{T_1} &= \frac{p_2 V_2}{p_1 V_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} \left(\frac{p_2}{p_1}\right) \\ &= \left(\frac{p_2}{p_1}\right)^{\left(1-\frac{1}{\gamma}\right)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \end{aligned} \quad (2.25)$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \quad (2.26)$$

2.7.5 Reversible Polytropic Process

In polytropic process, both heat and work transfers take place. It is denoted by the general equation $pV^n = C$, where n is the polytropic index. The following equations can be written by analogy to the equations for the reversible adiabatic process which is only a special case of polytropic process with $n = \gamma$. Hence, for a polytropic process

$$p_1 V_1^n = p_2 V_2^n \quad (2.27)$$

$$\begin{aligned} \frac{T_1}{T_2} &= \left(\frac{V_2}{V_1}\right)^{n-1} \\ \frac{T_1}{T_2} &= \left(\frac{p_1}{p_2}\right)^{\left(\frac{n-1}{n}\right)} \end{aligned} \quad (2.28)$$

and

$$w_{1-2} = \frac{p_1 V_1 - p_2 V_2}{n - 1} \quad (2.29)$$

2.7.6 Heat Transfer During Polytropic Process

Heat transfer per unit mass,

$$\begin{aligned} q_{1-2} &= (u_2 - u_1) + \int_1^2 p dV \\ &= C_v(T_2 - T_1) + \frac{R(T_2 - T_1)}{1 - n} \\ &= \left(C_v + \frac{R}{1 - n}\right)(T_2 - T_1) \\ &= \left(C_v + \frac{C_p - C_v}{1 - n}\right)(T_2 - T_1) \end{aligned} \quad (2.30)$$

$$\begin{aligned}
&= \left(\frac{C_p - nC_v}{1 - n} \right) (T_2 - T_1) \\
&= \left(\frac{C_v}{1 - n} \right) \left(\frac{C_p}{C_v} - n \right) (T_2 - T_1) \\
&= \left(\frac{C_v}{1 - n} \right) (\gamma - n) (T_2 - T_1) \\
&= \left(\frac{\gamma - n}{1 - n} \right) C_v (T_2 - T_1) \quad (2.31)
\end{aligned}$$

Hence,

$$q_{1-2} = C_n (T_2 - T_1)$$

where,

$$C_n = \left(\frac{\gamma - n}{1 - n} \right) C_v \quad (2.32)$$

Table 2.1 gives the formulae for various process relations for easy reference.

2.8 ANALYSIS OF FLOW PROCESS

Consider a device shown in the Fig.2.4 through which a fluid flows at uniform rate and which absorbs heat and does work, also at a uniform rate. In engines, the gas flow, heat flow and work output vary throughout each cycle, but if a sufficiently long time interval (such as a minute) be chosen, then, even an engine can be considered to be operating under steady-flow conditions.

Applying the principle of conservation of energy to such a system during the chosen time interval, the total energy in the mass of fluid which enters the machine across boundary 1-1 plus the heat added to the fluid through the walls of the machine, minus the work done by the machine, must equal the total energy left in the equal mass of fluid crossing boundary 2-2 and leaving the machine. Expressed in the mathematical form,

$$Q - W_x = m \left[\left(h_2 + \frac{C_2^2}{2} + gz_2 \right) - \left(h_1 + \frac{C_1^2}{2} + gz_1 \right) \right]$$

or

$$\frac{Q - W_x}{m} = \Delta \left(h + \frac{C^2}{2} + gz \right) \quad (2.33)$$

Table 2.1 Summary of Process Relations for a Perfect Gas

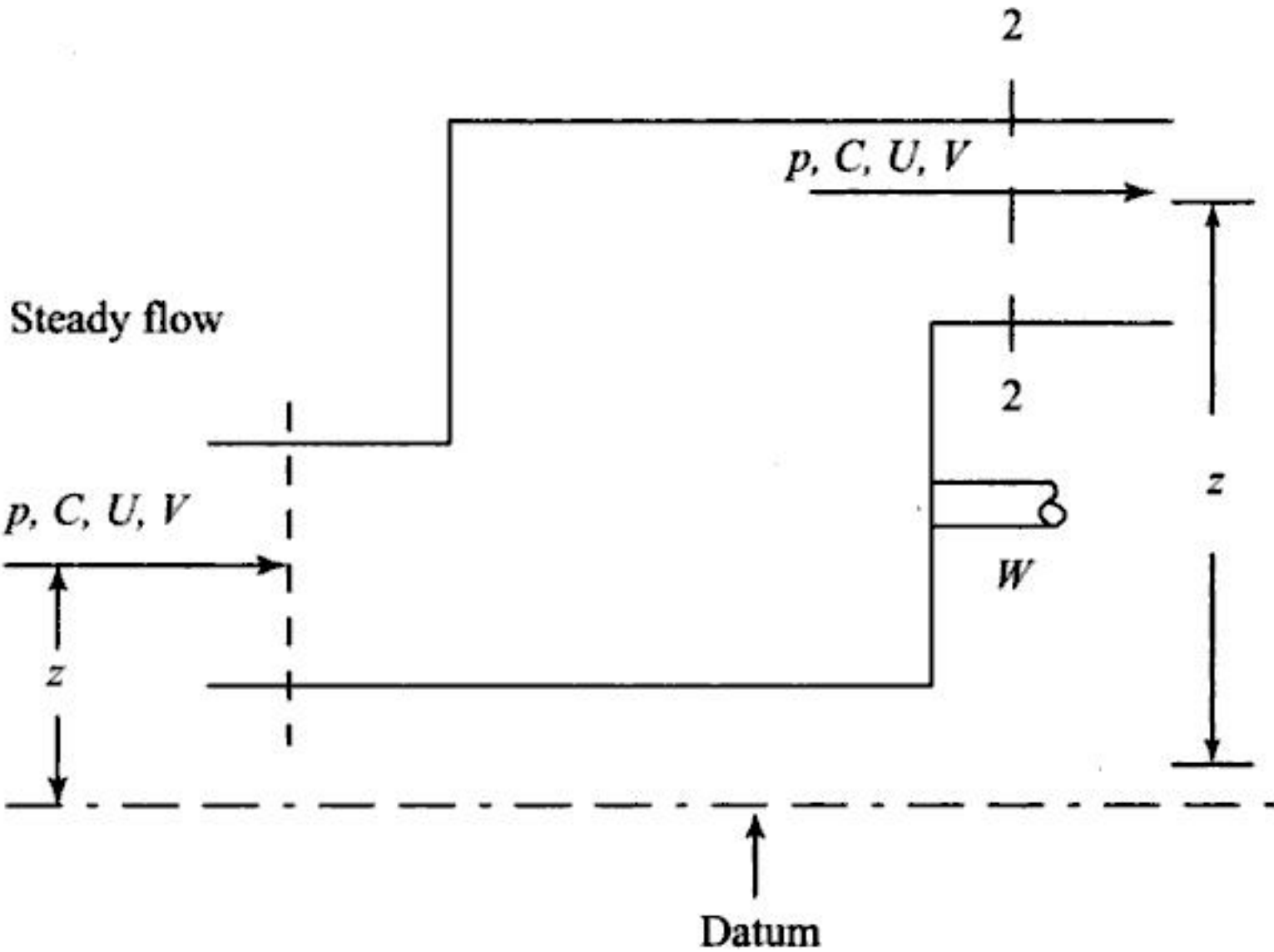
Process	Index n	p, V, T relation	Heat Transfer	Work $\int p dV$	Work $\int V dp$	ΔU
Constant Volume	∞	$T_1/T_2 = p_1/p_2$	$mC_v(T_2 - T_1)$	0	$V(p_2 - p_1)$	$mC_v(T_2 - T_1)$
Constant Pressure	0	$T_1/T_2 = V_1/V_2$	$mC_p(T_2 - T_1)$	$p(V_2 - V_1)$	0	$mC_v(T_2 - T_1)$
Isothermal	1	$p_1 V_1 = p_2 V_2$	$p_1 V_1 \log_e \left(\frac{V_2}{V_1} \right)$	$p_1 V_1 \log_e \left(\frac{V_2}{V_1} \right)$	$p_1 V_1 \log_e \left(\frac{V_1}{V_2} \right)$	0
Isentropic	γ	$p_1 V_1^\gamma = p_2 V_2^\gamma$ $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1} \right)^{\gamma-1}$ $= \left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}}$	0	$\frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$	$\frac{\gamma}{\gamma-1} (p_2 V_2 - p_1 V_1)$	$mC_v(T_2 - T_1)$
Polytropic	n	$p_1 V_1^n = p_2 V_2^n$ $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1} \right)^{n-1}$ $= \left(\frac{p_1}{p_2} \right)^{\frac{n-1}{n}}$	$\left(\frac{\gamma-n}{1-n} \right) mC_v(T_2 - T_1)$	$\frac{p_1 V_1 - p_2 V_2}{n - 1}$	$\frac{n}{n-1} (p_2 V_2 - p_1 V_1)$	$mC_v(T_2 - T_1)$

In the above equation, m denotes mass. It is known as the general energy equation and is the one which illustrates the first law of thermodynamics.

Here, W_x is the external work and given by

$$W = W_x - mp_1V_1 + mp_2V_2 \quad (2.34)$$

where W is the net work and V stands for volume.



- | | | |
|---------------------------------|--------------------------|------------------------------|
| $U \rightarrow$ Internal energy | $H \rightarrow$ Enthalpy | $z \rightarrow$ Datum height |
| $p \rightarrow$ Pressure | $V \rightarrow$ Volume | $C \rightarrow$ Velocity |
| $U \rightarrow$ Internal energy | $H \rightarrow$ Enthalpy | $z \rightarrow$ Datum height |

Fig. 2.4 Illustration of Steady Flow Process

2.8.1 Flow Work

The term pV in the above equation is called flow work and must be considered whenever flow takes place. It is the work necessary either to push certain quantity of mass into or out of the system.

2.9 WORK, POWER AND EFFICIENCY

In this section let us define work, power and efficiency and also the relation between them.

2.9.1 Work

Work might be defined as the energy expended when motion takes place in opposition to a force. Work thus represents energy being transferred from one place to another.

2.9.2 Power

Power is the rate of doing work. It is generally useless to do a certain piece of work unless it can be done within a reasonable length of time. When work is done at the rate of 736 Nm/s or 75 kgf m/s, the quantity of power being developed is known as one horse power. In SI units power is expressed in kW = 1 kN m/s.

2.9.3 Efficiency

It is common practice in engineering to establish a figure of merit for a device by comparing the actual performance of the device with the performance it would have had under some arbitrary set of ideal conditions. The ratio of actual performance to the ideal performance is called the efficiency of the device. Since the ideal performance is usually unattainable, the efficiency is usually less than 1. In internal combustion engines, one of the most important efficiencies is the thermal efficiency, which is defined as

$$\begin{aligned}\text{Thermal efficiency} = \eta_{th} &= \frac{\text{Work delivered by the engine}}{\text{Chemical energy in the fuel}} \\ &= \frac{\text{Work output}}{\text{Fuel energy input}}\end{aligned}$$

Obviously, the output and input may be measured over any convenient interval such as one cycle or one minute. The output and input energies must be expressed in same kind of units, since η_{th} is a number without units.

Let W be the work delivered by the engine (Nm) and m_f be the mass of fuel (kg) required and CV be the calorific value of the fuel (kJ/kg), then the efficiency η_{th} is given by

$$\eta_{th} = \frac{W}{m_f CV} \quad (2.35)$$

If W is the indicated work then η_{th} is called the indicated thermal efficiency, η_{ith} , and if W is the work delivered at the output shaft then it is called the brake thermal efficiency, η_{bth} . The ratio of the

brake thermal efficiency to the indicated thermal efficiency is called the mechanical efficiency, η_m . These efficiencies have already been defined in Chapter 1.

Worked out Examples

2.1 A closed system undergoes a thermodynamic cycle consisting of four separate and distinct processes. The work and heat transfer for each of these processes is given below

Process	Heat Transfer (J/s)	Work Transfer (Nm/s)
1-2	7000	5300
2-3	-3500	9100
3-4	17500	8700
4-1	0	-2100

Show that the above data is in accordance with the first law of thermodynamics and determine : (i) Net rate of work output (ii) Efficiency of the cycle (iii) Change in internal energy in each process

Solution

For the above closed cycle 1-2-3-4-1,

$$\begin{aligned}\oint \delta Q &= 7000 - 3500 + 17500 + 0 \\ &= 21000 \text{ J/s} \\ \oint \delta W &= 5300 + 9100 + 8700 - 2100 \\ &= 21000 \text{ Nm/s} = 21000 \text{ J/s}\end{aligned}$$

As $\oint \delta Q = \oint \delta W$, the above cycle satisfies the first law of thermodynamics

$$\begin{aligned}\text{Power} &= \text{Net rate of work} = 21000 \text{ J/s} \\ &= 21 \text{ N/m}^2\text{kW} \quad \text{Ans}\end{aligned}$$

$$\begin{aligned}\text{Thermal efficiency} &= \frac{\text{Net work}}{\text{Heat input}} = \frac{21000}{7000 + 17500} \times 100 \\ &= 85.71\% \quad \text{Ans}\end{aligned}$$

Change in internal energy of each process can be estimated by using the relation, $\delta Q = du + \delta W$

Process	δQ J/s	δW Nm/s	$du = \delta Q - \delta W$ J/s
1-2	7000	5300	1700
2-3	-3500	9100	-12600
3-4	17500	8700	8800
4-1	0	-2100	2100

2.2 Air at the rate of 10 kg/min flows steadily through an air compressor in which the inlet and discharge pipe lines are at the same level. Following data regarding the fluid are available:

	At inlet	At outlet
Fluid velocity	5 m/s	10 m/s
Fluid pressure	1 bar	8 bar
Specific volume	0.5 m ³ /kg	0.20 m ³ /kg

The internal energy of the air leaving the compressor was 250 kJ/kg greater than that entering and during the process the system lost 140 kJ/s of energy dissipated as heat to the cooling water and to the environment, Find (i) Rate of shaft work input to the air (ii) Ratio of inlet pipe diameter to outlet pipe diameter

Solution

Consider 1 kg/min of mass flow :

Heat lost to cooling water and environment

=

$\frac{140 \times 60}{10}$

=

840 kJ/kg

Applying steady flow energy equation, (the potential energy changes are neglected as the inlet and outlet pipes are at the same level).

$$U_1 + p_1V_1 + \frac{C_1^2}{2} + q = U_2 + p_2V_2 + \frac{C_2^2}{2} + w$$

therefore,

$$w = (U_1 - U_2) + (p_1V_1 - p_2V_2) + \frac{C_1^2 - C_2^2}{2} + q$$

$$= -250 \times 10^3 + (1 \times 10^5 \times 0.5 - 8 \times 10^5 \times 0.2) + \frac{5^2 - 10^2}{2} - 140 \times 10^3$$

$$= -500037.5 \text{ J/kg} \approx -500 \text{ kJ/kg}$$

$$W = mw = 10 \times (-500)$$

$$= -5000 \text{ kJ/min} = -83.33 \text{ kJ/s}$$

$$\text{Power input} = 83.33 \text{ N/m}^2 \text{ kW} \quad \underline{\underline{\text{Ans}}}$$

From continuity equation, $A_1 C_1 \rho_1 = A_2 C_2 \rho_2$. Since, the specific volume, $v_1 = \frac{1}{\rho_1}$. We have,

$$\frac{A_1 C_1}{v_1} = \frac{A_2 C_2}{v_2}$$

$$\frac{A_1}{A_2} = \frac{v_1 C_2}{v_2 C_1} = \frac{0.5 \times 10}{0.2 \times 5} = 5$$

$$\frac{D_1}{D_2} = \sqrt{5} = 2.236 \quad \underline{\underline{\text{Ans}}}$$

2.3 A tank of volume 0.1 m^3 contains 4 kg nitrogen, 1.5 kg oxygen and 0.75 kg carbon dioxide. If the temperature of the mixture is 20°C , determine : (i) the total pressure of the mixture, (ii) the gas constant of the mixture. (Given that $R_{\text{N}_2} = 296.8 \text{ J/kg K}$, $R_{\text{O}_2} = 259.83 \text{ J/kg K}$ and $R_{\text{CO}_2} = 188.9 \text{ J/kg K}$).

Solution

$$\begin{aligned} \text{N}_2 &: m_1 = 4.00 \text{ kg}; \quad t_1 = t = 20^\circ \text{C}; \quad V_1 = V = 0.1 \text{ m}^3 \\ \text{O}_2 &: m_2 = 1.50 \text{ kg}; \quad t_2 = t = 20^\circ \text{C}; \quad V_2 = V = 0.1 \text{ m}^3 \\ \text{CO}_2 &: m_3 = 0.75 \text{ kg}; \quad t_3 = t = 20^\circ \text{C}; \quad V_3 = V = 0.1 \text{ m}^3 \end{aligned}$$

Assumptions :

- (i) the component gases and the mixture behave like an ideal gas
- (ii) the mixture obeys the Gibbs-Dalton law

For N_2 :

$$\begin{aligned} p_1 &= \frac{m_1 R_{N_2} T}{V} \\ &= \frac{4 \times 296.8 \times (273 + 20)}{0.1} = 34.78 \times 10^5 \text{ N/m}^2 \end{aligned}$$

For O_2 :

$$\begin{aligned} p_2 &= \frac{m_2 R_{O_2} T}{V} \\ &= \frac{1.5 \times 259.83 \times (273 + 20)}{0.1} = 11.42 \times 10^5 \text{ N/m}^2 \end{aligned}$$

For CO_2 :

$$\begin{aligned} p_3 &= \frac{m_3 R_{CO_2} T}{V} \\ &= \frac{0.75 \times 188.9 \times (273 + 20)}{0.1} = 4.15 \times 10^5 \text{ N/m}^2 \end{aligned}$$

Then, total pressure, P

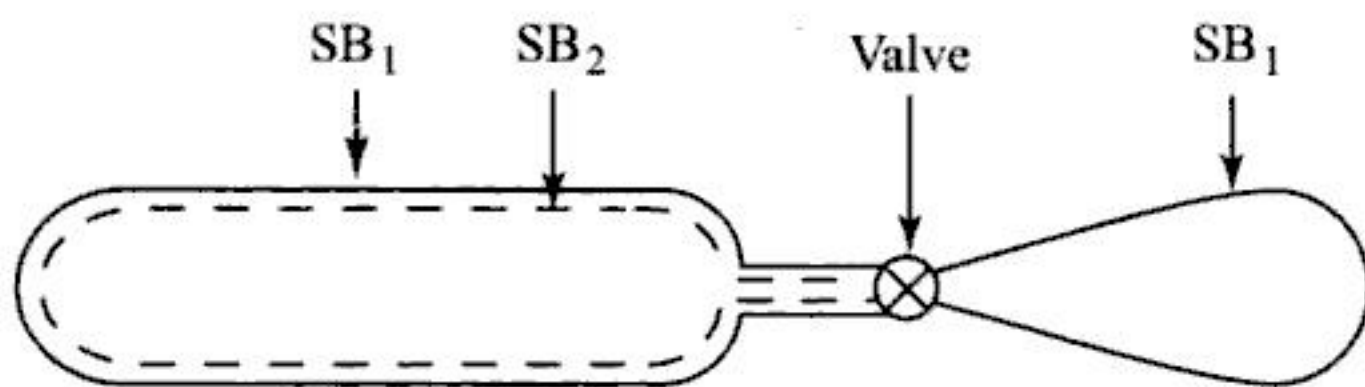
$$\begin{aligned} p &= p_1 + p_2 + p_3 = (34.78 + 11.42 + 4.15) \times 10^5 \\ &= \mathbf{50.35 \times 10^5 \text{ N/m}^2} \end{aligned} \quad \underline{\underline{\text{Ans}}}$$

$$\begin{aligned} R &= \frac{(m_{N_2} R_{N_2} + m_{O_2} R_{O_2} + m_{CO_2} R_{CO_2})}{m} \\ &= \frac{(4 \times 296.8 + 1.5 \times 259.83 + 0.75 \times 188.92)}{4 + 1.5 + 0.75} \\ &= \mathbf{274.98 \text{ N/m}^2 \text{ J/kg K}} \end{aligned} \quad \underline{\underline{\text{Ans}}}$$

2.4 Determine the work done by the air which enters an evacuated bottle from the atmosphere when the cock is opened. The atmospheric pressure is $1.013 \times 10^5 \text{ N/m}^2$ and 0.3 m^3 of air (measured at atmospheric conditions) enter.

Solution

No work is done by the part of the boundary in contact with the bottle. Only the moving external part need to be considered. Over



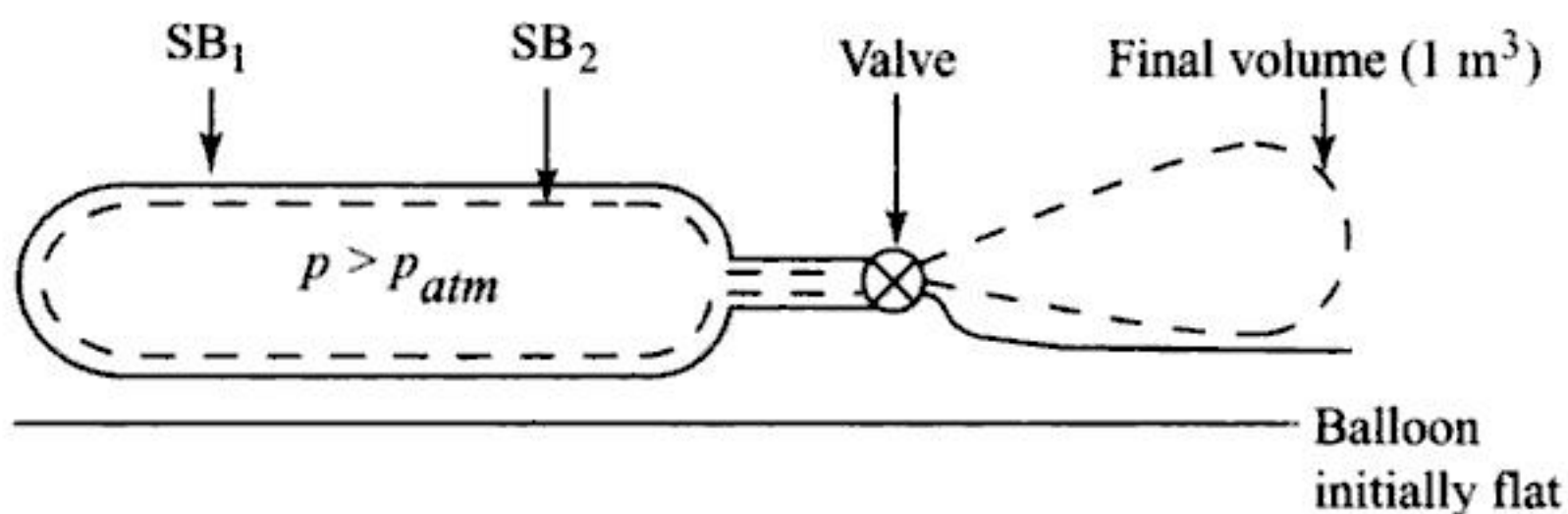
this part pressure is uniform at $1.013 \times 10^5 \text{ N/m}^2$, therefore,

$$\begin{aligned}
 W_d &= \int_{\text{free-air boundary}} p dV + \int_{\text{Bottle}} p dV \\
 &= P_{atm} \int dV + 0 = 1.013 \times 10^5 (-0.3) \\
 &= -30390 \text{ N/m}^2 \text{ Nm} \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

The work is negative as the boundary is contracting.

2.5 A balloon of flexible material is to be filled with air from a storage bottle until it has a volume of 1 m^3 . The atmospheric pressure is $1.01 \times 10^5 \text{ N/m}^2$. Determine the work done by the system comprising the air initially in the bottle, given that the balloon is light and requires no stretching.

Solution



Initially the system boundary coincides with the inner surface of the storage bottle. At the end of the process the boundary also encloses the 1 m^3 content of the balloon.

The displacement work which is the only work in this process, is obtained by taking the summation of the values of $\int p dV$ for each part of the boundary. As there is no change in the volume of the bottle, dV is zero for the part of the boundary which is in contact

with the bottle surface. Hence the pressure inside the cylinder is not necessary in the calculation. Therefore,

$$\begin{aligned}
 W_{\text{displacement}} &= \int_{\text{Balloon}} p dV + \int_{\text{Bottle}} p dV \\
 &= p_{\text{atm}} \int dV + 0 = 1.01 \times 10^5 \times 1.0 \\
 &= 1.01 \times 10^5 \text{ N/m}^2 \text{ Nm} \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

Review Questions

- 2.1 What are the main objectives in studying the theory of I.C. engines?
- 2.2 What are the important fundamental quantities used in the application and analysis of I.C. engines? Explain them briefly.
- 2.3 What is meant by mole?
- 2.4 For a mixture of gases show that $P_1/P_t = N_1/N_t$
- 2.5 What are the various forms of energy normally used in engine applications. Briefly explain them.
- 2.6 What are isochoric and isobaric processes? Explain them.
- 2.7 For an isentropic process, show that (i) $T_2/T_1 = (V_1/V_2)^{\gamma-1}$ and (ii) $T_2/T_1 = (p_1/p_2)^{(\gamma-1)/\gamma}$
- 2.8 Derive an expression for heat transfer during a polytropic process.
- 2.9 Explain how an internal combustion engine can be analyzed from the point of view of steady flow process.
- 2.10 Define: (i) work (ii) power and (iii) efficiency

Exercise

- 2.1 The piston of an oil engine, of area 50 cm^2 , moves downwards 10 cm and draws in 300 cc of fresh air from the atmosphere. The pressure in the cylinder is uniform during the process at 0.8 bar , while the atmospheric pressure is 1.013 bar . The difference in the pressure is accounted for by flow resistance in the induction

pipe and inlet valve. Determine the displacement work done by the air finally in the cylinder. *Ans:* -9.61 J

- 2.2 Air at 600 K and 1.2 bar undergoes an adiabatic expansion. The new value of pressure is 0.6 bar. The mass of the air, which may be treated as an ideal gas, is 0.12 kg. Determine (i) the work done by the air and (ii) the change in the internal energy of the air. *Ans:* (i) 9.285 kJ (ii) -9.285 kJ

- 2.3 An open tank is filled to the brim with a liquid of density 1100 kg/m^3 . A spherical balloon 0.5 m in diameter is immersed in the liquid with its centre 2.5 m below the free liquid surface. Gas from a storage vessel is used to inflate the balloon thereby causing the tank to overflow. The atmospheric pressure is 10^5 N/m^2 . Evaluate the work done by the gas in the balloon on the storage vessel as the balloon diameter increases to 1 m. Assume $g = 9.81 \text{ m/s}^2$. *Ans:* $58.2 \times 10^3 \text{ Nm}$

- 2.4 A gas flows steadily through a rotary compressor. The gas enters the compressor at a temperature of 16° , a pressure of 10^5 N/m^2 and an enthalpy of 391.2 KJ/kg. The gas leaves the compressor at a temperature of 245°C , a pressure of $6 \times 10^5 \text{ N/m}^2$ and an enthalpy of 534.5 KJ/kg. There is no net heat transfer to or from the gas as it flows through the compressor.

- (i) Evaluate the external work done per unit mass of gas assuming the gas velocities at entry and exit to be negligible.
- (ii) Evaluate the external work done per unit mass of gas when the gas velocity at entry is 80 m/s and at exit is 160 m/s.

Ans: (i) 143.3 KJ/kg (ii) 152.9 KJ/kg

- 2.5 The relation between the properties of gaseous oxygen gas may be expressed over a restricted range by :

$$Pv = 260t + 71 \times 10^3 \quad \text{and} \quad t = 1.52u - 273$$

where P is in N/m^2 , v in m^3/kg , t in $^\circ \text{C}$ and u in kJ/kg .

- (i) Evaluate the specific heat at constant volume and specific heat at constant pressure in kJ/kg K .
- (ii) Show that for any process executed by unit mass of oxygen, $\Delta u = C_v \Delta t$ and $\Delta h = C_p \Delta t$

Ans: (i) 0.658 kJ/kg K (ii) 0.918 kJ/kg K

- 2.6 Two kg of a gas at 10 bar expands adiabatically and reversibly till the pressure falls to 5 bar. During the process 170 kJ of non-flow work is done by the system, and the temperature drops from 377 °C to 257 °C. Calculate (i) the value of the index of expansion and (ii) characteristic gas constant.

Ans: (i) 1.416 (ii) 294 J/kg K

- 2.7 Show that for a Vander Waals gas which has the equation of state

$$\left(P + \frac{a}{V_2} = RT \right)$$

The work done at constant temperature per unit mass of gas is

$$RT \ln \frac{V_2 - b}{V_1 - b} - a \left(\frac{1}{V_1} - \frac{1}{V_2} \right)$$

where V_1 and V_2 denote the initial and final specific volumes respectively.

- 2.8 In an air-standard cycle, heat is supplied at constant volume resulting in an increase in temperature of air from T_1 to T_2 . The air is then expanded isentropically till its temperature falls to T_1 . Finally, it is returned to its original state by a reversible isothermal compression process.

Show that the efficiency of the cycle is given by

$$\eta = 1 - \frac{T_1}{T_2 - T_1} \ln \frac{T_2}{T_1}$$

- 2.9 A volume of 5 m³ of air at a pressure of 1 bar and 27 °C is compressed adiabatically to 5 bar. The compressed air is then expanded isothermally to original volume. Find :

- (i) The final pressure of the air after expansion
- (ii) The quantity of heat added from the beginning of compression to the end of expansion
- (iii) The quantity of heat that must be added or subtracted to reduce the air after expansion to the original state of pressure, volume and temperature.

Ans: (i) 1.58 bar (ii) 910.09 kJ (iii) -729.8 kJ

- 2.10 A perfect gas of molecular weight 30 has the ratio of specific heats 1.3. One kg of the gas at 1 bar and 27 °C is compressed adiabatically inside a cylinder to 7 times its

initial pressure. Heat is now extracted from the system at constant pressure until it reaches a state (temperature 27°C) such that the gas when expanded isothermally from this state will reach the state before adiabatic compression. Calculate the heat transfer during constant pressure cooling and the work done during isothermal expansion. What is the amount of work transfer during the cycle and in which direction does it take place. $R = 0.287 \text{ kJ/kg K}$.

Ans: (i) -204 kJ/kg (ii) 161.8 kJ/kg (iii) -42.2 kJ/kg

- 2.11 A perfect gas flows steadily through a horizontal cooler. The mass flow rate through the cooler is 1 kg/s . The pressure and temperature are 2 bar and 127°C at entry and 1.5 bar and 7°C at exit. The cross sectional areas at entry and exit are each 80 cm^2 . Calculate the velocities of the gas at entry and exit of the cooler and the heat transfer rate in the cooler. Take for the gas $C_v = 0.70 \text{ kJ/kg K}$ and $R = 160 \text{ J/kg K}$.

Ans: (i) 40 m/s (ii) 37.5 m/s (iii) -103.3 kJ/s

- 2.12 Air inside a cylinder is compressed from the same initial state, without friction such that the compression ratio, $\frac{V_2}{V_1}$ is 15. In one case it is compressed isothermally and in the other case it is compressed polytropically with polytropic index, $n = 1.3$. Calculate the ratios of work done and heat transfer in two cases. What will be the ratio of final pressure in the two cases.

Ans: (i) 0.649 (ii) 2.59 (iii) 0.444

3

AIR-STANDARD CYCLES AND THEIR ANALYSIS

3.1 INTRODUCTION

The operating cycle of an internal combustion engine can be broken down into a sequence of separate processes viz., intake, compression, combustion, expansion and exhaust. The internal combustion engine does not operate on a thermodynamic cycle as it involves an open system i.e., the working fluid enters the system at one set of conditions and leaves at another. However, it is often possible to analyze the open cycle as though it were a closed one by imagining one or more processes that would bring the working fluid at the exit conditions back to the condition of the starting point.

The accurate analysis of internal combustion engine processes is very complicated. In order to understand them it is advantageous to analyze the performance of an idealized closed cycle that closely approximates the real cycle. One such approach is the air-standard cycle, which is based on the following assumptions:

- (i) The working medium is assumed to be a perfect gas and follows the relation $pV = mRT$ or $p = \rho RT$.
- (ii) There is no change in the mass of the working medium.
- (iii) All the processes that constitute the cycle are reversible.
- (iv) Heat is assumed to be supplied from a constant high temperature source and not from chemical reactions during the cycle.

- (v) Some heat is assumed to be rejected to a constant low temperature sink during the cycle.
- (vi) It is assumed that there are no heat losses from the system to the surroundings.
- (vii) The working medium has constant specific heats throughout the cycle.
- (viii) The physical constants viz., C_p , C_v , γ and M of working medium are the same as those of air at standard atmospheric conditions. For example in SI units,

$$\begin{array}{ll} C_p = 1.005 \text{ kJ/kg K} & M = 29 \text{ kg/kmol} \\ C_v = 0.717 \text{ kJ/kg K} & \gamma = 1.4 \end{array}$$

Due to these assumptions, the analysis becomes over-simplified and the results do not agree with those of the actual engine. Work output, peak pressure, peak temperature and thermal efficiency based on air-standard cycles will be the maximum that can be attained and will differ considerably from those of the actual engine. It is often used, mainly because of the simplicity in getting approximate answers to the complicated processes in internal combustion engines.

In this chapter, we will review the various cycles and also derive the equations for work output, mean effective pressure, efficiency etc. Also, comparison will be made between Otto, Dual and Diesel cycles to see which cycle is more efficient under a set of given operating conditions.

3.2 THE CARNOT CYCLE

Sadi Carnot, a French engineer, proposed a reversible cycle in 1824, in which the working medium receives heat at a higher temperature and rejects heat at a lower temperature. The cycle will consist of two isothermal and two reversible adiabatic processes as shown in Fig.3.1. Carnot cycle is represented as a standard of perfection and engines can be compared with it to judge the degree of perfection. It gives the concept of maximizing work output between two temperature limits.

The working of an engine based on the Carnot cycle can be explained referring to Fig.3.2 which shows a cylinder and piston arrangement working without friction. The walls of cylinder are assumed to be perfect insulators. The cylinder head is so arranged

that it can be a perfect heat conductor as well as a perfect heat insulator.

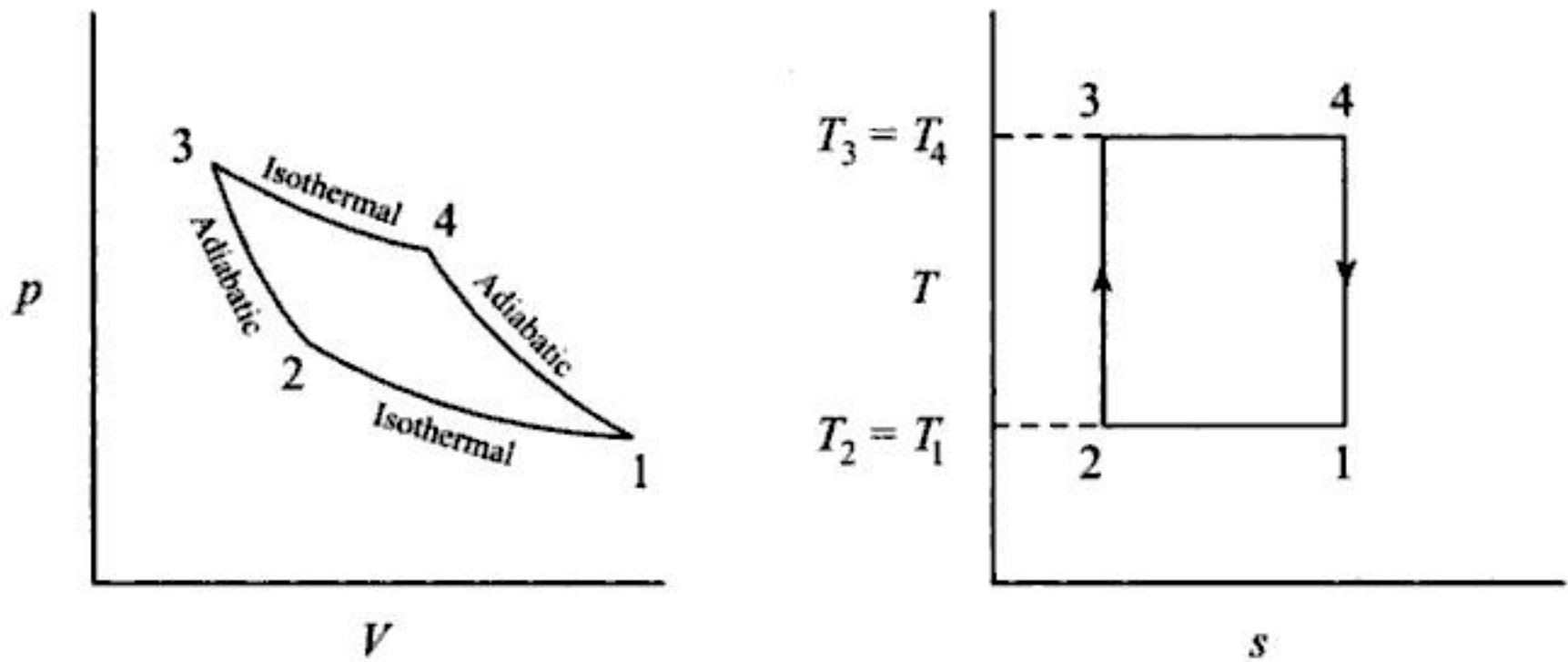


Fig. 3.1 Carnot Engine

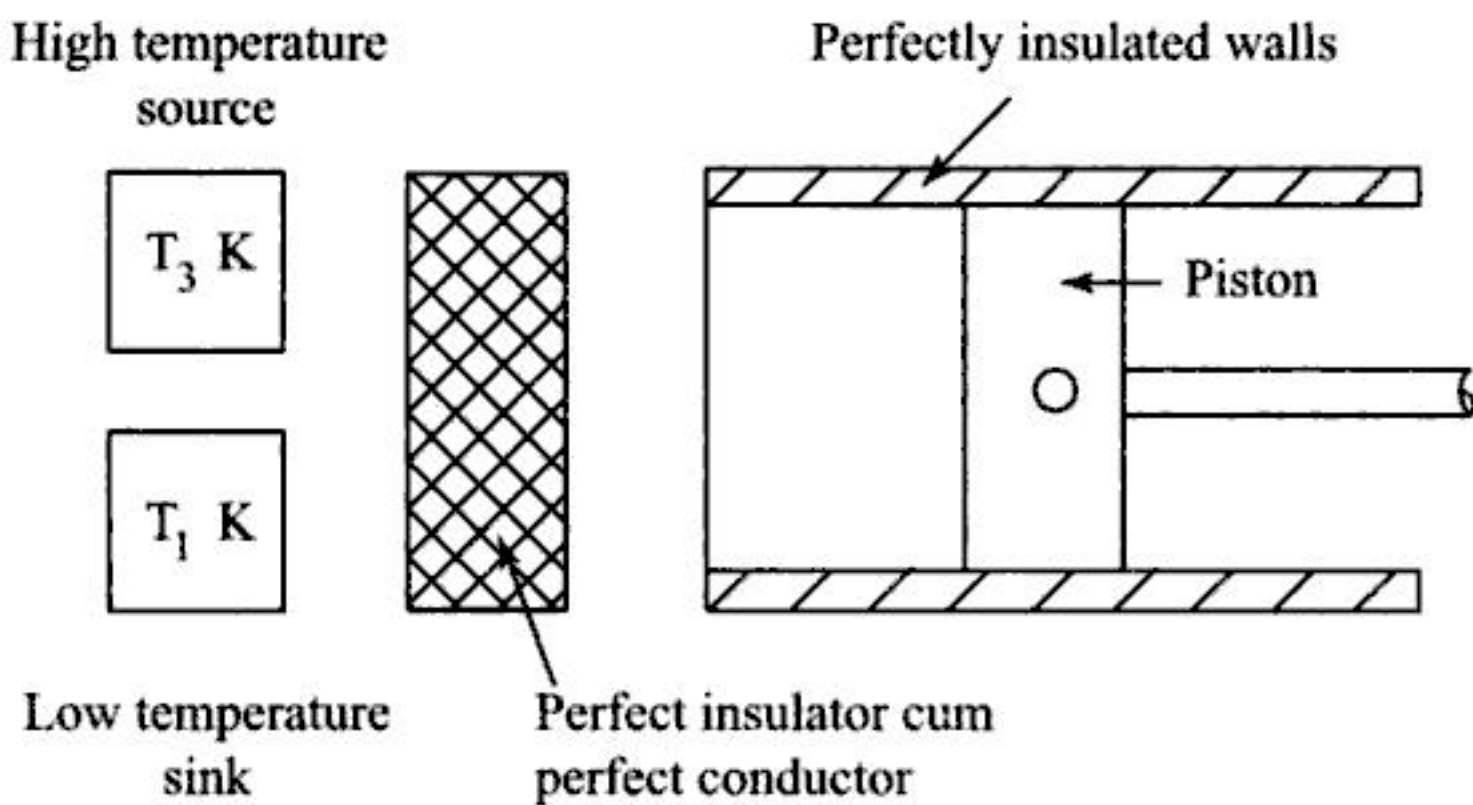


Fig. 3.2 Working Principle of a Carnot Engine

First the heat is transferred from a high temperature source, (T_3), to the working medium in the cylinder and as a result the working medium expands. This is represented by the isothermal process 3→4 in Fig.3.1. Now the cylinder head is sealed and it acts as a perfect insulator. The working medium in the cylinder is now allowed to expand further from state 4 to state 1 and is represented by reversible adiabatic process 4→1 in p - V and T - s diagrams in Fig.3.1. Now the system is brought into contact with a constant low temperature sink, (T_1), as the cylinder head is now made to act as a perfect heat conductor. Some heat is rejected to the sink without altering the

temperature of sink and as a result the working medium is compressed from state 1 to 2 which is represented by isothermal line 1→2. Finally the cylinder head is made again to act as a perfect insulator and the working medium is compressed adiabatically from state 2 to 3 which is represented by process 2→3. Thus the cycle is completed.

Analyzing the cycle thermodynamically the efficiency of the cycle can be written as

$$\eta_{Carnot} = \frac{\text{Work done by the system during the cycle (W)}}{\text{Heat supplied to the system during the cycle (Q}_S\text{)}}$$

According to the first law of thermodynamics,

$$\text{Work done} = \text{Heat supplied} - \text{Heat rejected}$$

$$W = Q_S - Q_R \quad (3.1)$$

Considering the isothermal processes 1→2 and 3→4, we get

$$Q_R = mRT_1 \log_e \frac{V_1}{V_2} \quad (3.2)$$

$$Q_S = mRT_3 \log_e \frac{V_4}{V_3} \quad (3.3)$$

Considering adiabatic processes 2→3 and 4→1

$$\frac{V_3}{V_2} = \left(\frac{T_2}{T_3} \right)^{\left(\frac{1}{\gamma-1} \right)} \quad (3.4)$$

and

$$\frac{V_4}{V_1} = \left(\frac{T_1}{T_4} \right)^{\left(\frac{1}{\gamma-1} \right)} \quad (3.5)$$

Since $T_1 = T_2$ and $T_4 = T_3$ we have,

$$\frac{V_4}{V_1} = \frac{V_3}{V_2}$$

or

$$\frac{V_4}{V_3} = \frac{V_1}{V_2} = r \quad (\text{say}) \quad (3.6)$$

then,

$$\eta_{Carnot} = \frac{mRT_3 \log_e r - mRT_1 \log_e r}{mRT_3 \log_e r} \quad (3.7)$$

$$= \frac{T_3 - T_1}{T_3} = 1 - \frac{T_1}{T_3} \quad (3.8)$$

The lower temperature i.e., sink temperature, T_1 , is normally the atmospheric temperature or the cooling water temperature and hence fixed. So the increase in thermal efficiency can be achieved only by increasing the source temperature. In other words, the upper temperature is required to be maintained as high as possible, to achieve maximum thermal efficiency. Between two fixed temperatures Carnot cycle (and other reversible cycles) has the maximum possible efficiency compared to other air-standard cycles. In spite of this advantage, Carnot cycle does not provide a suitable basis for the operation of an engine using a gaseous working fluid because the work output from this cycle will be quite low.

Mean effective pressure, p_m , is defined as that hypothetical constant pressure acting on the piston during its expansion stroke producing the same work output as that from the actual cycle. Mathematically,

$$p_m = \frac{\text{Work Output}}{\text{Swept Volume}} \quad (3.9)$$

It can be shown as

$$p_m = \frac{\text{Area of indicator diagram}}{\text{Length of diagram}} \times \text{constant} \quad (3.10)$$

The constant depends on the mechanism used to get the indicator diagram and has the units, bar/m. These formulae are quite often used to calculate the performance of an internal combustion engine. If the work output is the indicated output then it is called indicated mean effective pressure, p_{im} , and if the work output is the brake output then it is called brake mean effective pressure, p_{bm} .

3.3 THE STIRLING CYCLE

The Carnot cycle has a low mean effective pressure because of its very low work output. Hence, one of the modified forms of the cycle to produce higher mean effective pressure whilst theoretically achieving full Carnot cycle efficiency is the Stirling cycle. It consists of two isothermal and two constant volume processes. The heat rejection and addition take place at constant temperature. The p - V and T - s diagrams for the Stirling cycle are shown in Figs.3.3(a) and 3.3(b) respectively. It is clear from Fig.3.3(b) that the amount of heat addition and rejection during constant volume processes is same.

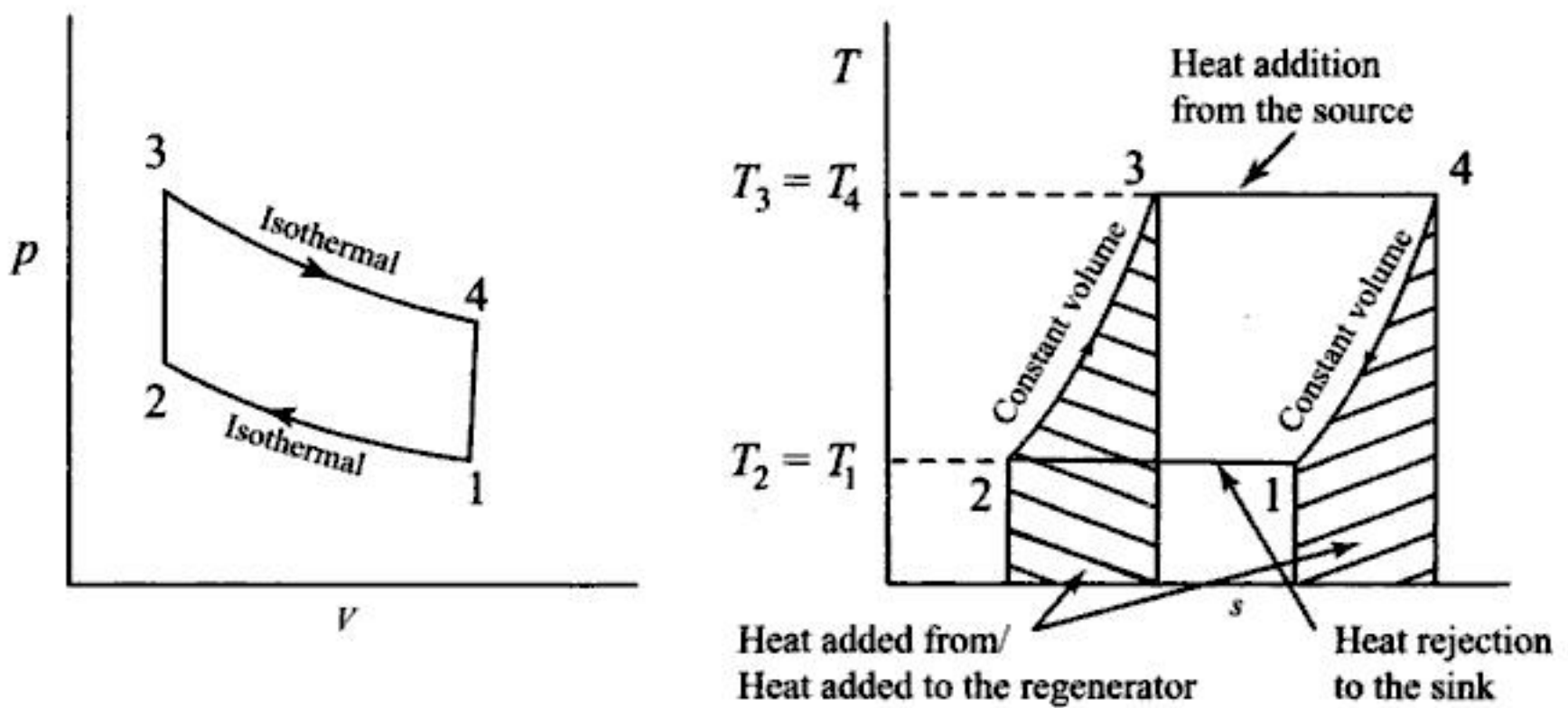


Fig. 3.3 Stirling Cycle

Hence, the efficiency of the cycle is given as

$$\eta_{\text{Stirling}} = \frac{RT_3 \log_e \left(\frac{V_4}{V_3} \right) - RT_1 \log_e \left(\frac{V_1}{V_2} \right)}{RT_3 \log_e \left(\frac{V_4}{V_3} \right)} \quad (3.11)$$

But $V_3 = V_2$ and $V_4 = V_1$

$$\eta_{\text{Stirling}} = \frac{T_3 - T_1}{T_3} \quad (3.12)$$

same as Carnot efficiency

The Stirling cycle was used earlier for hot air engines and became obsolete as Otto and Diesel cycles came into use. The design of Stirling engine involves a major difficulty in the design and construction of heat exchanger to operate continuously at very high temperatures. However, with the development in metallurgy and intensive research in this type of engine, the Stirling engine has staged a come back in practical appearance. In practice, the heat exchanger efficiency cannot be 100%. Hence the Stirling cycle efficiency will be less than Carnot efficiency and can be written as

$$\eta = \frac{R(T_3 - T_1) \log_e r}{RT_3 \log_e r + (1 - \epsilon)C_v(T_3 - T_1)} \quad (3.13)$$

where ϵ is the heat exchanger effectiveness.

3.4 THE ERICSSON CYCLE

The Ericsson cycle consists of two isothermal and two constant pressure processes. The heat addition and rejection take place at constant

pressure as well as isothermal processes. Since the process $2 \rightarrow 3$ and $3 \rightarrow 4$ are parallel to each other on the T - s diagram, the net effect is that the heat need be added only at constant temperature $T_3 = T_4$ and rejected at the constant temperature $T_1 = T_2$.

The cycle is shown on p - V and T - s diagrams in Fig.3.4(a) and 3.4(b) respectively. The advantage of the Ericsson cycle over the Carnot and Stirling cycles is its smaller pressure ratio for a given ratio of maximum to minimum specific volume with higher mean effective pressure.

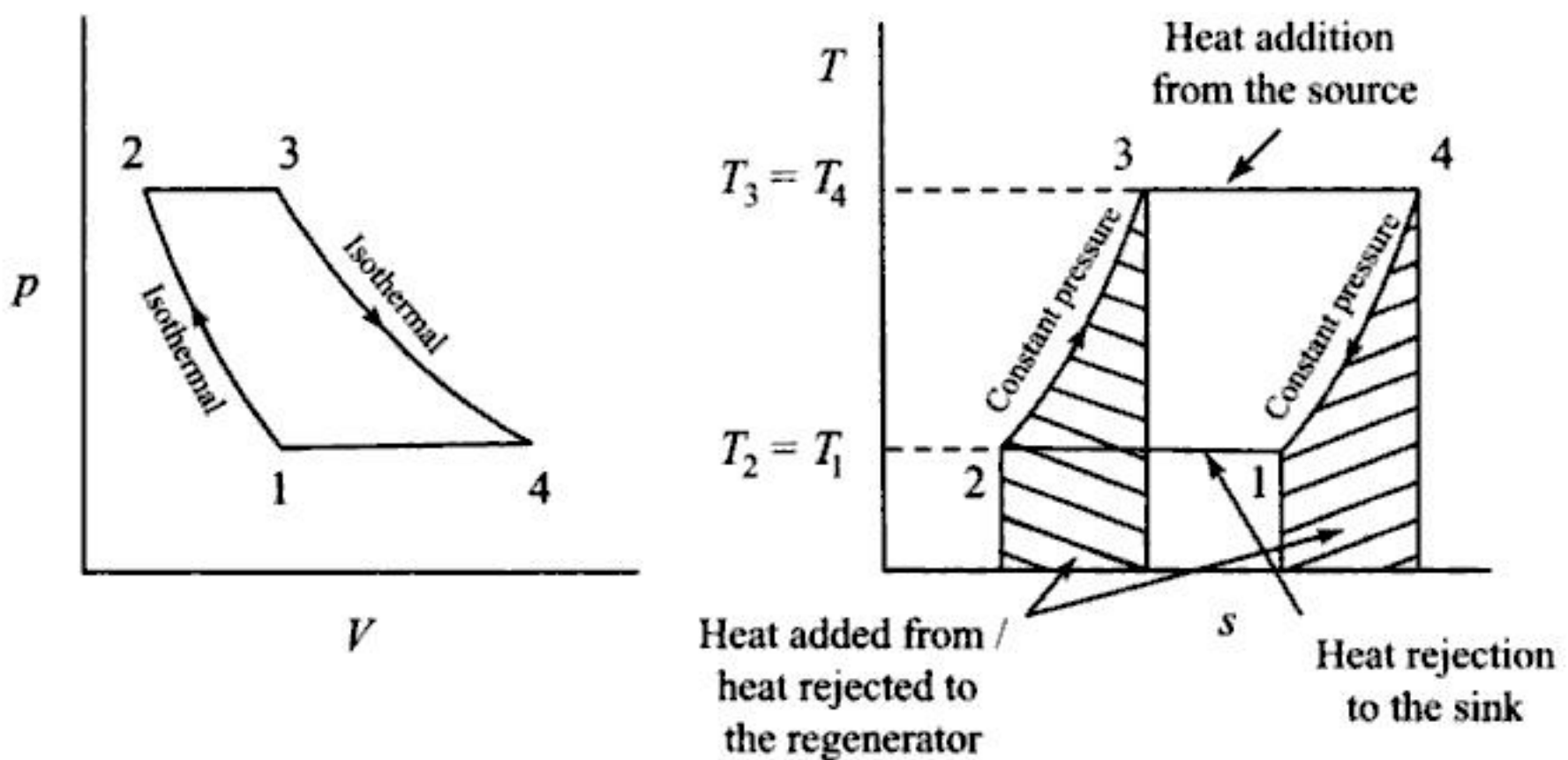


Fig. 3.4 Ericsson Cycle

The Ericsson cycle does not find practical application in piston engines but is approached by a gas turbine employing a large number of stages with heat exchangers, insulators and reheaters.

3.5 THE OTTO CYCLE

The main drawback of the Carnot cycle is its impracticability due to high pressure and high volume ratios employed with comparatively low mean effective pressure. Nicolaus Otto (1876), proposed a constant-volume heat addition cycle which forms the basis for the working of today's spark-ignition engines. The cycle is shown on p - V and T - s diagrams in Fig.3.5(a) and 3.5(b) respectively.

When the engine is working on full throttle, the processes $0 \rightarrow 1$ and $1 \rightarrow 0$ on the p - V diagram represents suction and exhaust processes and their effect is nullified. The process $1 \rightarrow 2$ represents isentropic compression of the air when the piston moves from

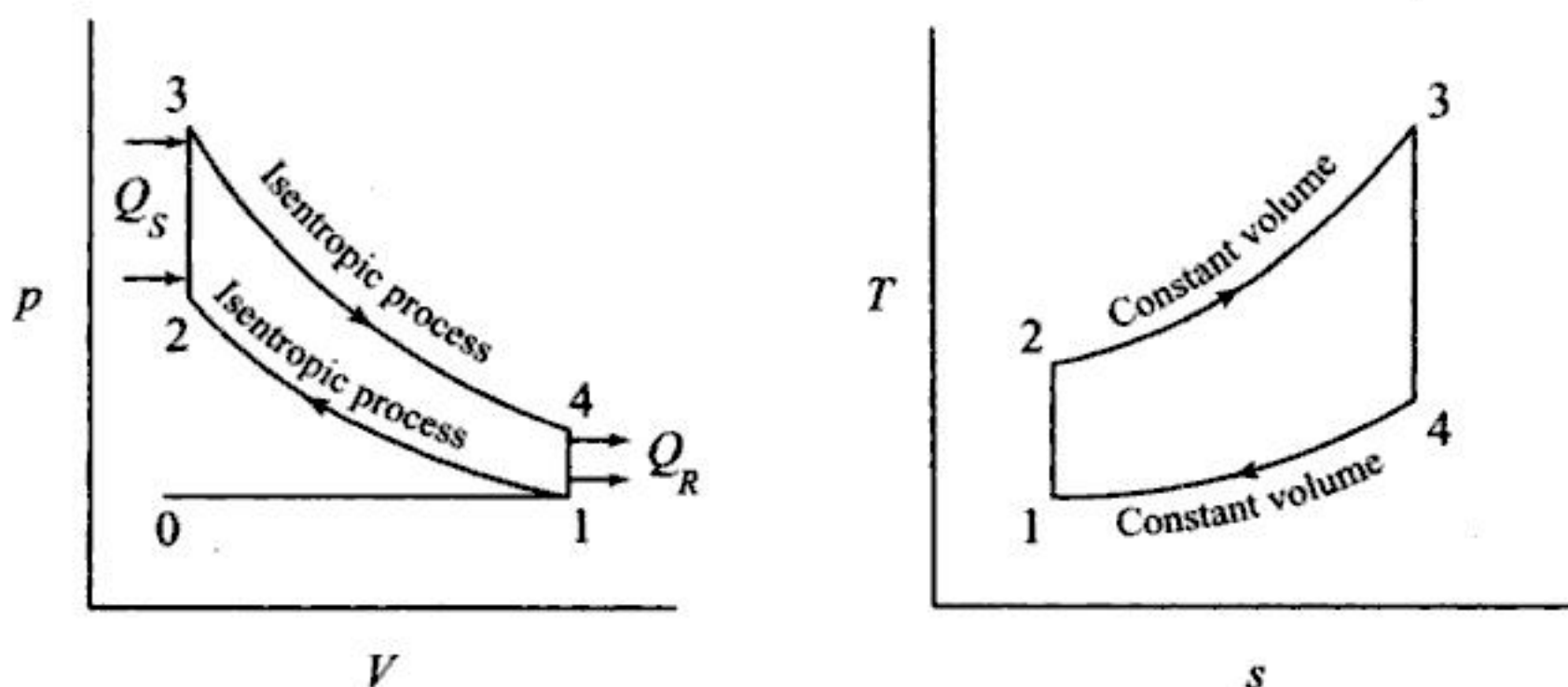


Fig. 3.5 Otto Cycle

bottom dead centre to top dead centre. During the process 2→3 heat is supplied reversibly at constant volume. This process corresponds to spark-ignition and combustion in the actual engine. The processes 3→4 and 4→1 represent isentropic expansion and constant volume heat rejection respectively.

3.5.1 Thermal Efficiency

The thermal efficiency of Otto cycle can be written as

$$\eta_{\text{Otto}} = \frac{Q_S - Q_R}{Q_S} \quad (3.14)$$

Considering constant volume processes 2→3 and 4→1, the heat supplied and rejected of air can be written as

$$Q_S = mC_v(T_3 - T_2) \quad (3.15)$$

$$Q_R = mC_v(T_4 - T_1) \quad (3.16)$$

$$\begin{aligned} \eta_{\text{Otto}} &= \frac{m(T_3 - T_2) - m(T_4 - T_1)}{m(T_3 - T_2)} \\ &= 1 - \frac{T_4 - T_1}{T_3 - T_2} \end{aligned} \quad (3.17)$$

Considering isentropic processes 1→2 and 3→4, we have

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2} \right)^{(\gamma-1)} \quad (3.18)$$

and

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{(\gamma-1)} \quad (3.19)$$

But the volume ratios V_1/V_2 and V_4/V_3 are equal to the compression ratio, r . Therefore,

$$\frac{V_1}{V_2} = \frac{V_4}{V_3} = r \quad (3.20)$$

therefore,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad (3.21)$$

From Eq.3.21, it can be easily shown that

$$\frac{T_4}{T_3} = \frac{T_1}{T_2} = \frac{T_4 - T_1}{T_3 - T_2} \quad (3.22)$$

$$\eta_{Otto} = 1 - \frac{T_1}{T_2} \quad (3.23)$$

$$= 1 - \frac{1}{\left(\frac{V_1}{V_2}\right)^{(\gamma-1)}} \quad (3.24)$$

$$= 1 - \frac{1}{r^{(\gamma-1)}} \quad (3.25)$$

Note that the thermal efficiency of Otto cycle is a function of compression ratio r and the ratio of specific heats, γ . As γ is assumed to be a constant for any working fluid, the efficiency is increased by increasing the compression ratio. Further, the efficiency is independent of heat supplied and pressure ratio. The use of gases with higher γ values would increase efficiency of Otto cycle. Fig.3.6 shows the effect of γ and r on the efficiency.

3.5.2 Work Output

The net work output for an Otto cycle can be expressed as

$$W = \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1} \quad (3.26)$$

Also

$$\begin{aligned} \frac{p_2}{p_1} &= \frac{p_3}{p_4} = r^\gamma \\ \frac{p_3}{p_2} &= \frac{p_4}{p_1} = r_p \quad (\text{say}) \end{aligned} \quad (3.27)$$

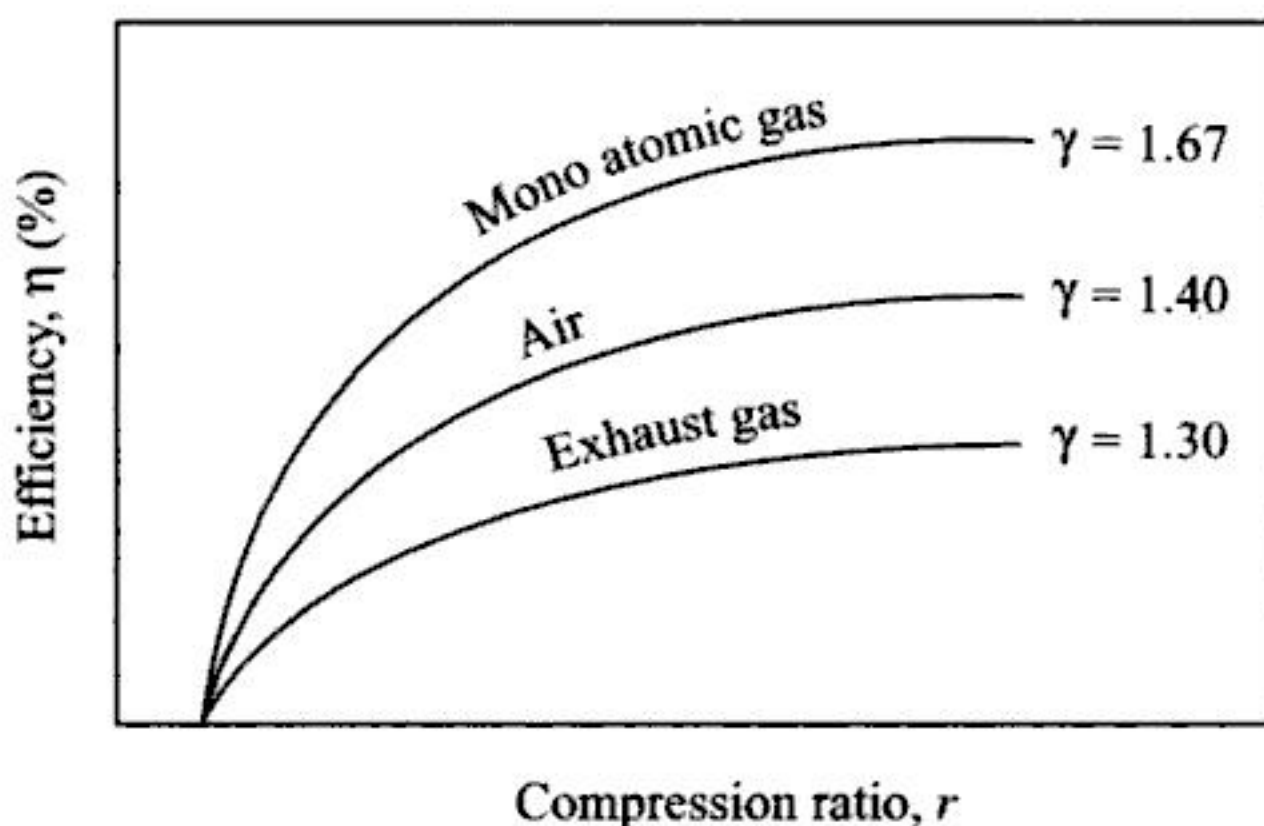


Fig. 3.6 Effect of r and γ on Efficiency for Otto Cycle

$$V_1 = rV_2 \quad \text{and} \quad V_4 = rV_3$$

therefore,

$$W = \frac{p_1 V_1}{\gamma - 1} \left(\frac{p_3 V_3}{p_1 V_1} - \frac{p_4 V_4}{p_1 V_1} - \frac{p_2 V_2}{p_1 V_1} + 1 \right) \quad (3.28)$$

$$= \frac{p_1 V_1}{\gamma - 1} \left(\frac{r_p r^\gamma}{r} - r_p - \frac{r^\gamma}{r} + 1 \right)$$

$$= \frac{p_1 V_1}{\gamma - 1} (r_p r^{\gamma-1} - r_p - r^{\gamma-1} + 1)$$

$$= \frac{p_1 V_1}{\gamma - 1} (r_p - 1)(r^{\gamma-1} - 1) \quad (3.29)$$

3.5.3 Mean Effective Pressure

The mean effective pressure of the cycle is given by

$$p_m = \frac{\text{Work output}}{\text{Swept volume}} \quad (3.30)$$

$$\text{Swept volume} = V_1 - V_2 = V_2(r - 1)$$

$$\begin{aligned} p_m &= \frac{\frac{1}{\gamma-1} p_1 V_1 (r_p - 1) (r^{\gamma-1} - 1)}{V_2(r - 1)} \\ &= \frac{p_1 r (r_p - 1) (r^{\gamma-1} - 1)}{(\gamma - 1)(r - 1)} \end{aligned} \quad (3.31)$$

Thus, it can be seen that the work output is directly proportional to pressure ratio, r_p . The mean effective pressure which is an indication of the internal work output increases with a pressure ratio at a fixed value of compression ratio and ratio of specific heats. For an Otto cycle, an increase in the compression ratio leads to an increase in the mean effective pressure as well as the thermal efficiency.

3.6 THE DIESEL CYCLE

In actual spark-ignition engines, the upper limit of compression ratio is limited by the self-ignition temperature of the fuel. This limitation on the compression ratio can be circumvented if air and fuel are compressed separately and brought together at the time of combustion. In such an arrangement fuel can be injected into the cylinder which contains compressed air at a higher temperature than the self-ignition temperature of the fuel. Hence the fuel ignites on its own accord and requires no special device like an ignition system in a spark-ignition engine. Such engines work on heavy liquid fuels. These engines are called compression-ignition engines and they work on a ideal cycle known as Diesel cycle. The difference between Otto and Diesel cycles is in the process of heat addition. In Otto cycle the heat addition takes place at constant volume whereas in the Diesel cycle it is at constant pressure. For this reason, the Diesel cycle is often referred to as the constant-pressure cycle. It is better to avoid this term as it creates confusion with Joules cycle. The Diesel cycle is shown on p - V and T - s diagrams in Fig.3.7(a) and 3.7(b) respectively.

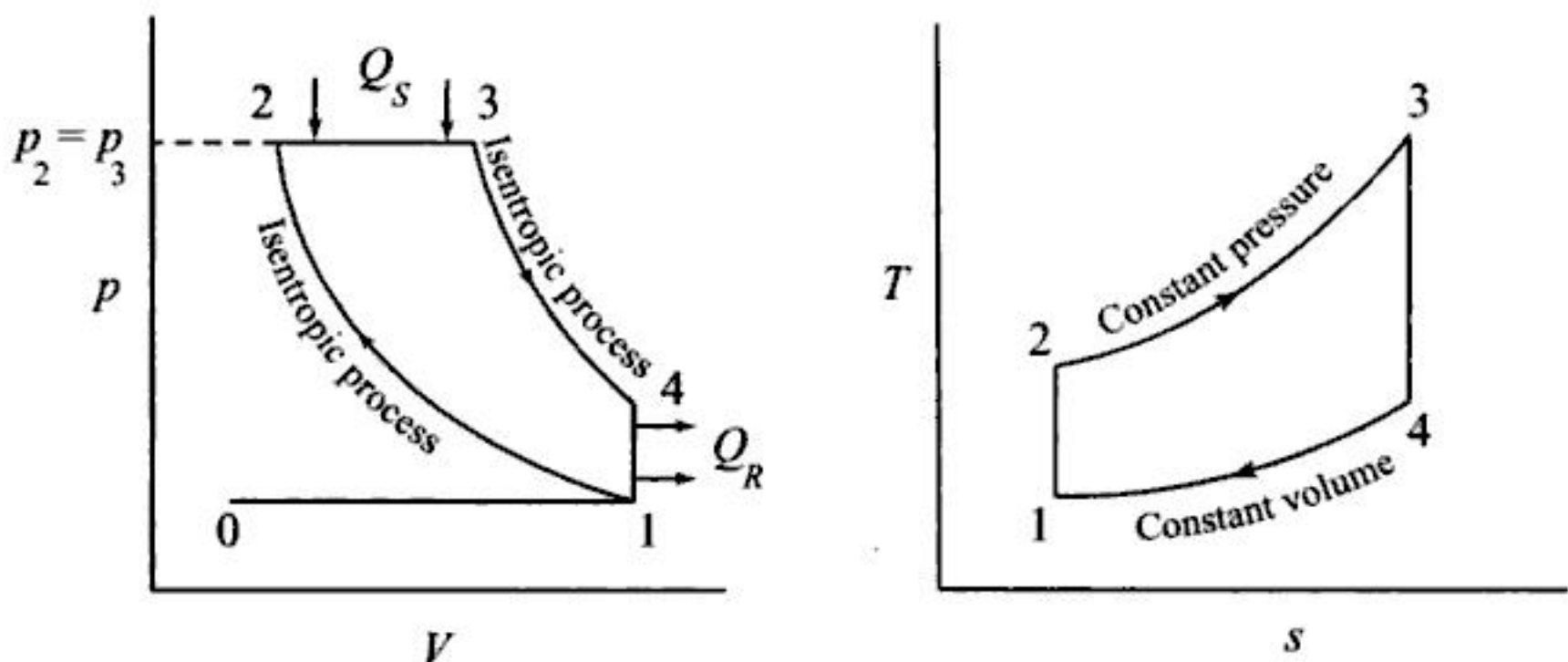


Fig. 3.7 Diesel Cycle

To analyze the diesel cycle the suction and exhaust strokes, represented by $0 \rightarrow 1$ and $1 \rightarrow 0$, are neglected as in the case of the Otto cycle. Here, the volume ratio $\frac{V_1}{V_2}$ is the compression ratio, r . The volume ratio $\frac{V_3}{V_2}$ is called the cut-off ratio, r_c .

3.6.1 Thermal Efficiency

The thermal efficiency of the Diesel cycle is given by

$$\begin{aligned}\eta_{Diesel} &= \frac{Q_S - Q_R}{Q_S} \\ &= \frac{mC_p(T_3 - T_2) - mC_v(T_4 - T_1)}{mC_p(T_3 - T_2)}\end{aligned}\quad (3.32)$$

$$\begin{aligned}&= 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)} \\ &= 1 - \frac{1}{\gamma} \left(\frac{T_4 - T_1}{T_3 - T_2} \right)\end{aligned}\quad (3.33)$$

Considering the process $1 \rightarrow 2$

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{(\gamma-1)} = T_1 r^{(\gamma-1)} \quad (3.34)$$

Considering the constant pressure process $2 \rightarrow 3$, we have

$$\begin{aligned}\frac{V_2}{T_2} &= \frac{V_3}{T_3} \\ \frac{T_3}{T_2} &= \frac{V_3}{V_2} = r_c \quad (\text{say}) \\ T_3 &= T_2 r_c\end{aligned}\quad (3.35)$$

From Eqs. 3.34 and 3.35, we have

$$T_3 = T_1 r^{(\gamma-1)} r_c \quad (3.36)$$

Considering process $3 \rightarrow 4$, we have

$$\begin{aligned}T_4 &= T_3 \left(\frac{V_3}{V_4} \right)^{(\gamma-1)} \\ &= T_3 \left(\frac{V_3}{V_2} \times \frac{V_2}{V_4} \right)^{(\gamma-1)}\end{aligned}\quad (3.37)$$

$$= T_3 \left(\frac{r_c}{r} \right)^{(\gamma-1)} \quad (3.38)$$

From Eqs. 3.36 and 3.37, we have

$$\begin{aligned} T_4 &= T_1 r^{(\gamma-1)} r_c \left(\frac{r_c}{r} \right)^{(\gamma-1)} = T_1 r_c^\gamma \\ \eta_{Diesel} &= 1 - \frac{1}{\gamma} \left[\frac{T_1 (r_c^\gamma - 1)}{T_1 (r^{(\gamma-1)} r_c - r^{(\gamma-1)})} \right] \\ &= 1 - \frac{1}{\gamma} \left[\frac{(r_c^\gamma - 1)}{r^{(\gamma-1)} r_c - r^{(\gamma-1)}} \right] \\ &= 1 - \frac{1}{r^{(\gamma-1)}} \left[\frac{r_c^\gamma - 1}{\gamma (r_c - 1)} \right] \end{aligned} \quad (3.39)$$

It may be noted that the efficiency of the Diesel cycle is different from that of the Otto cycle only in the bracketed factor. This factor is always greater than unity. Hence for a given compression ratio, the Otto cycle is more efficient. In diesel engines the fuel cut-off ratio, r_c , depends on output, being maximum for maximum output. Therefore, unlike the Otto cycle the air-standard efficiency of the Diesel cycle depends on output. The higher efficiency of the Otto cycle as compared to the Diesel cycle for the same compression ratio is of no practical importance. In practice the operating compression ratios of diesel engines are much higher compared to spark-ignition engines working on Otto cycle. The normal range of compression ratio for diesel engine is 16 to 20 whereas for spark-ignition engines it is 6 to 10. Due to the higher compression ratios used in diesel engines the efficiency of a diesel engine is more than that of the gasoline engine.

3.6.2 Work Output

The net work output for a Diesel cycle is given by

$$\begin{aligned} W &= p_2(V_3 - V_2) + \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1} \\ &= p_2 V_2 (r_c - 1) + \frac{p_3 r_c V_2 - p_4 r V_2}{\gamma - 1} - \frac{p_2 V_2 - p_1 r V_2}{\gamma - 1} \\ &= V_2 \left[\frac{p_2 (r_c - 1)(\gamma - 1) + p_3 r_c - p_4 r - (p_2 - p_1 r)}{\gamma - 1} \right] \end{aligned} \quad (3.40)$$

$$\begin{aligned}
&= V_2 \left[\frac{p_2(r_c - 1)(\gamma - 1) + p_3 \left(r_c - \frac{p_4}{p_3} r \right) - p_2 \left(1 - \frac{p_1}{p_2} r \right)}{\gamma - 1} \right] \\
&= p_2 V_2 \left[\frac{(r_c - 1)(\gamma - 1) + \left(r_c - r_c^\gamma r^{(1-\gamma)} \right) - \left(1 - r^{(1-\gamma)} \right)}{\gamma - 1} \right] \\
&= \frac{p_1 V_1 r^{(\gamma-1)} \left[\gamma(r_c - 1) - r^{(1-\gamma)}(r_c^\gamma - 1) \right]}{\gamma - 1} \quad (3.41)
\end{aligned}$$

3.6.3 Mean Effective Pressure

The expression for mean effective pressure can be shown to be

$$p_m = \frac{p_1 V_1 \left[r^{(\gamma-1)} \gamma (r_c - 1) - (r_c^\gamma - 1) \right]}{(\gamma - 1) V_1 \left(\frac{r-1}{r} \right)} \quad (3.42)$$

$$= \frac{p_1 [\gamma r^\gamma (r_c - 1) - r(r_c^\gamma - 1)]}{(\gamma - 1)(r - 1)} \quad (3.43)$$

3.7 THE DUAL CYCLE

In the Otto cycle, combustion is assumed at constant volume while in Diesel cycle combustion is at constant pressure. In practice they are far from real. Since, some time interval is required for the chemical reactions during combustion process, the combustion cannot take place at constant volume. Similarly, due to rapid uncontrolled combustion in diesel engines, combustion does not occur at constant pressure. The Dual cycle, also called a mixed cycle or limited pressure cycle, is a compromise between Otto and Diesel cycles. Figures 3.8(a) and 3.8(b) show the Dual cycle on p - V and T - s diagrams respectively.

In a Dual cycle a part of the heat is first supplied to the system at constant volume and then the remaining part at constant pressure.

3.7.1 Thermal Efficiency

The efficiency of the cycle may be written as

$$\eta_{Dual} = \frac{Q_S - Q_R}{Q_S} \quad (3.44)$$

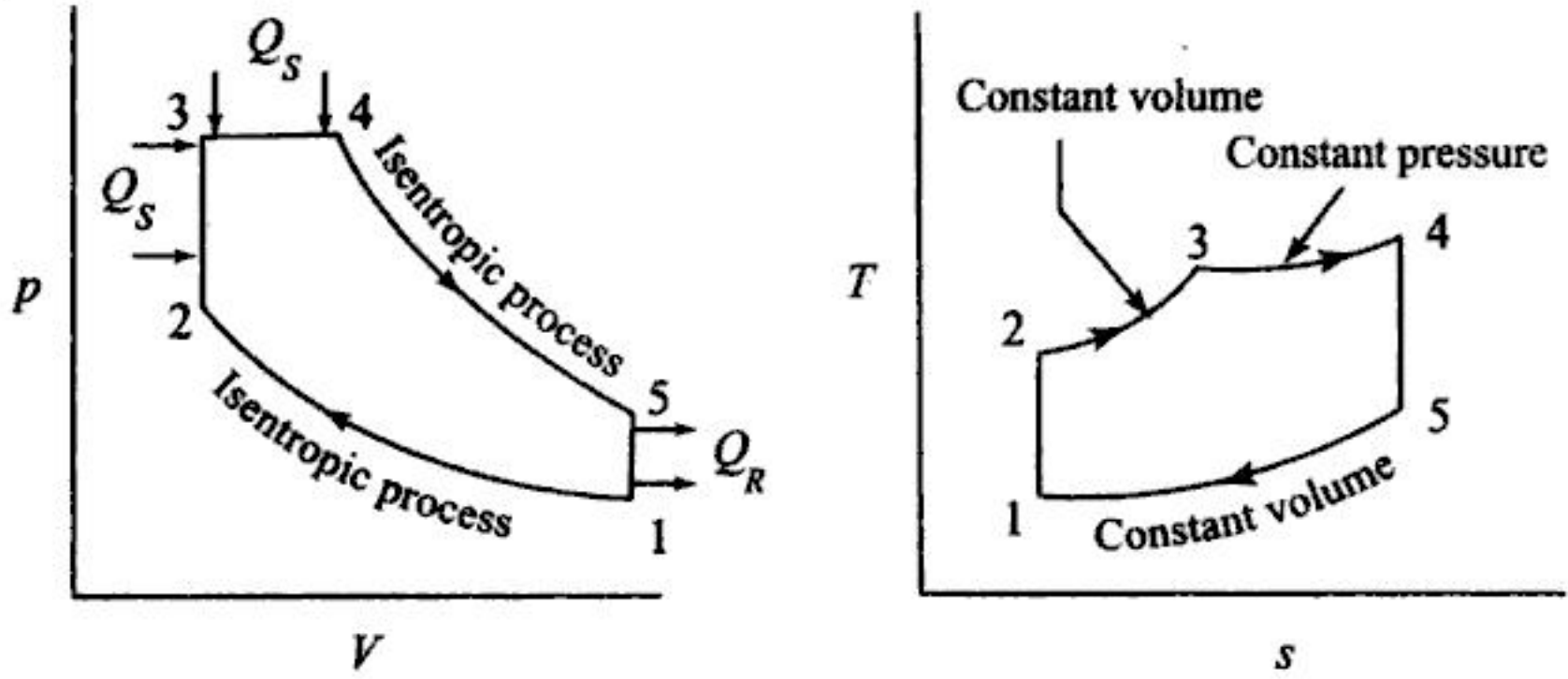


Fig. 3.8 Dual Cycle

$$\begin{aligned}
 &= \frac{mC_v(T_3 - T_2) + mC_p(T_4 - T_3) - mC_v(T_5 - T_1)}{mC_v(T_3 - T_2) + mC_p(T_4 - T_3)} \\
 &= 1 - \frac{T_5 - T_1}{(T_3 - T_2) + \gamma(T_4 - T_3)} \quad (3.45)
 \end{aligned}$$

Now,

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{(\gamma-1)} = T_1 r^{(\gamma-1)} \quad (3.46)$$

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right) = T_1 r_p r^{(\gamma-1)} \quad (3.47)$$

where r_p is the pressure ratio in the constant volume heat addition process and is equal to $\frac{p_3}{p_2}$.

Cut-off ratio r_c is given by $\left(\frac{V_4}{V_3} \right)$

$$T_4 = T_3 \frac{V_4}{V_3} = T_3 r_c$$

Substituting for T_3 from Eq.3.47

$$T_4 = T_1 r_c r_p r^{(\gamma-1)} \quad (3.48)$$

and

$$T_5 = T_4 \left(\frac{V_4}{V_5} \right)^{(\gamma-1)} = T_1 r_p r_c r^{(\gamma-1)} \left(\frac{V_4}{V_5} \right)^{(\gamma-1)} \quad (3.49)$$

Now

$$\frac{V_4}{V_5} = \frac{V_4}{V_1} = \frac{V_4}{V_3} \times \frac{V_3}{V_1} \quad (3.50)$$

$$= \frac{V_4}{V_3} \times \frac{V_2}{V_1} \quad (\text{since } V_2 = V_3)$$

Therefore,

$$\frac{V_4}{V_5} = \frac{r_c}{r} \quad (3.51)$$

where $\frac{V_4}{V_5}$ is the expansion ratio. Now,

$$\begin{aligned} T_5 &= T_1 r_p r_c r^{\gamma-1} \left(\frac{r_c}{r} \right)^{\gamma-1} \\ &= T_1 r_p r_c^\gamma \end{aligned} \quad (3.52)$$

Substituting for T_2, T_3, T_4 and T_5 into Eq.3.45 and simplifying

$$\eta = 1 - \frac{1}{r^{(\gamma-1)}} \left[\frac{r_p r_c^\gamma - 1}{(r_p - 1) + r_p \gamma (r_c - 1)} \right] \quad (3.53)$$

It can be seen from the above equation that a value of $r_p > 1$ results in an increased efficiency for a given value of r_c and γ . Thus the efficiency of Dual cycle lies between that of the Otto cycle and the Diesel cycle having same compression ratio.

With $r_c = 1$, it becomes an Otto cycle, and with $r_p = 1$, it becomes a Diesel cycle.

3.7.2 Work Output

The net work output of the cycle is given by

$$\begin{aligned} W &= p_3(V_4 - V_3) + \frac{p_4 V_4 - p_5 V_5}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1} \\ &= \frac{p_1 V_1}{\gamma - 1} \left[(\gamma - 1) \left(\frac{p_4 V_4}{p_1 V_1} - \frac{p_3 V_3}{p_1 V_1} \right) + \frac{p_4 V_4}{p_1 V_1} - \frac{p_5 V_5}{p_1 V_1} - \frac{p_2 V_2}{p_1 V_1} + 1 \right] \\ &= \frac{p_1 V_1}{\gamma - 1} \left[(\gamma - 1) (r_c r_p r^{\gamma-1} - r_p r^{\gamma-1}) + r_c r_p r^{\gamma-1} \right. \\ &\quad \left. - r_p r_c^\gamma - r^{\gamma-1} + 1 \right] \\ &= \frac{p_1 V_1}{\gamma - 1} \left[\gamma r_c r_p r^{\gamma-1} - \gamma r_p r^{\gamma-1} + r_p r^{\gamma-1} - r_p r_c^\gamma - r^{\gamma-1} + 1 \right] \\ &= \frac{p_1 V_1}{\gamma - 1} \left[\gamma r_p r^{\gamma-1} (r_c - 1) + r^{\gamma-1} (r_p - 1) - (r_p r_c^\gamma - 1) \right] \end{aligned} \quad (3.54)$$

3.7.3 Mean Effective Pressure

The mean effective pressure is given by

$$\begin{aligned}
 p_m &= \frac{\text{Work output}}{\text{Swept volume}} = \frac{W}{V_s} \\
 &= \frac{1}{V_1 - V_2} \frac{p_1 V_1}{\gamma - 1} \left[\gamma r_p r^{\gamma-1} (r_c - 1) + r^{\gamma-1} (r_p - 1) - (r_p r_c^\gamma - 1) \right] \\
 &= \frac{1}{\left(1 - \frac{V_2}{V_1}\right)} \frac{p_1}{(\gamma - 1)} \left[\gamma r_p r^{\gamma-1} (r_c - 1) + r^{\gamma-1} (r_p - 1) \right. \\
 &\quad \left. - (r_p r_c^\gamma - 1) \right] \\
 &= p_1 \frac{[\gamma r_p r^\gamma (r_c - 1) + r^\gamma (r_p - 1) - r(r_p r_c^\gamma - 1)]}{(\gamma - 1)(r - 1)} \quad (3.55)
 \end{aligned}$$

3.8 COMPARISON OF THE OTTO, DIESEL AND DUAL CYCLES

The important variable factors which are used as the basis for comparison of the cycles are compression ratio, peak pressure, heat addition, heat rejection and the net work. In order to compare the performance of the Otto, Diesel and Dual combustion cycles some of the variable factors must be fixed. In this section, a comparison of these three cycles is made for the same compression ratio, same heat addition, constant maximum pressure and temperature, same heat rejection and net work output. This analysis will show which cycle is more efficient for a given set of operating conditions.

3.8.1 Same Compression Ratio and Heat Addition

The Otto cycle $1 \rightarrow 2 \rightarrow 3 \rightarrow 4 \rightarrow 1$, the Diesel cycle $1 \rightarrow 2 \rightarrow 3' \rightarrow 4' \rightarrow 1$ and the Dual cycle $1 \rightarrow 2 \rightarrow 2'' \rightarrow 3'' \rightarrow 4'' \rightarrow 1$ are shown in p - V and T - s diagrams in Fig.3.9(a) and 3.9(b) respectively for the same compression ratio and heat input.

From the T - s diagram, it can be seen that Area $5236 = \text{Area } 523'6' = \text{Area } 522''3''6''$ as this area represents the heat input which is the same for all cycles.

All the cycles start from the same initial state point 1 and the air is compressed from state 1 to 2 as the compression ratio is same. It is seen from the T - s diagram for the same heat input, the heat rejection in Otto cycle (area 5146) is minimum and heat rejection

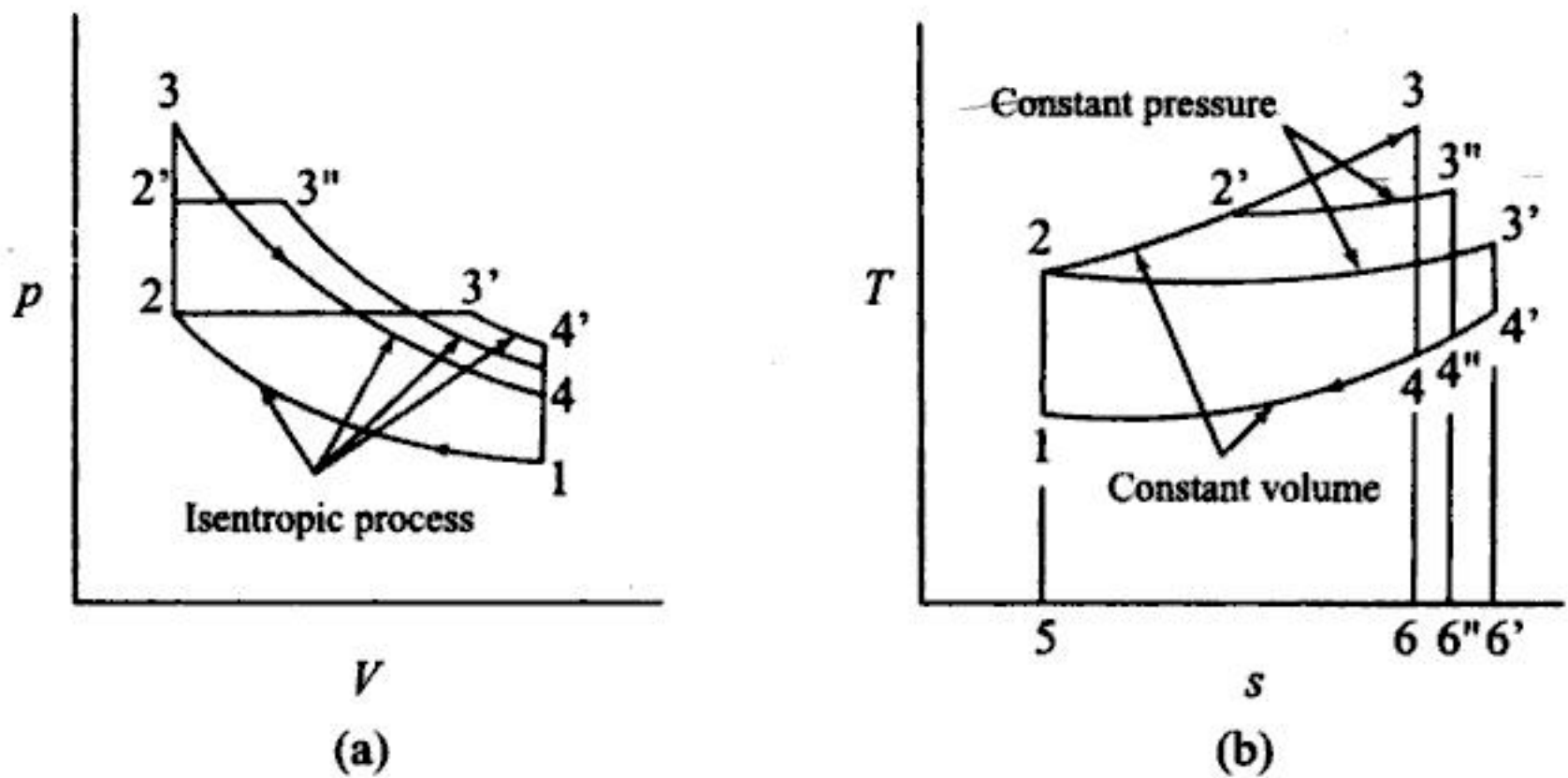


Fig. 3.9 Same Compression Ratio and Heat Addition

in Diesel cycle (514'6') is maximum. Consequently Otto cycle has the highest work output and efficiency. Diesel cycle has the least efficiency and Dual cycle having the efficiency between the two.

One more observation can be made i.e., Otto cycle allows the working medium to expand more whereas Diesel cycle is least in this respect. The reason is heat is added before expansion in the case of former (Otto cycle) and the last portion of heat supplied to the fluid has a relatively short expansion in case of the latter (Diesel cycle).

3.8.2 Same Compression Ratio and Heat Rejection

Efficiency of Otto cycle is given by [Figs.3.10(a) and 3.10(b)]

$$\eta_{\text{Otto}} = 1 - \frac{Q_R}{Q_S}$$

where Q_S is the heat supplied in the Otto cycle and is equal to the area under the curve 2→3 on the T - s diagram [Fig.3.10(b)]. The efficiency of the Diesel cycle is given by

$$\eta_{\text{Diesel}} = 1 - \frac{Q_R}{Q'_S}$$

where Q'_S is heat supplied in the Diesel cycle and is equal to the area under the curve 2→3' on the T - s diagram [Fig.3.10(b)].

From the T - s diagram in Fig.3.10 it is clear that $Q_S > Q'_S$ i.e., heat supplied in the Otto cycle is more than that of the Diesel cycle. Hence, it is evident that, the efficiency of the Otto cycle is greater

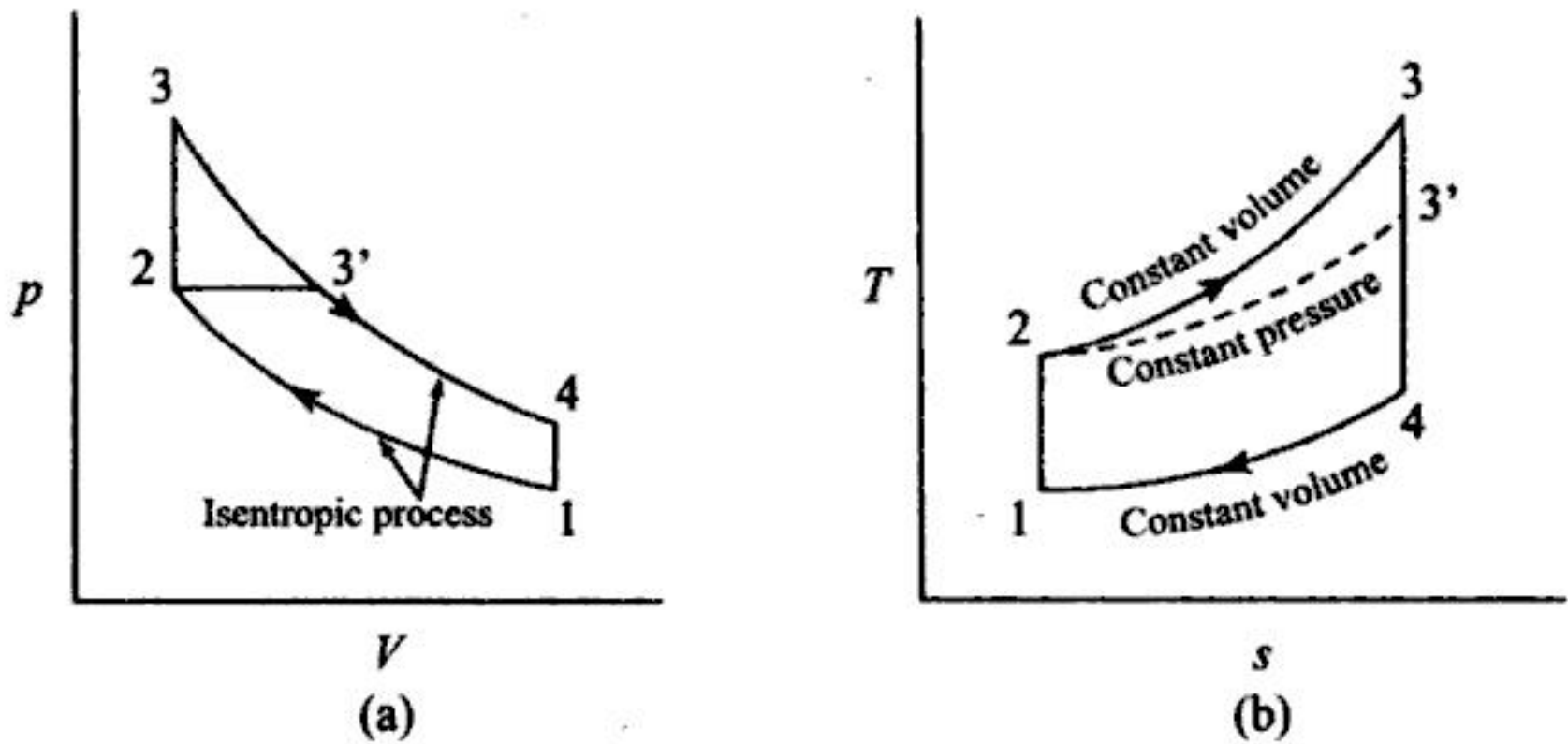


Fig. 3.10 Same Compression Ratio and Heat Rejection

than the efficiency of the Diesel cycle for a given compression ratio and heat rejection.

3.8.3 Same Peak Pressure, Peak Temperature and Heat Rejection

Figures 3.11(a) and 3.11(b) show the Otto cycle 1→2→3→4 and Diesel cycle 1→2'→3→4 on p - V and T - s coordinates, where the peak pressure and temperature and the amount of heat rejected are the same.

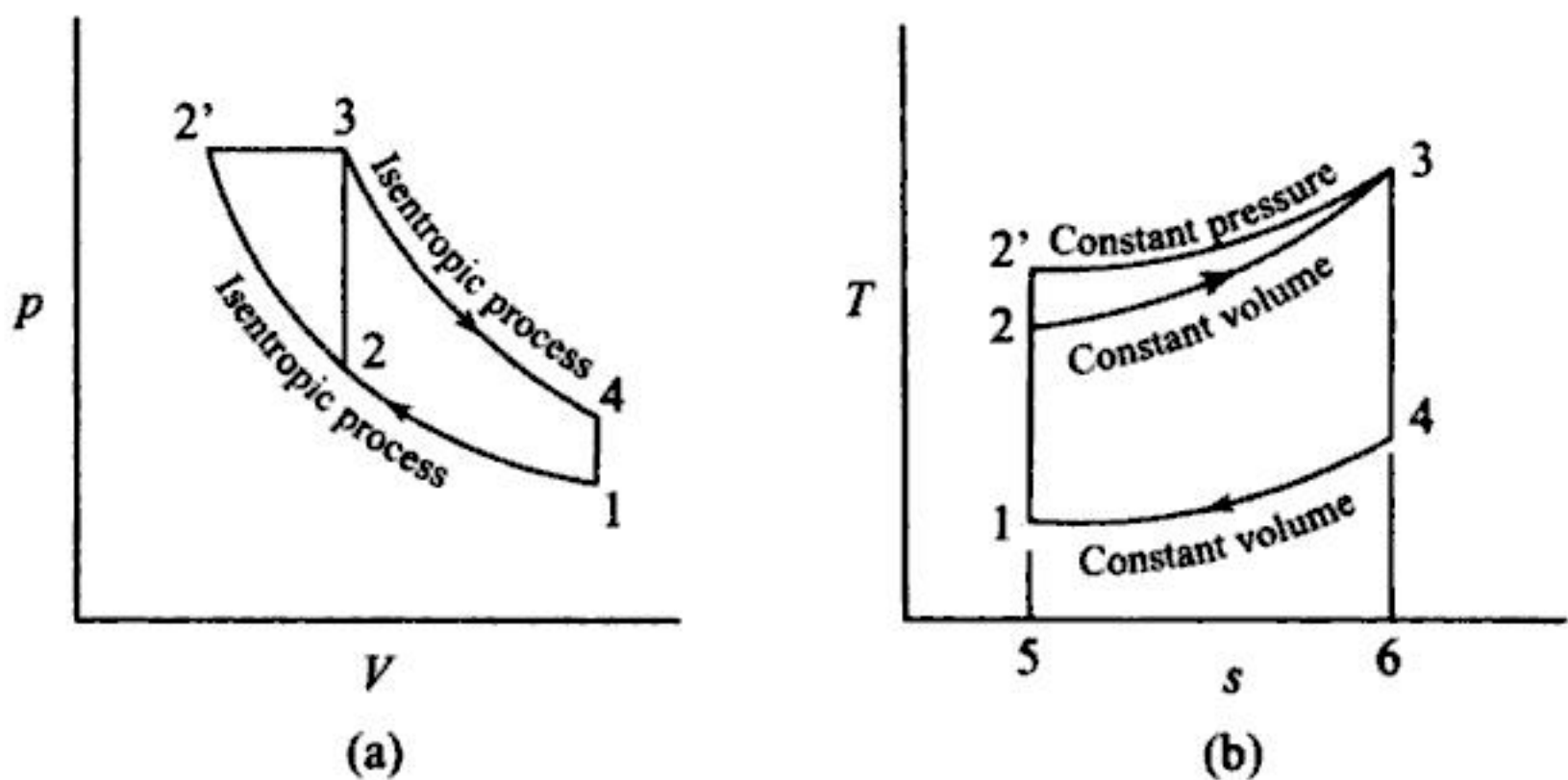


Fig. 3.11 Same Peak Pressure and Temperature

The efficiency of the Otto cycle $1 \rightarrow 2 \rightarrow 3 \rightarrow 4$ is given by

$$\eta_{\text{Otto}} = 1 - \frac{Q_R}{Q_S}$$

where Q_S is the area under the curve $2 \rightarrow 3$ in Fig.3.11(b).

The efficiency of the Diesel cycle, $1 \rightarrow 2 \rightarrow 3' \rightarrow 3 \rightarrow 4$ is

$$\eta_{\text{Diesel}} = 1 - \frac{Q_R}{Q'_S}$$

where Q'_S is the area under the curve $2' \rightarrow 3$ in Fig.3.11(b).

It is evident from Fig.3.11 that $Q'_S > Q_S$. Therefore, the Diesel cycle efficiency is greater than the Otto cycle efficiency when both engines are built to withstand the same thermal and mechanical stresses.

3.8.4 Same Maximum Pressure and Heat Input

For same maximum pressure and same heat input the Otto cycle (12341) and Diesel cycle (12'3'4'1) are shown on p - V and T - s diagrams in Figs.3.12(a) and 3.12(b) respectively.

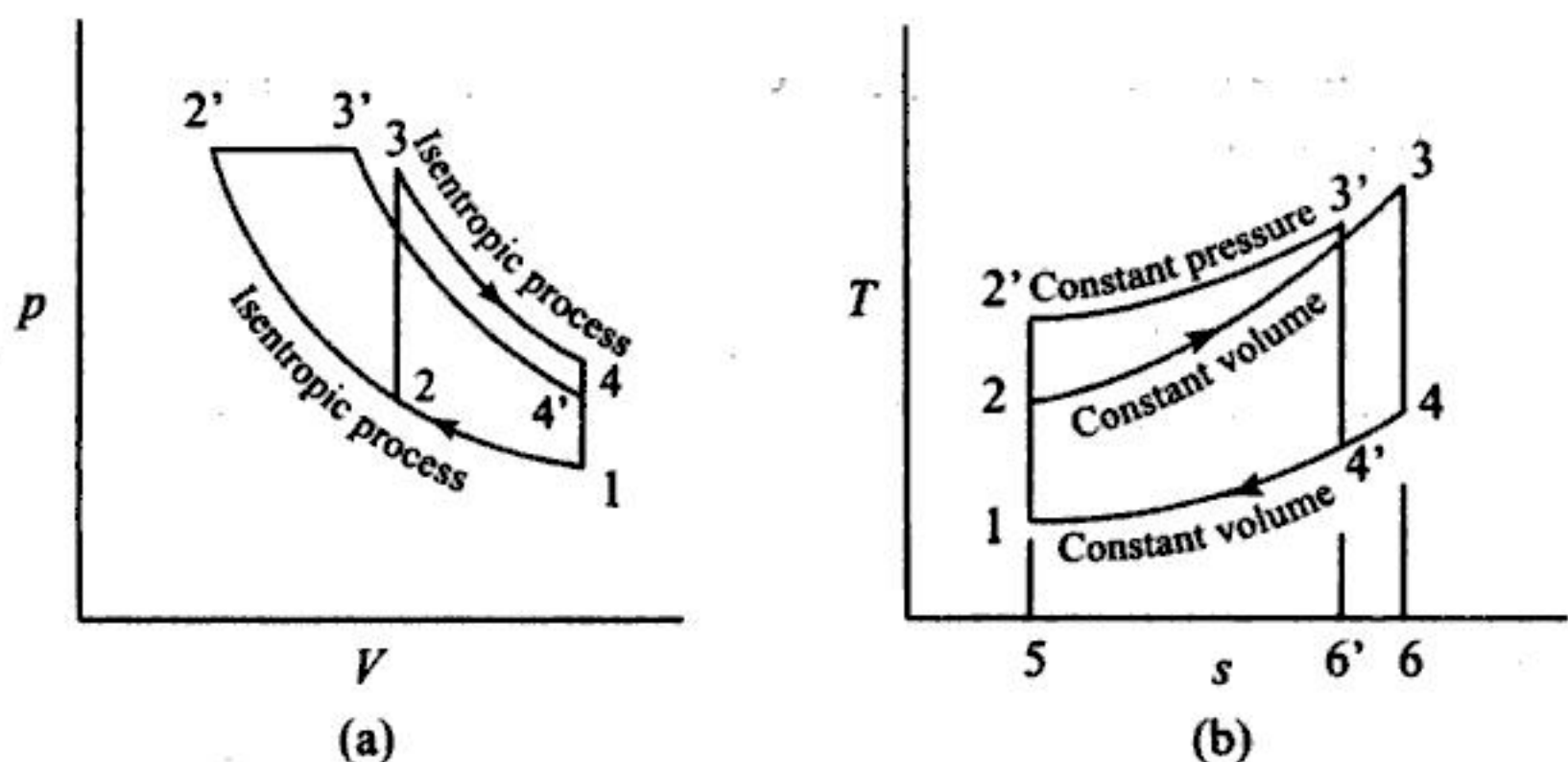


Fig. 3.12 Same Maximum Pressure and Heat Input

It is evident from the figure that the heat rejection for Otto cycle (area 1564 on T - s diagram) is more than the heat rejected in Diesel cycle (156'4'). Hence Diesel cycle is more efficient than Otto cycle for the condition of same maximum pressure and heat input. One can make a note that with these conditions the Diesel cycle has higher compression ratio $\frac{V_1}{V_2'}$ than that of Otto cycle $\frac{V_1}{V_2}$. One should also note that the cycle which is having higher efficiency allows maximum expansion. The Dual cycle efficiency will be between these two.

3.8.5 Same Maximum Pressure and Work Output

The efficiency, η , can be written as

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{\text{Work done}}{\text{Work done} + \text{Heat rejected}}$$

Refer to T - s diagram in Fig.3.12(b). For same work output the area 1234 (work output of Otto cycle) and area 12'3'4' (work output of Diesel cycle) are same. To achieve this, the entropy at 3 should be greater than entropy at 3'. It is clear that the heat rejection for Otto cycle is more than that of Diesel cycle. Hence, for these conditions the Diesel cycle is more efficient than the Otto cycle. The efficiency of Dual cycle lies between the two cycles.

3.9 THE LENOIR CYCLE

The Lenoir cycle consists of the following processes [see Fig.3.13(a)]. Constant volume heat addition (1→2); isentropic expansion (2→3); constant pressure heat rejection (3→1). The Lenoir cycle is used for pulse jet engines.

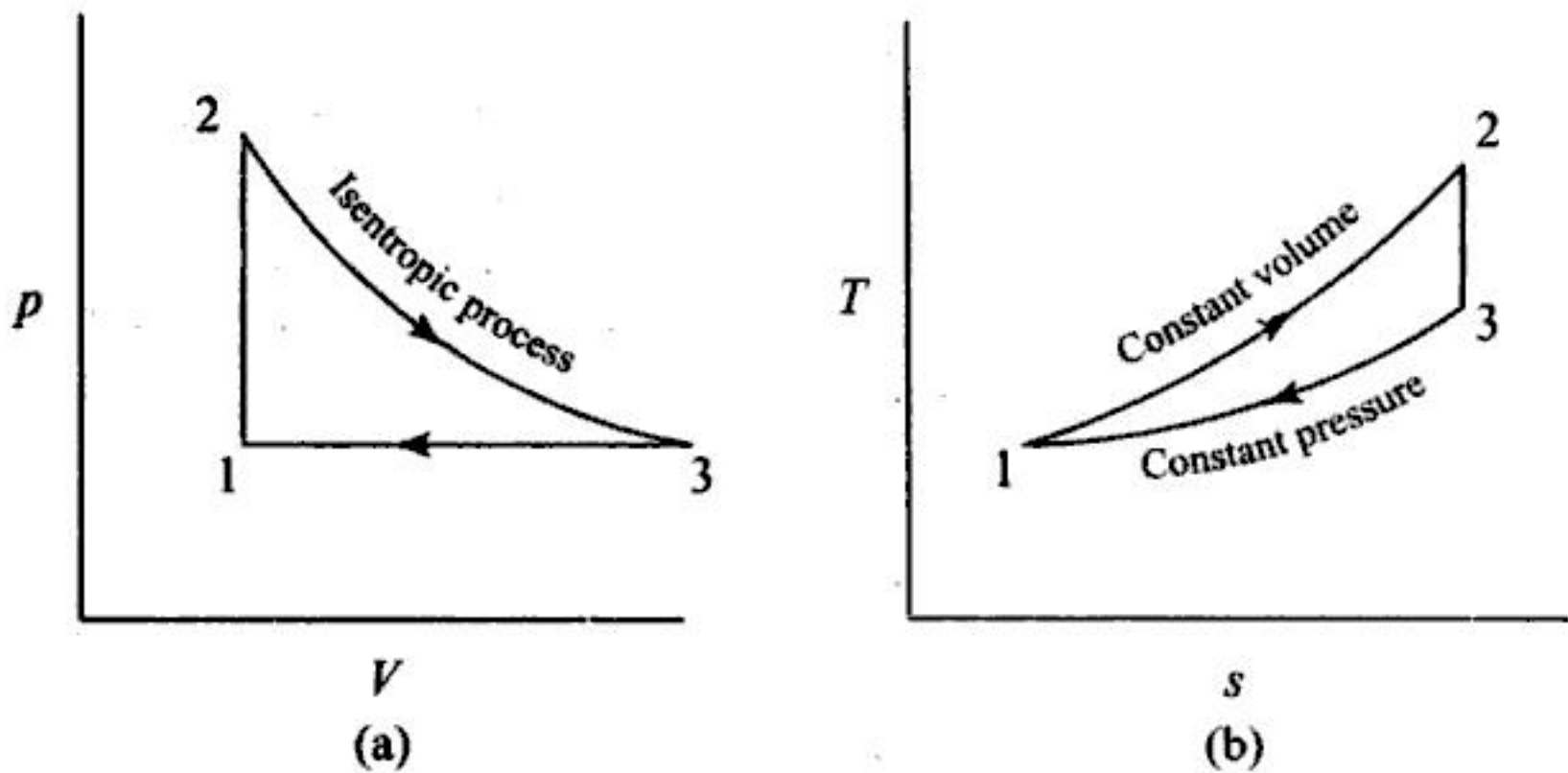


Fig. 3.13 Lenoir Cycle

$$\eta_{\text{Lenoir}} = \frac{Q_S - Q_F}{Q_S}$$

$$Q_S = mC_v(T_2 - T_1) \quad (3.56)$$

$$Q_R = mC_p(T_3 - T_1) \quad (3.57)$$

$$\begin{aligned}\eta_{\text{Lenoir}} &= \frac{mC_v(T_2 - T_1) - mC_p(T_3 - T_1)}{mC_v(T_2 - T_1)} \quad (3.58) \\ &= 1 - \gamma \left(\frac{T_3 - T_1}{T_2 - T_1} \right)\end{aligned}$$

Taking $p_2/p_1 = r_p$, we have $T_2 = T_1 r_p$ and

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} \quad (3.59)$$

$$\begin{aligned}T_3 &= T_2 \left(\frac{1}{r_p} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} = T_1 r_p \left(\frac{1}{r_p} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} = T_1 r_p^{(1/\gamma)} \\ \eta &= 1 - \gamma \left(\frac{T_1 r_p^{(1/\gamma)} - T_1}{T_1 r_p - T_1} \right) = 1 - \gamma \left(\frac{r_p^{(1/\gamma)} - 1}{r_p - 1} \right) \quad (3.60)\end{aligned}$$

Thus the efficiency of the Lenoir cycle depends upon the pressure ratio as well as the ratio of specific heats, viz., γ .

3.10 THE ATKINSON CYCLE

Atkinson cycle is an ideal cycle for Otto engine exhausting to a gas turbine. In this cycle the isentropic expansion (3→4) of an Otto cycle (1234) is further allowed to proceed to the lowest cycle pressure so as to increase the work output. With this modification the cycle is known as Atkinson cycle. The cycle is shown on p - V and T - s diagrams in Figs.3.14(a) and 3.14(b) respectively.

$$\eta_{\text{Atkinson}} = \frac{Q_S - Q_R}{Q_S} \quad (3.61)$$

$$= \frac{mC_v(T_3 - T_2) - mC_p(T_{4'} - T_1)}{mC_v(T_3 - T_2)} \quad (3.62)$$

$$= 1 - \gamma \left(\frac{T_{4'} - T_1}{T_3 - T_2} \right) \quad (3.63)$$

the compression ratio, $r = \frac{V_1}{V_2}$ and the expansion ratio $e = \frac{V_{4'}}{V_3}$. Now,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2} \right)^{(\gamma-1)} = r^{(\gamma-1)} \quad (3.64)$$

Therefore,

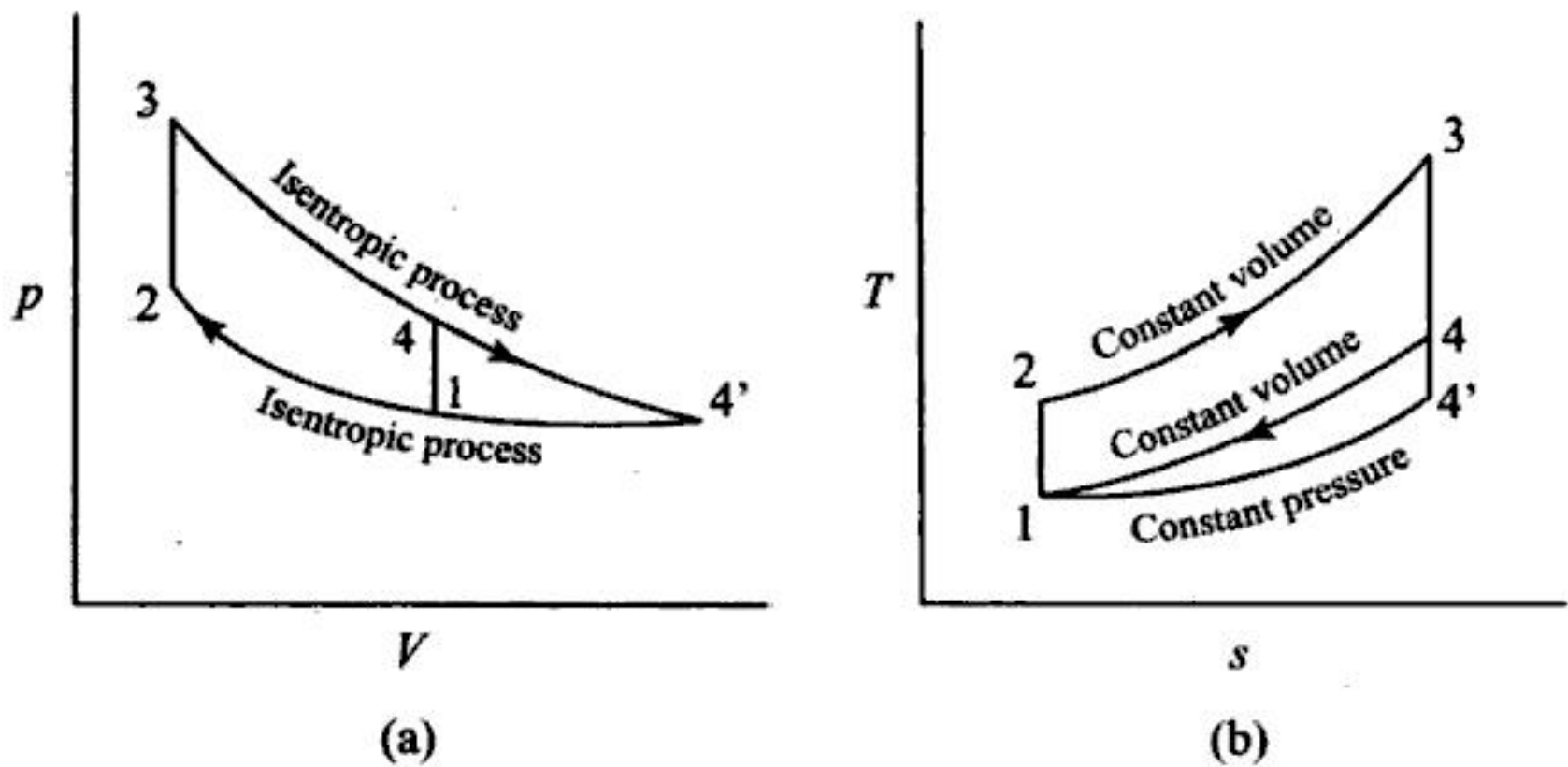


Fig. 3.14 Atkinson Cycle

$$T_2 = T_1 r^{(\gamma-1)} \quad (3.65)$$

$$\begin{aligned} \frac{T_3}{T_2} &= \frac{p_3}{p_2} = \left(\frac{p_3}{p_{4'}} \times \frac{p_{4'}}{p_2} \right) \\ &= \left(\frac{p_3}{p_{4'}} \times \frac{p_1}{p_2} \right) \end{aligned} \quad (3.66)$$

$$\frac{p_3}{p_{4'}} = \left(\frac{V_{4'}}{V_3} \right)^\gamma = e^\gamma \quad (3.67)$$

$$\frac{p_1}{p_2} = \left(\frac{V_2}{V_1} \right)^\gamma = \frac{1}{r^\gamma} \quad (3.68)$$

Substituting Eqs. 3.67 and 3.68 in Eq. 3.66,

$$\frac{T_3}{T_2} = \frac{e^\gamma}{r^\gamma} \quad (3.69)$$

$$T_3 = T_2 \frac{e^\gamma}{r^\gamma} = T_1 r^{(\gamma-1)} \frac{e^\gamma}{r^\gamma} = T_1 \frac{e^\gamma}{r} \quad (3.70)$$

$$\frac{T_{4'}}{T_3} = \left(\frac{V_3}{V_{4'}} \right)^{(\gamma-1)} = \frac{1}{e^{(\gamma-1)}}$$

$$T_{4'} = T_3 \frac{1}{e^{(\gamma-1)}} \quad (3.71)$$

$$= T_1 \left(\frac{e^\gamma}{r} \right) \left(\frac{1}{e^{(\gamma-1)}} \right)$$

$$T_{4'} = T_1 \frac{e}{r} \quad (3.72)$$

Substituting the values of T_2 , T_3 , $T_{4'}$ in the Eq.3.63,

$$\begin{aligned} \eta_{Atkinson} &= 1 - \gamma \left[\frac{T_1 e/r - T_1}{T_1 e^\gamma/r - T_1 r^{(\gamma-1)}} \right] \\ &= 1 - \gamma \left[\frac{e - r}{e^\gamma - r^\gamma} \right] \end{aligned} \quad (3.73)$$

3.11 THE BRAYTON CYCLE

The Brayton cycle is a theoretical cycle for gas turbines. This cycle consists of two reversible adiabatic or isentropic processes and two constant pressure processes.

Figure 3.15 shows the Brayton cycle on p - V and T - s coordinates. The cycle is similar to the Diesel cycle in compression and heat addition. The isentropic expansion of the Diesel cycle is further extended followed by constant pressure heat rejection.

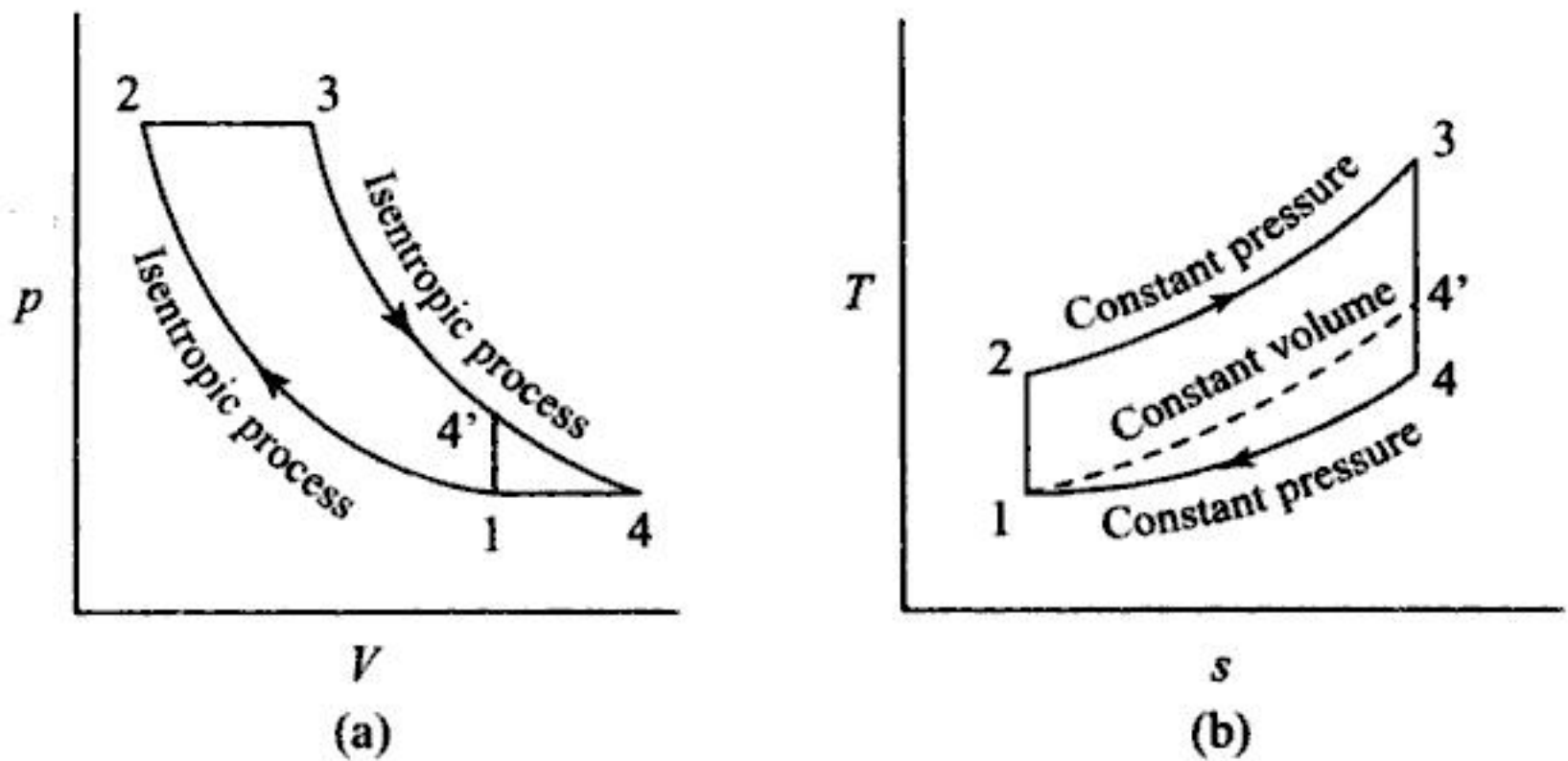


Fig. 3.15 Brayton Cycle

$$\begin{aligned} \eta_{Brayton} &= \frac{Q_S - Q_R}{Q_S} \\ &= \frac{mC_p(T_3 - T_2) - mC_p(T_4 - T_1)}{mC_p(T_3 - T_2)} \\ &= 1 - \frac{T_4 - T_1}{T_3 - T_2} \end{aligned} \quad (3.74)$$

If r is compression ratio i.e., (V_1/V_2) and r_p is the pressure ratio i.e., (p_2/p_1) then,

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \quad (3.75)$$

$$= \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = r^{(\gamma-1)} \quad (3.76)$$

$$T_4 = \frac{T_3}{r^{(\gamma-1)}} \quad (3.77)$$

$$T_1 = \frac{T_2}{r^{(\gamma-1)}} \quad (3.78)$$

$$\begin{aligned} \eta_{Brayton} &= 1 - \frac{(T_3/r^{(\gamma-1)}) - (T_2/r^{(\gamma-1)})}{T_3 - T_2} \\ &= 1 - \frac{1}{r^{(\gamma-1)}} \end{aligned} \quad (3.79)$$

$$r = \frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}} = r_p^{\frac{1}{\gamma}} \quad (3.80)$$

$$= 1 - \frac{1}{\left(r_p^{\frac{1}{\gamma}}\right)^{\gamma-1}} = 1 - \frac{1}{r_p^{\left(\frac{\gamma-1}{\gamma}\right)}} \quad (3.81)$$

From Eq.3.81, it is seen that the efficiency of the Brayton cycle depends only on the pressure ratio and the ratio of specific heat, γ .

$$\text{Network output} = \text{Expansion work} - \text{Compression work}$$

$$= C_p(T_3 - T_4) - C_p(T_2 - T_1)$$

$$= C_p T_1 \left(\frac{T_3}{T_1} - \frac{T_4}{T_1} - \frac{T_2}{T_1} + 1 \right)$$

$$\frac{W}{C_p T_1} = \frac{T_3}{T_1} - \frac{T_4}{T_1} - \frac{T_2}{T_1} + 1 \quad (3.82)$$

It can be easily seen from the Eq.3.82 (work output) that, the work output of the cycle depends on initial temperature, T_1 , the ratio of the maximum to minimum temperature, $\frac{T_3}{T_1}$, pressure ratio, r_p and γ which are used in the calculation of $\frac{T_2}{T_1}$. Therefore, for the same pressure ratio and initial conditions work output depends on the maximum temperature of the cycle.

Worked out Examples**OTTO CYCLE**

- 3.1 An engine working on Otto cycle has the following conditions : Pressure at the beginning of compression is 1 bar and pressure at the end of compression is 11 bar. Calculate the compression ratio and air-standard efficiency of the engine. Assume $\gamma = 1.4$.

Solution

$$r = \frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{1}{\gamma}\right)} = 11^{\frac{1}{1.4}}$$

$$= 5.54 \quad \underline{\underline{\text{Ans}}}$$

$$\text{Air-standard efficiency} = 1 - \frac{1}{r^{\gamma-1}}$$

$$= 1 - \left(\frac{1}{5.54}\right)^{0.4} = 0.496$$

$$= 49.6\% \quad \underline{\underline{\text{Ans}}}$$

- 3.2 In an engine working on ideal Otto cycle the temperatures at the beginning and end of compression are 50 °C and 373 °C. Find the compression ratio and the air-standard efficiency of the engine.

Solution

$$r = \frac{V_1}{V_2} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{\gamma-1}} = \left(\frac{646}{323}\right)^{\frac{1}{0.4}}$$

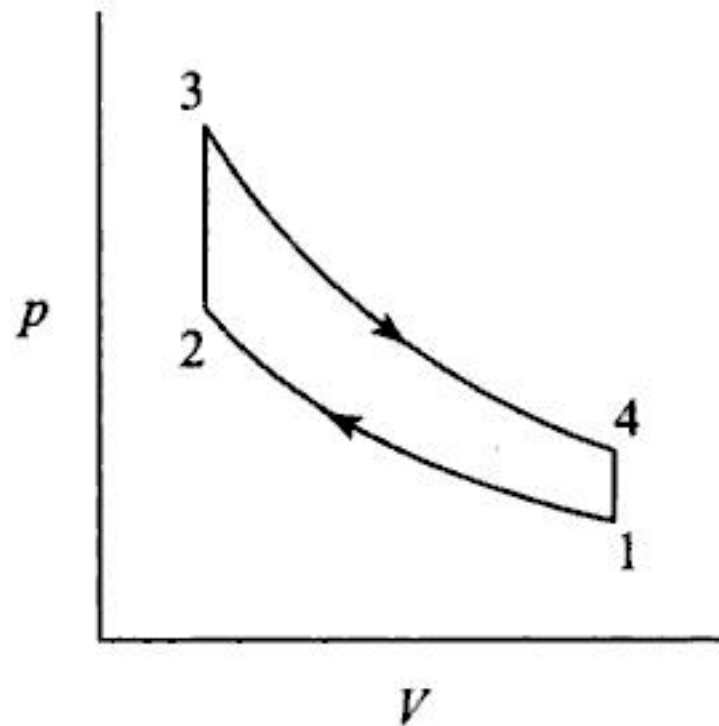
$$= 5.66 \quad \underline{\underline{\text{Ans}}}$$

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{T_1}{T_2}$$

$$= 1 - \frac{323}{646} = 0.5 = 50\% \quad \underline{\underline{\text{Ans}}}$$

3.3 In an Otto cycle air at 17 °C and 1 bar is compressed adiabatically until the pressure is 15 bar. Heat is added at constant volume until the pressure rises to 40 bar. Calculate the air-standard efficiency, the compression ratio and the mean effective pressure for the cycle. Assume $C_v = 0.717$ kJ/kg K and $R = 8.314$ kJ/kmol K.

Solution



Consider the process 1 – 2

$$p_1 V_1^\gamma = p_2 V_2^\gamma$$

$$\frac{V_1}{V_2} = r = \left(\frac{p_2}{p_1} \right)^{\frac{1}{\gamma}}$$

$$= \left(\frac{15}{1} \right)^{\frac{1}{1.4}} = 6.91$$

$$\eta = 1 - \left(\frac{1}{r} \right)^{\gamma-1} = 1 - \left(\frac{1}{6.91} \right)^{0.4}$$

$$= 0.539 = 53.9\% \quad \text{Ans}$$

$$T_2 = \frac{p_2 V_2}{p_1 V_1} T_1 = \frac{15}{1} \times \frac{1}{6.91} \times 290$$

$$= 629.5 \text{ K}$$

Consider the process 2 – 3

$$T_3 = \frac{p_3 T_2}{p_2} = \frac{40}{15} \times 629.5 = 1678.7$$

$$\text{Heat supplied} = C_v(T_3 - T_2)$$

$$= 0.717 \times (1678.7 - 629.5)$$

$$= 752.3 \text{ kJ/kg}$$

$$\text{Work done} = \eta \times q_s$$

$$= 0.539 \times 752.3 = 405.5 \text{ kJ/kg}$$

$$p_m = \frac{\text{Work done}}{\text{Swept volume}}$$

$$v_1 = \frac{V_1}{m} = M \frac{RT_1}{p_1}$$

$$= \frac{8314 \times 290}{29 \times 1 \times 10^5} = 0.8314 \text{ m}^3/\text{kg}$$

$$v_1 - v_2 = \frac{5.91}{6.91} \times 0.8314 = 0.711 \text{ m}^3/\text{kg}$$

$$p_m = \frac{405.5}{0.711} \times 10^3 = 5.70 \times 10^5 \text{ N/m}^2$$

$$= 5.70 \text{ bar}$$

Ans

3.4 Fuel supplied to an SI engine has a calorific value 42000 kJ/kg. The pressure in the cylinder at 30% and 70% of the compression stroke are 1.3 bar and 2.6 bar respectively. Assuming that the compression follows the law $pV^{1.3} = \text{constant}$. Find the compression ratio. If the relative efficiency of the engine compared with the air-standard efficiency is 50%. Calculate the fuel consumption in kg/kW h.

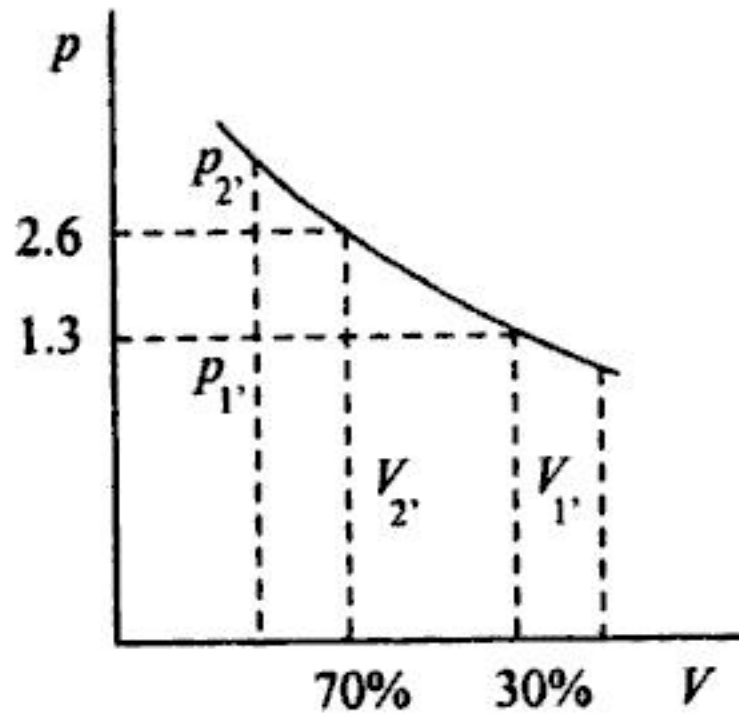
Solution

$$V_2 = 1$$

$$V_{1'} = 1 + 0.7(r - 1) = 0.7r + 0.3$$

$$V_{2'} = 1 + 0.3(r - 1) = 0.3r + 0.7$$

$$\frac{V_{1'}}{V_{2'}} = \left(\frac{p_2}{p_1} \right)^{\frac{1}{n}}$$



$$= \left(\frac{2.6}{1.3} \right)^{\frac{1}{1.3}} = 1.7$$

$$\frac{0.7r + 0.3}{0.3r + 0.7} = 1.7$$

$$r = 4.68 \quad \text{Ans}$$

$$\text{Air-standard efficiency} = 1 - \frac{1}{r^{\gamma-1}}$$

$$= 1 - \frac{1}{4.68^{0.4}} = 0.46$$

$$= 46\%$$

$$\text{Relative efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Air-standard efficiency}}$$

$$\eta_{ith} = 0.5 \times 0.46 = 0.23$$

$$\eta_{th} = \frac{ip}{CV \times \dot{m}}$$

where $\dot{m}$ is in kg/s

$$\frac{\dot{m}}{ip} = \frac{1}{42000 \times 0.23}$$

$$= 1.035 \times 10^{-4} \text{ kg/kW s}$$

$$= 1.035 \times 10^{-4} \times 3600 \text{ kg/kW h}$$

$$isfc = 0.373 \text{ kg/kW h} \quad \text{Ans}$$

- 3.5 A gas engine working on the Otto cycle has a cylinder of diameter 200 mm and stroke 250 mm. The clearance volume is 1570 cc. Find the air-standard efficiency. Assume $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K for air.

Solution

$$\begin{aligned}
 \text{Stroke volume, } V_s &= \frac{\pi}{4} d^2 L = \frac{\pi}{4} \times 20^2 \times 25 \\
 &= 7853.98 \text{ cc} \\
 \text{Compression ratio, } r &= 1 + \frac{V_s}{V_c} = 1 + \frac{7853.98}{1570} \\
 &= 6.00 \\
 \gamma &= \frac{C_p}{C_v} = \frac{1.004}{0.717} = 1.4 \\
 \text{Air-standard efficiency} &= 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{6^{(0.4)}} \\
 &= 0.512 = 51.2\% \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

- 3.6 In a S.I. engine working on the ideal Otto cycle, the compression ratio is 5.5. The pressure and temperature at the beginning of compression are 1 bar and 27 °C respectively. The peak pressure is 30 bar. Determine the pressure and temperatures at the salient points, the air-standard efficiency and the mean effective pressure. Assume ratio of specific heats to be 1.4 for air.

Solution

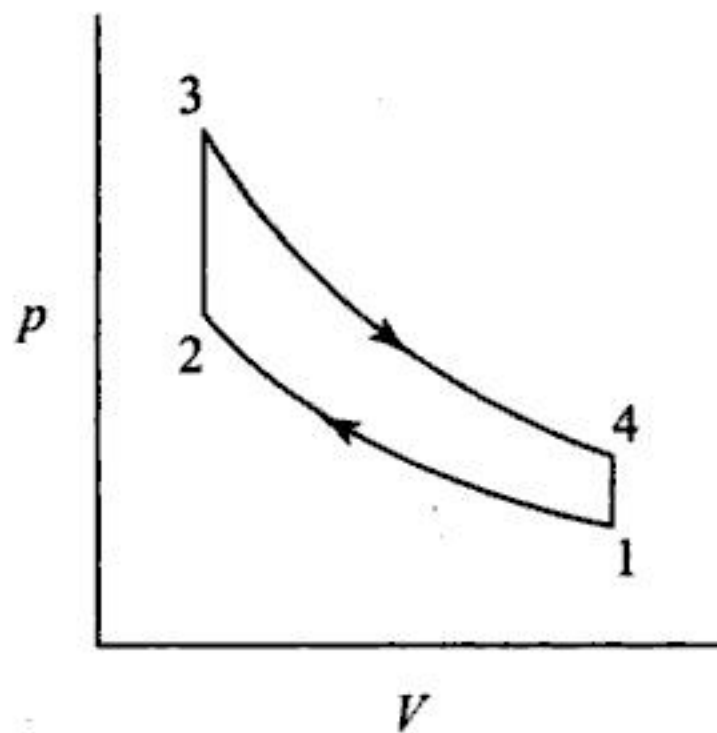
$$\begin{aligned}
 \text{Since } V_2 &= V_3 = V_c \\
 V_1 &= rV_2 = rV_c,
 \end{aligned}$$

Consider the process 1 – 2,

$$\frac{p_2}{p_1} = r^\gamma = 5.5^{1.4} = 10.88$$

$$p_2 = 10.88 \times 1 \times 10^5 = 10.88 \times 10^5 \text{ N/m}^2 \quad \underline{\underline{\text{Ans}}}$$

$$\frac{T_2}{T_1} = r^{\gamma-1} = 5.5^{0.4} = 1.978$$



$$T_2 = 1.978 \times 300 = 593.4 \text{ K} = 320.4^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

Consider the process 2 – 3,

$$p_3 = 30 \times 10^5 \text{ N/m}^2$$

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \frac{30}{10.88} = 2.757$$

$$T_3 = 2.757 \times 593.4 = 1636 \text{ K}$$

$$= 1363^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

Consider the process 3 – 4,

$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3} \right)^\gamma = \left(\frac{V_1}{V_2} \right)^\gamma$$

$$= r^\gamma = 5.5^{1.4} = 10.88$$

$$p_4 = \frac{p_3}{10.88} = 2.76 \times 10^5 \text{ N/m}^2 \quad \underline{\underline{\text{Ans}}}$$

$$\frac{T_3}{T_4} = r^{\gamma-1} = 5.5^{0.4} = 1.978$$

$$T_4 = \frac{T_3}{1.978} = \frac{1636}{1.978} = 827.1 \text{ K}$$

$$= 554.1^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{(\gamma-1)}} = 1 - \frac{1}{5.5^{0.4}} = 0.4943$$

$$= 49.43\% \quad \underline{\underline{\text{Ans}}}$$

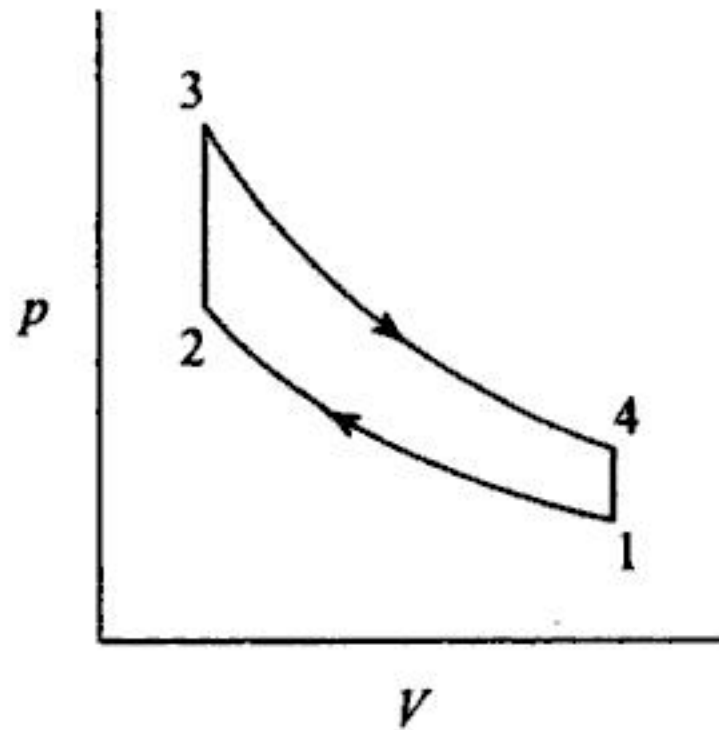
$$\begin{aligned}
 p_m &= \frac{\text{Indicated work/cycle}}{V_s} \\
 &= \frac{\text{Area of } p\text{-}V \text{ diagram } 1234}{V_s} \\
 \text{Area } 1234 &= \text{Area under } 3\text{-}4 - \text{Area under } 2\text{-}1 \\
 &= \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1} \\
 &= \frac{30 \times 10^5 \times V_c - 2.76 \times 10^5 \times 5.5 V_c}{0.4} \\
 &\quad - \frac{10.88 \times 10^5 \times V_c - 1 \times 10^5 \times 5.5 V_c}{0.4} \\
 &= 23.6 \times 10^5 \times V_c = p_m \times V_s \\
 p_m &= \frac{23.6 \times 10^5 \times V_c}{V_s} \\
 &= \frac{23.6 \times 10^5 \times V_c}{4.5 \times V_c} = 5.24 \times 10^5 \text{ N/m}^2 \\
 &= \mathbf{5.24 \text{ bar}} \quad \text{Ans}
 \end{aligned}$$

3.7 A gas engine operating on the ideal Otto cycle has a compression ratio of 6:1. The pressure and temperature at the commencement of compression are 1 bar and 27 °C. The heat added during the constant volume combustion process is 1170 kJ/kg. Determine the peak pressure and temperature, work output per kg of air and air-standard efficiency. Assume $C_v = 0.717 \text{ kJ/kg K}$ and $\gamma = 1.4$ for air.

Solution

Consider the process 1-2

$$\begin{aligned}
 \frac{p_2}{p_1} &= r^\gamma \\
 &= 6^{1.4} = 12.28 \\
 p_2 &= 12.28 \times 10^5 \text{ N/m}^2
 \end{aligned}$$



$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r^{\gamma-1} = 6^{0.4} = 2.05$$

$$T_2 = 2.05 \times 300 = 615\text{ K} = 342^\circ\text{ C}$$

Consider the process 2-3

For unit mass flow

$$\begin{aligned} q_s &= q_{2-3} = C_v(T_3 - T_2) \\ &= 1170 \text{ kJ/kg} \end{aligned}$$

$$T_3 - T_2 = \frac{1170}{0.717} = 1631.8$$

$$T_3 = 1631.8 + 615 = 2246.8 \text{ K}$$

$$= 1973.8^\circ\text{ C} \quad \underline{\underline{\text{Ans}}}$$

$$\frac{p_3}{p_2} = \frac{T_3}{T_2} = \frac{2246.8}{615} = 3.65$$

$$\text{Peak pressure, } p_3 = 3.65 \times 12.28 \times 10^5$$

$$= 44.82 \times 10^5 \text{ N/m}^2 = 44.82 \text{ bar} \quad \underline{\underline{\text{Ans}}}$$

$$\text{Work output} = \text{Area of } p\text{-}V \text{ diagram}$$

$$= \text{Area under } (3-4) - \text{Area under } (2-1)$$

$$= \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1}$$

$$= \frac{mR}{\gamma - 1} [(T_3 - T_4) - (T_2 - T_1)]$$

$$R = C_p - C_v = 1.004 - 0.717$$

$$= 0.287 \text{ kJ/kg K}$$

$$\frac{T_3}{T_4} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = r^{(\gamma-1)}$$

$$= 6^{0.4} = 2.048$$

$$T_4 = \frac{T_3}{2.048} = \frac{2246.8}{2.048} = 1097.1 \text{ K}$$

$$\text{Work output/kg} = \frac{0.287}{0.4} \cdot [(2246.8 - 1097.1) - (615 - 300)]$$

$$= \mathbf{598.9 \text{ kJ}} \quad \underline{\underline{\text{Ans}}}$$

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{(\gamma-1)}}$$

$$= 1 - \frac{1}{6^{0.4}} = 0.5116$$

$$= \mathbf{51.16\%} \quad \underline{\underline{\text{Ans}}}$$

3.8 A spark-ignition engine working on ideal Otto cycle has the compression ratio 6. The initial pressure and temperature of air are 1 bar and 37 °C. The maximum pressure in the cycle is 30 bar. For unit mass flow, calculate (i) p , V and T at various salient points of the cycle and (ii) the ratio of heat supplied to the heat rejected. Assume $\gamma = 1.4$ and $R = 8.314 \text{ kJ/kmol K}$.

Solution

Consider point 1,

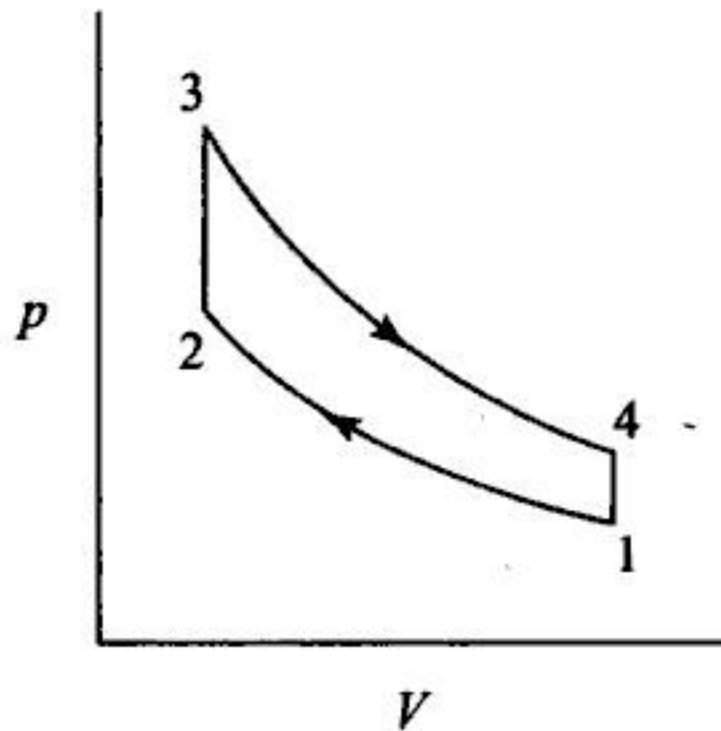
$$n = \frac{m}{M} = \frac{1}{29}$$

$$V_1 = \frac{nRT_1}{p_1} = \frac{1 \times 8314 \times 310}{29 \times 10^5} = \mathbf{0.889 \text{ m}^3} \quad \underline{\underline{\text{Ans}}}$$

Consider point 2,

$$p_2 = p_1 r^\gamma = 10^5 \times 6^{1.4}$$

$$= \mathbf{12.3 \times 10^5 \text{ N/m}^2 = 12.3 \text{ bar}} \quad \underline{\underline{\text{Ans}}}$$



$$V_2 = \frac{V_1}{6} = \frac{0.889}{6} = 0.148 \text{ m}^3 \quad \underline{\underline{\text{Ans}}}$$

$$\begin{aligned} T_2 &= \frac{p_2 V_2}{p_1 V_1} T_1 \\ &= \frac{12.3 \times 10^5 \times 0.148 \times 310}{1 \times 10^5 \times 0.889} = 634.8 \text{ K} \\ &= 361.8^\circ \text{ C} \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

Consider point 3,

$$V_3 = V_2 = 0.148 \text{ m}^3 \quad \underline{\underline{\text{Ans}}}$$

$$p_3 = 30 \times 10^5 \text{ N/m}^2 = 30 \text{ bar} \quad \underline{\underline{\text{Ans}}}$$

$$\frac{p_3}{T_3} = \frac{p_2}{T_2}$$

$$\begin{aligned} T_3 &= \frac{30 \times 10^5}{12.3 \times 10^5} \times 634.8 = 1548 \text{ K} \\ &= 1275^\circ \text{ C} \quad \underline{\underline{\text{Ans}}} \end{aligned}$$

Consider point 4,

$$p_3 V_3^\gamma = p_4 V_4^\gamma$$

$$p_4 = p_3 \left(\frac{V_3}{V_4} \right)^\gamma = 30 \times 10^5 \left(\frac{1}{6} \right)^{1.4}$$

$$= 2.44 \times 10^5 \text{ N/m}^2 = 2.44 \text{ bar} \quad \underline{\underline{\text{Ans}}}$$

$$V_4 = V_1 = 0.889 \text{ m}^3 \quad \underline{\underline{\text{Ans}}}$$

$$T_4 = T_1 \frac{p_4}{p_1} = 310 \times \frac{2.44 \times 10^5}{1 \times 10^5}$$

$$= 756.4 \text{ K} = 483.4^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

$$C_v = \frac{R}{M(\gamma - 1)} = \frac{8.314}{29 \times 0.4}$$

$$= 0.717 \text{ kJ/kg K}$$

For unit mass,

$$\text{Heat supplied} = C_v(T_3 - T_2)$$

$$= 0.717 \times (1548 - 635.5) = 654.3 \text{ kJ}$$

$$\text{Heat rejected} = C_v(T_4 - T_1)$$

$$= 0.717 \times (756.4 - 310) = 320.1 \text{ kJ}$$

$$\frac{\text{Heat supplied}}{\text{Heat rejected}} = \frac{654.3}{320.1} = 2.04 \quad \underline{\underline{\text{Ans}}}$$

3.9 In an Otto engine, pressure and temperature at the beginning of compression are 1 bar and 37°C respectively. Calculate the theoretical thermal efficiency of this cycle if the pressure at the end of the adiabatic compression is 15 bar. Peak temperature during the cycle is 2000 K. Calculate (i) the heat supplied per kg of air (ii) the work done per kg of air and (iii) the pressure at the end of adiabatic expansion. Take $C_v = 0.717 \text{ kJ/kg K}$ and $\gamma = 1.4$.

Solution

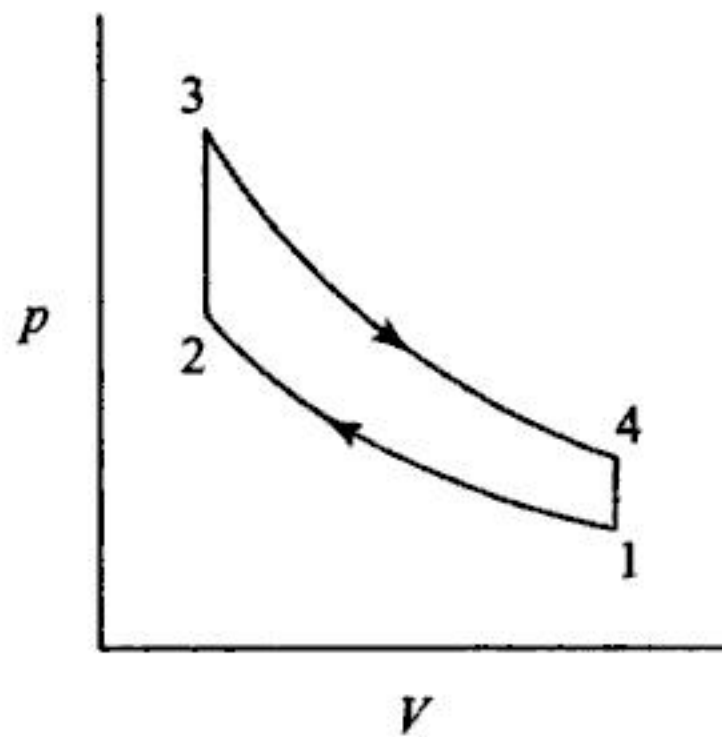
Consider the process 1-2

$$T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = 310 \times (15)^{(0.4/1.4)}$$

$$= 672 \text{ K}$$

$$\eta = 1 - \frac{T_1}{T_2} = 1 - \frac{310}{672} = 0.539$$

$$= 53.9\%$$



Consider the process 2 – 3

For unit mass flow

$$\begin{aligned}
 \text{Heat supplied, } q_s &= C_v(T_3 - T_2) \\
 &= 0.717 \times (2000 - 672) \\
 &= \mathbf{952.2 \text{ kJ/kg}} \quad \text{Ans}
 \end{aligned}$$

$$\eta = \frac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}} = \frac{w}{q_s}$$

$$\begin{aligned}
 w &= \eta q_s = 0.539 \times 952.2 \\
 &= \mathbf{513.2 \text{ kJ/kg}} \quad \text{Ans}
 \end{aligned}$$

$$\begin{aligned}
 p_3 &= p_2 \left(\frac{T_3}{T_2} \right) = 15 \times \frac{2000}{672} \\
 &= \mathbf{44.64 \text{ bar}}
 \end{aligned}$$

Consider the process 3 – 4

$$\begin{aligned}
 \frac{p_3}{p_4} &= \left(\frac{V_4}{V_3} \right)^\gamma \\
 &= \left(\frac{V_1}{V_2} \right)^\gamma = \frac{p_2}{p_1} \\
 p_4 &= p_3 \left(\frac{p_1}{p_2} \right) = \frac{44.64 \times 1}{15} \\
 &= \mathbf{2.98 \text{ bar}} \quad \text{Ans}
 \end{aligned}$$

- 3.10 Compare the efficiencies of ideal Atkinson cycle and Otto cycle for a compression ratio is 5.5. The pressure and temperature of air at the beginning of compression stroke are 1 bar and 27 °C respectively. The peak pressure is 25 bar for both cycles. Assume $\gamma = 1.4$ for air.

Solution

$$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{5.5^{0.4}} = 49.43\%$$

(Refer Fig.3.14)

$$\eta_{Atkinson} = 1 - \frac{\gamma(e - r)}{e^{\gamma} - r^{\gamma}}$$

$$r = 5.5$$

$$e = \frac{V_{4'}}{V_3} = \left(\frac{p_3}{p_{4'}} \right)^{\frac{1}{\gamma}} = \left(\frac{25}{1} \right)^{\frac{1}{1.4}} = 9.966$$

$$e^{\gamma} = 9.966^{1.4} = 25$$

$$r^{\gamma} = 5.5^{1.4} = 10.88$$

$$\eta_{Atkinson} = 1 - \frac{1.4 \times (9.966 - 5.5)}{25 - 10.88} = 55.72\%$$

$$\frac{\eta_{Atkinson}}{\eta_{Otto}} = \frac{55.72}{49.43} = 1.127 \quad \text{Ans} \quad \leftarrow$$

DIESEL CYCLE

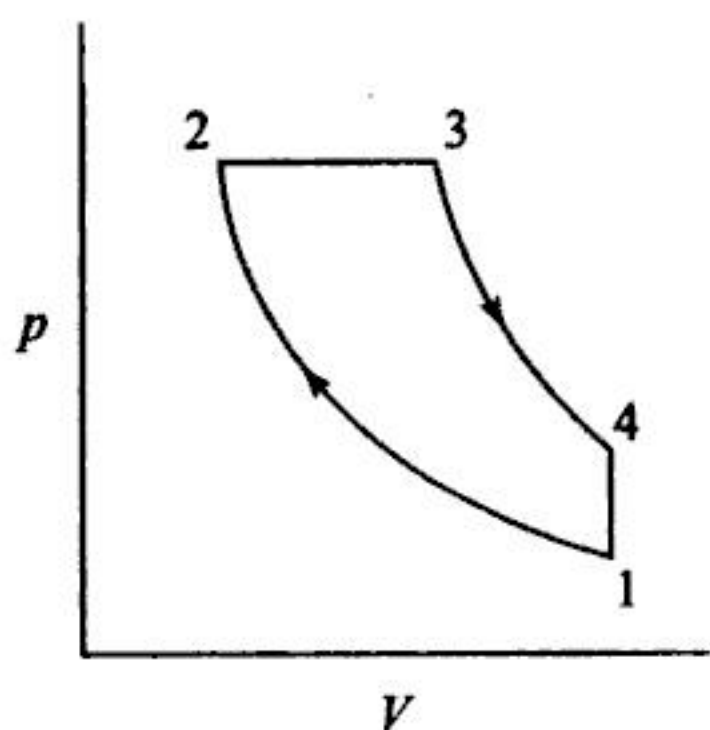
- 3.11 A Diesel engine has a compression ratio of 20 and cut-off takes place at 5% of the stroke. Find the air-standard efficiency. Assume $\gamma = 1.4$.

Solution

$$r = \frac{V_1}{V_2} = 20$$

$$V_1 = 20V_2$$

$$V_s = 20V_2 - V_2 = 19V_2$$



$$V_3 = 0.05V_s + V_2$$

$$= 0.05 \times 19V_2 + V_2 = 1.95V_2$$

$$r_c = \frac{V_3}{V_2} = \frac{1.95V_2}{V_2} = 1.95$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{r_c^\gamma - 1}{\gamma(r_c - 1)}$$

$$= 1 - \frac{1}{20^{0.4}} \times \left[\frac{1.95^{1.4} - 1}{1.4 \times (1.95 - 1)} \right] = 0.649$$

$$= 64.9\%$$

Ans

3.12 Determine the ideal efficiency of the diesel engine having a cylinder with bore 250 mm, stroke 375 mm and a clearance volume of 1500 cc, with fuel cut-off occurring at 5% of the stroke. Assume $\gamma = 1.4$ for air.

Solution

$$V_s = \frac{\pi}{4} d^2 L = \frac{\pi}{4} \times 25^2 \times 37.5$$

$$= 18407.8 \text{ cc}$$

$$r = 1 + \frac{V_s}{V_c} = 1 + \frac{18407.8}{1500} = 13.27$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{r_c^\gamma - 1}{\gamma(r_c - 1)}$$

$$r_c = \frac{V_3}{V_2}$$

$$\begin{aligned} \text{Cut-off volume} &= V_3 - V_2 = 0.05V_s \\ &= 0.05 \times 12.27V_c \end{aligned}$$

$$V_2 = V_c$$

$$V_3 = 1.6135V_c$$

$$r_c = \frac{V_3}{V_2} = 1.6135$$

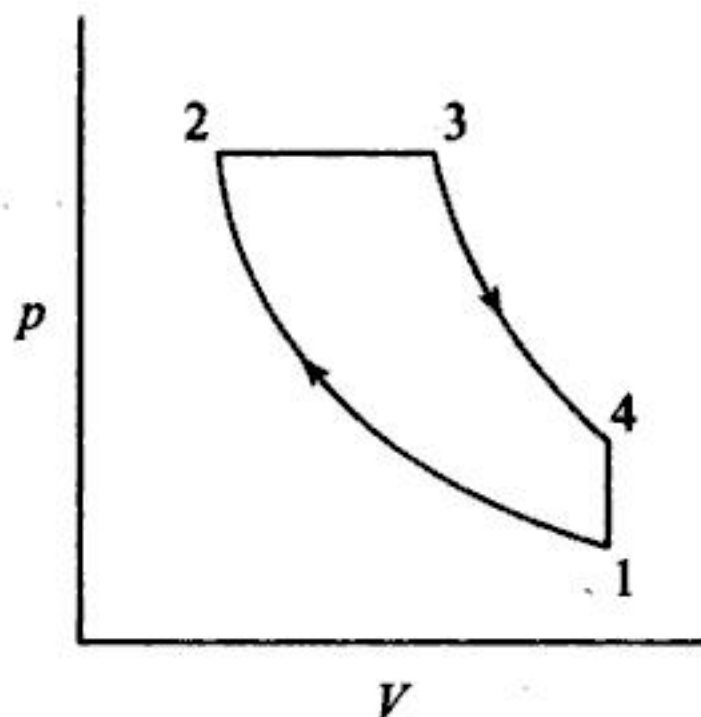
$$\eta = 1 - \frac{1}{13.27^{0.4}} \times \frac{1.6135^{1.4} - 1}{1.4 \times (1.6135 - 1)}$$

$$= 0.6052 = 60.52\%$$

Ans

3.13 In an engine working on Diesel cycle inlet pressure and temperature are 1 bar and 17 °C respectively. Pressure at the end of adiabatic compression is 35 bar. The ratio of expansion i.e. after constant pressure heat addition is 5. Calculate the heat addition, heat rejection and the efficiency of the cycle. Assume $\gamma = 1.4$, $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K.

Solution



Consider the process 1 – 2

$$\frac{V_1}{V_2} = r = \left(\frac{p_2}{p_1} \right)^{\frac{1}{\gamma}}$$

$$= \left(\frac{35}{1}\right)^{\frac{1}{1.4}} = 12.674$$

$$\text{Cut-off ratio} = \frac{V_3}{V_2} = \frac{V_3}{V_1} \times \frac{V_1}{V_2}$$

$$= \frac{\text{Compression ratio}}{\text{Expansion ratio}}$$

$$= \frac{12.674}{5} = 2.535$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{35}{1}\right)^{0.286} = 2.76$$

$$T_2 = 2.76 \times 290 = 801.7 \text{ K}$$

Consider the process 2 – 3

$$T_3 = T_2 \frac{V_3}{V_2} = 801.7 \times \frac{V_3}{V_2}$$

$$= 801.7 \times 2.535 = 2032.3 \text{ K}$$

Consider the process 3 – 4

$$T_4 = T_3 \left(\frac{V_3}{V_4}\right)^{\gamma-1} = 2032.3 \times \left(\frac{1}{5}\right)^{0.4}$$

$$= 1067.6 \text{ K}$$

$$\text{Heat added} = C_p(T_3 - T_2) = 1.004 \times (2032.3 - 801.7)$$

$$= 1235.5 \text{ kJ/kg} \quad \text{Ans}$$

$$\text{Heat rejected} = C_v(T_4 - T_1) = 0.717 \times (1067.6 - 290)$$

$$= 557.5 \text{ kJ/kg} \quad \text{Ans}$$

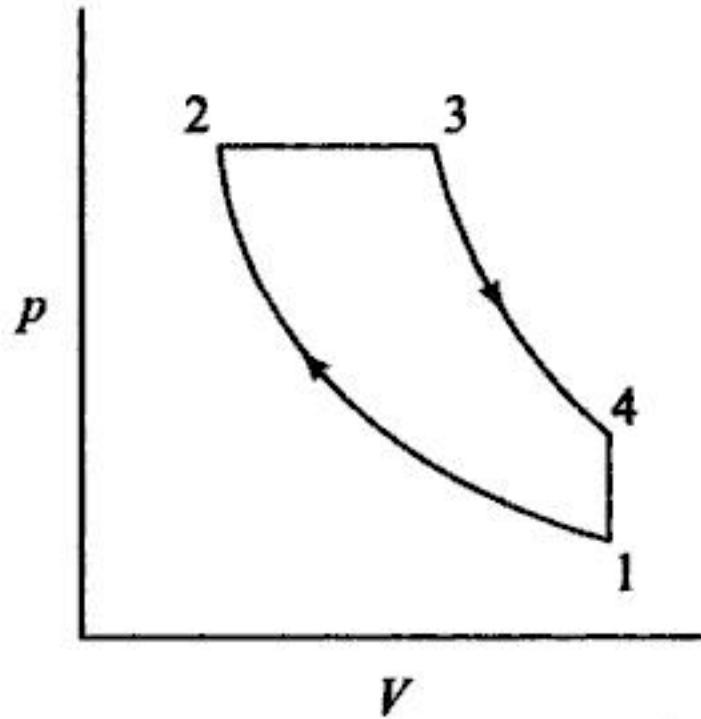
$$\text{Efficiency} = \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}}$$

$$= \frac{1235.5 - 557.5}{1235.5} = 0.549$$

$$= 54.9\% \quad \text{Ans}$$

- 3.14 A Diesel engine is working with a compression ratio of 15 and expansion ratio of 10. Calculate the air-standard efficiency of the cycle. Assume $\gamma = 1.4$.

Solution



$$r = \frac{V_1}{V_2} = 15$$

$$r_e = \frac{V_4}{V_3} = 10$$

$$\begin{aligned} \eta &= 1 - \frac{1}{\gamma} \frac{1}{r^{\gamma-1}} \left[\frac{\left(\frac{r}{r_e}\right)^{\gamma} - 1}{\left(\frac{r}{r_e}\right) - 1} \right] \\ &= 1 - \frac{1}{1.4} \times \frac{1}{15^{0.4}} \times \left[\frac{\left(\frac{15}{10}\right)^{1.4} - 1}{\left(\frac{15}{10}\right) - 1} \right] = 0.63 \\ &= \mathbf{63\%} \end{aligned}$$

Ans

- 3.15 A Diesel engine works on Diesel cycle with a compression ratio of 15 and cut-off ratio of 1.75. Calculate the air-standard efficiency assuming $\gamma = 1.4$.

Solution

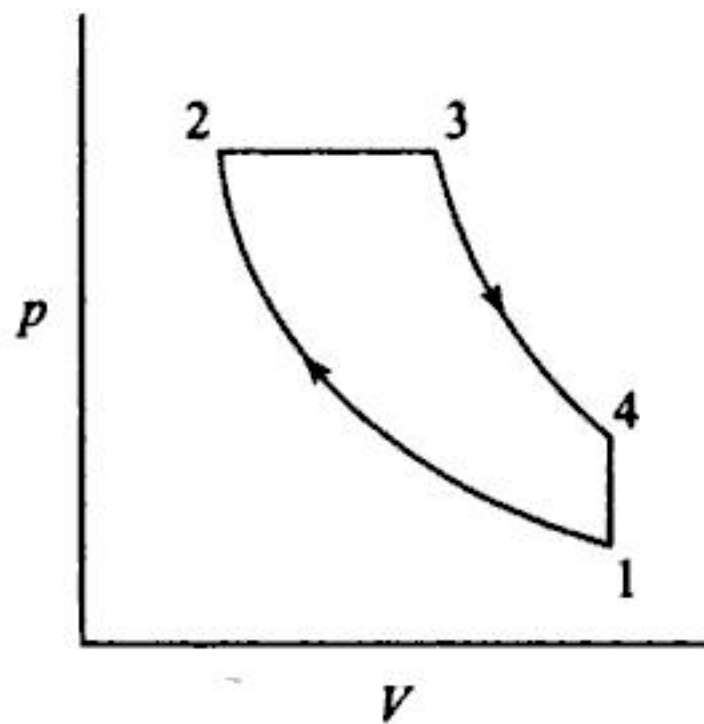
$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{1}{\gamma} \left(\frac{r_c^{\gamma} - 1}{r - 1} \right)$$

$$= 1 - \frac{1}{15^{0.4}} \times \frac{1}{1.4} \times \left(\frac{1.75^{1.4} - 1}{1.75 - 1} \right) = 0.617$$

$$= 61.7\%$$

Ans

3.16 A Diesel cycle operates at a pressure of 1 bar at the beginning of compression and the volume is compressed to $\frac{1}{16}$ of the initial volume. Heat is supplied until the volume is twice that of the clearance volume. Calculate the mean effective pressure of the cycle. Take $\gamma = 1.4$.



Solution

$$V_1 = 16V_2$$

$$V_3 = 2V_2$$

Swept volume

$$= V_1 - V_2 = (r - 1)V_2 = 15 V_2$$

$$V_2 = \frac{V_s}{15}$$

Consider the process 1 - 2

$$p_2 = p_1 \left(\frac{V_1}{V_2} \right)^\gamma = 1 \times 16^{1.4} = 48.5 \text{ bar}$$

$$p_2 = p_3 = 48.5 \text{ bar}$$

$$p_4 = p_3 \left(\frac{V_3}{V_4} \right)^{1.4} = 48.5 \times \left(\frac{2}{16} \right)^{1.4}$$

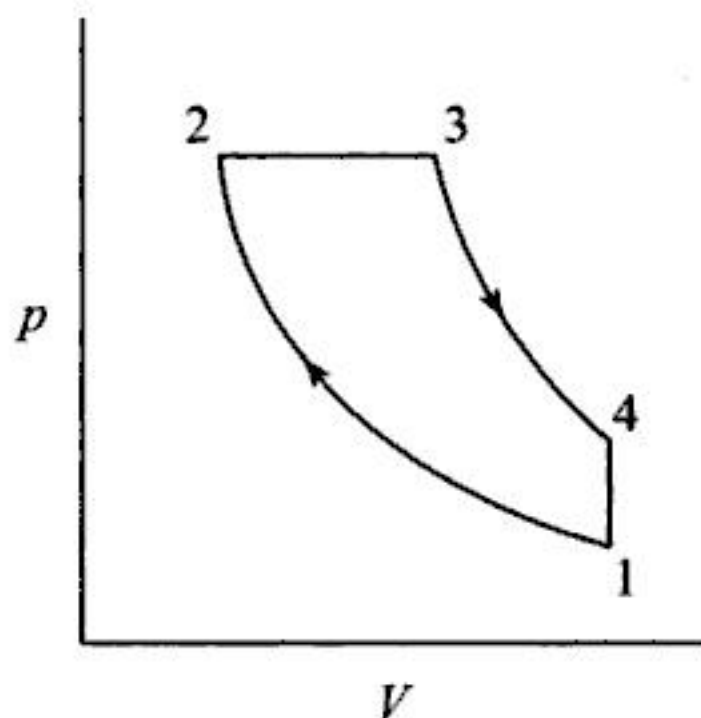
$$= \frac{48.5}{8^{1.4}} = 2.64 \text{ bar} \quad (\text{Since, } V_4 = V_1)$$

Mean effective pressure, p_m , is given by

$$\begin{aligned} p_m &= \frac{1}{V_s} \left[p_2(V_3 - V_2) + \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1} \right] \\ &= \frac{V_2}{V_s} \left[p_3 \left(\frac{V_3}{V_2} - 1 \right) + \frac{\left(p_3 \frac{V_3}{V_2} - p_4 \frac{V_4}{V_2} \right)}{\gamma - 1} - \frac{\left(p_2 - p_1 \frac{V_1}{V_2} \right)}{\gamma - 1} \right] \\ &= \frac{1}{15} \times \left[48.5 \times (2 - 1) + \frac{48.5 \times 2 - 2.64 \times 16}{1.4 - 1} - \frac{48.5 - 1 \times 16}{1.4 - 1} \right] = 6.94 \text{ bar} \quad \text{Ans} \end{aligned}$$

- 3.17 In an engine working on the Diesel cycle the ratios of the weights of air and fuel supplied is 50 : 1. The temperature of air at the beginning of the compression is 60 °C and the compression ratio used is 14 : 1. What is the ideal efficiency of the engine. Calorific value of fuel used is 42000 kJ/kg. Assume $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K for air.

Solution



$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{r_c^\gamma - 1}{\gamma(r_c - 1)}$$

$$\gamma = \frac{C_p}{C_v} = \frac{1.004}{0.717} = 1.4$$

$$\text{Cut-off ratio, } r_c = \frac{V_3}{V_2} = \frac{T_3}{T_2}$$

Consider the process 1 – 2

$$\begin{aligned}\frac{T_2}{T_1} &= \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} \\ &= r^{(\gamma-1)} = 14^{0.4} = 2.874 \\ T_2 &= 2.874 \times 333 = 957.04 \text{ K}\end{aligned}$$

Consider the process 2 – 3

$$\text{Heat added/kg of air} = C_p (T_3 - T_2) = F/A \times CV$$

$$\begin{aligned}T_3 - T_2 &= \frac{F/A \times CV}{C_p} = \frac{42000}{50 \times 1.004} \\ &= 836.6\end{aligned}$$

$$T_3 = 1793.64 \text{ K}$$

$$r_c = \frac{T_3}{T_2} = \frac{1793.64}{957.04} = 1.874$$

$$\eta = 1 - \frac{1}{1.4 \times 14^{0.4}} \times \left(\frac{1.874^{1.4} - 1}{0.874} \right)$$

$$= 0.60 = 60\% \quad \underline{\underline{\text{Ans}}}$$

3.18 In an ideal Diesel cycle, the pressure and temperature are 1.03 bar and 27 °C respectively. The maximum pressure in the cycle is 47 bar and the heat supplied during the cycle is 545 kJ/kg. Determine (i) the compression ratio (ii) the temperature at the end of compression (iii) the temperature at the end of constant pressure combustion and (iv) the air-standard efficiency. Assume $\gamma = 1.4$ and $C_p = 1.004 \text{ kJ/kg K}$ for air.

Solution

$$p_2 = p_3 = 47 \times 10^5 \text{ N/m}^2$$

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^\gamma = r^\gamma$$

$$r = \left(\frac{p_2}{p_1}\right)^{\left(\frac{1}{\gamma}\right)} = \left(\frac{47}{1.03}\right)^{\left(\frac{1}{1.4}\right)}$$

$$= 15.32 \quad \underline{\underline{\text{Ans}}}$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{(\gamma-1)} = r^{(\gamma-1)}$$

$$= 15.32^{0.4} = 2.979$$

$$T_2 = 2.979 \times 300 = 893.7 \text{ K}$$

$$= 620.7^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

$$\text{Heat supplied/kg} = C_p (T_3 - T_2) = 545$$

$$T_3 - T_2 = \frac{545}{1.004} = 542.8$$

$$T_3 = 542.8 + 893.7 = 1436.5 \text{ K}$$

$$= 1163.5^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{r_c^\gamma - 1}{\gamma(r_c - 1)}$$

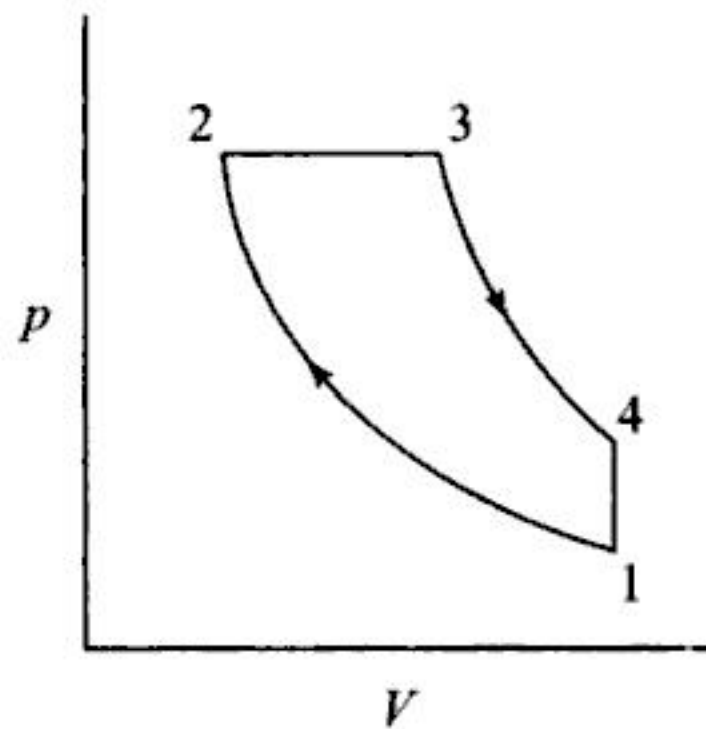
$$r_c = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{1436.5}{893.7} = 1.61$$

$$\eta_{\text{Diesel}} = 1 - \left[\frac{1}{1.4 \times 15.32^{0.4}} \times \left(\frac{1.61^{1.4} - 1}{0.61} \right) \right]$$

$$= 0.6275 = 62.75\% \quad \underline{\underline{\text{Ans}}}$$

3.19 A diesel engine operating on the air-standard Diesel cycle has six cylinders of 100 mm bore and 120 mm stroke. The engine speed is 1800 rpm. At the beginning of compression the pressure and temperature of air are 1.03 bar and 35 °C. If the clearance volume is 1/8th of the stroke volume, calculate (i) the pressure and temperature at the salient points of the cycle (ii) the compression ratio (iii) the efficiency of the cycle and (iv) the power output if the air is heated to 1500 °C Assume C_p and C_v of air to be 1.004 and 0.717 kJ/kg K respectively.

Solution



$$r = 1 + \frac{V_s}{V_c} = 1 + 8 = 9 \quad \text{Ans}$$

Consider the process 1 – 2,

$$\frac{p_2}{p_1} = r^\gamma = 9^{1.4} = 21.67$$

$$p_2 = 21.67 \times 1.03 \times 10^5 = 22.32 \times 10^5 \text{ N/m}^2$$

$$= \mathbf{22.32 \text{ bar}} \quad \text{Ans}$$

$$\frac{T_2}{T_1} = r^{(\gamma-1)} = 9^{0.4} = 2.408$$

$$T_2 = 308 \times 2.408 = 741.6 \text{ K} = \mathbf{468.6^\circ \text{C}} \quad \text{Ans}$$

Consider the process 2 – 3,

$$p_3 = p_2 = 22.32 \times 10^5 \text{ N/m}^2 = \mathbf{22.32 \text{ bar}} \quad \text{Ans}$$

$$T_3 = \mathbf{1773 \text{ K} = 1500^\circ \text{C}} \quad \text{Ans}$$

Consider the process 3 – 4,

$$\frac{T_3}{T_4} = r_e^{(\gamma-1)}$$

$$r_c = \frac{T_3}{T_2} = \frac{1773}{741.6} = 2.39$$

$$r_e = \frac{r}{r_c} = \frac{9}{2.391} = 3.764$$

$$\frac{T_3}{T_4} = 1.7$$

$$T_4 = \frac{T_3}{1.7} = \frac{1773}{1.7} = 1042.9 \text{ K}$$

$$= 769.9^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}$$

$$\frac{p_3}{p_4} = r_e^\gamma = 3.764^{1.4} = 6.396$$

$$p_4 = \frac{p_3}{6.396} = \frac{22.32 \times 10^5}{6.396}$$

$$= 3.49 \times 10^5 \text{ N/m}^2 = 3.49 \text{ bar} \quad \underline{\underline{\text{Ans}}}$$

$$\eta_{\text{Cycle}} = \frac{\text{Work output}}{\text{Heat added}} = 1 - \frac{\text{Heat rejected}}{\text{Heat added}}$$

$$= 1 - \frac{q_{4-1}}{q_{2-3}}$$

$$q_{4-1} = C_v (T_4 - T_1)$$

$$= 0.717 \times (1042.9 - 308) = 526.9 \text{ kJ/kg}$$

$$q_{2-3} = C_p (T_3 - T_2)$$

$$= 1.004 \times (1773 - 741.6)$$

$$= 1035.5 \text{ kJ/kg}$$

$$\eta_{\text{Cycle}} = 1 - \frac{526.9}{1035.5} = 0.4912$$

$$= 49.12\% \quad \underline{\underline{\text{Ans}}}$$

$$\text{Work output} = q_{2-3} - q_{4-1}$$

$$= 1035.5 - 526.9 = 508.6 \text{ kJ/kg}$$

$$\text{Power output} = \text{Work output} \times \dot{m}_a$$

$$\dot{m}_a = \frac{p_1 V_1}{RT_1} \times \frac{N}{2}$$

$$R = C_p - C_v = 0.287 \text{ kJ/kg K}$$

$$V_1 = V_s + V_c = \frac{9}{8} V_s$$

$$V_s = 6 \times \frac{\pi}{4} d^2 L = 6 \times \frac{\pi}{4} \times 10^2 \times 12$$

$$= 5654.8 \text{ cc} = 5.65 \times 10^{-3} \text{ m}^3$$

$$V_1 = 5.65 \times 10^{-3} \times \frac{9}{8} = 6.36 \times 10^{-3} \text{ m}^3$$

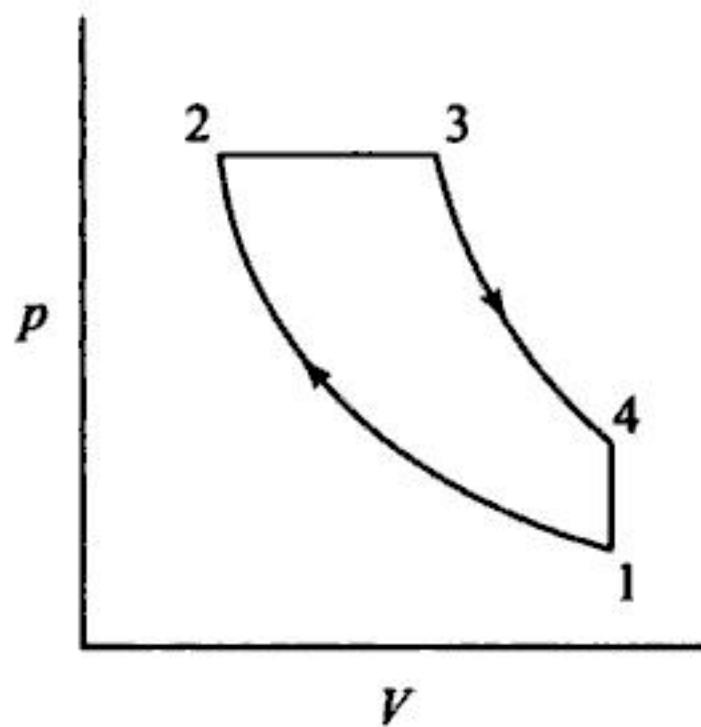
$$\dot{m}_a = \frac{1.03 \times 10^5 \times 6.36 \times 10^{-3} \times 30}{287 \times 308 \times 2}$$

$$= 0.111 \text{ kg/s}$$

$$\text{Power output} = 508.6 \times 0.111 = 56.45 \text{ kW} \quad \underline{\underline{\text{Ans}}}$$

3.20 The mean effective pressure of an ideal Diesel cycle is 8 bar. If the initial pressure is 1.03 bar and the compression ratio is 12, determine the cut-off ratio and the air-standard efficiency. Assume ratio of specific heats for air to be 1.4.

Solution



$$\begin{aligned} \text{Work output} &= p_m \times V_s \\ &= \text{Area 1234} \\ &= \text{Area under 2-3} + \text{Area under 3-4} - \end{aligned}$$

$$\text{Area under 2-1}$$

$$= p_2 (V_3 - V_2) + \frac{p_3 V_3 - p_4 V_4}{\gamma - 1} - \frac{p_2 V_2 - p_1 V_1}{\gamma - 1}$$

$$r = \frac{V_1}{V_2} = 1 + \frac{V_s}{V_c} = 12$$

$$V_s = 11V_c \quad ; \quad V_2 = V_c$$

$$V_1 = V_4 = 12V_2 = 12V_c$$

$$V_3 = r_c V_2 = r_c V_c$$

$$\frac{p_2}{p_1} = r^\gamma = 12^{1.4} = 32.42$$

$$\begin{aligned} p_2 &= 32.42 \times 1.03 \times 10^5 \\ &= 33.39 \times 10^5 \text{ N/m}^2 = p_3 \end{aligned}$$

$$\frac{p_3}{p_4} = \left(\frac{r}{r_c} \right)^{1.4} = \frac{12^{1.4}}{r_c^{1.4}} = \frac{32.42}{r_c^{1.4}}$$

$$p_4 = \frac{33.39}{32.42} \times r_c^{1.4} \times 10^5 = 1.03 r_c^{1.4} \times 10^5$$

$$\begin{aligned} \text{Area 1234} &= 33.39 (r_c V_c - V_c) + \\ &\quad \frac{33.39 \times r_c V_c - 1.03 r_c^{1.4} \times 12 V_c}{0.4} - \\ &\quad \frac{33.39 \times V_c - 1.03 \times 12 V_c}{0.4} \times 10^5 \end{aligned}$$

$$p_m \times V_s = 8 \times 11 V_c \times 10^5$$

$$\text{Area 1234} = p_m \times V_s$$

Substituting and simplifying

$$0.672 r_c - 0.178 r_c^{1.4} = 1$$

Solving by iteration,

$$r_c = 2.38 \quad \text{Ans}$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{r_c^\gamma - 1}{\gamma(r_c - 1)}$$

$$= 1 - \frac{1}{1.4 \times 12^{0.4}} \times \left(\frac{2.38^{1.4} - 1}{1.38} \right)$$

$$= 0.5466 = 54.66\% \quad \text{Ans}$$



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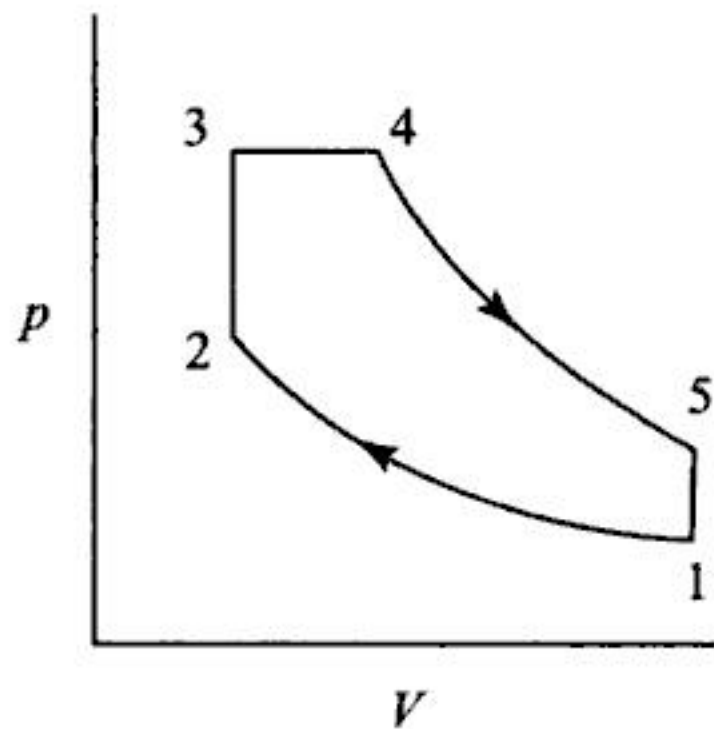


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various salient points of the cycle and the cycle efficiency. The pressure and temperature of air at the commencement of compression are 1 bar and 100 °C respectively. Assume $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K for air.

Solution



$$\frac{V_s}{V_c} = r - 1 = 9$$

$$\gamma = \frac{C_p}{C_v} = \frac{1.004}{0.717} = 1.4$$

Consider the process 1 – 2

$$\frac{p_2}{p_1} = r^\gamma = 10^{1.4} = 25.12$$

$$p_2 = 25.12 \times 10^5 \text{ N/m}^2 = \mathbf{25.12 \text{ bar}} \quad \xleftarrow{\text{Ans}}$$

$$\frac{T_2}{T_1} = r^{(\gamma-1)} = 10^{0.4} = 2.512$$

$$T_2 = 2.512 \times 373 = 936.9 \text{ K} = \mathbf{663.9^\circ \text{C}} \quad \xleftarrow{\text{Ans}}$$

Consider the process 2 – 3 and 3 – 4

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \frac{70}{25.12} = 2.787$$

$$T_3 = 2.787 \times 936.9 = 2611.1 \text{ K}$$

$$= \mathbf{2338^\circ \text{C}} \quad \xleftarrow{\text{Ans}}$$

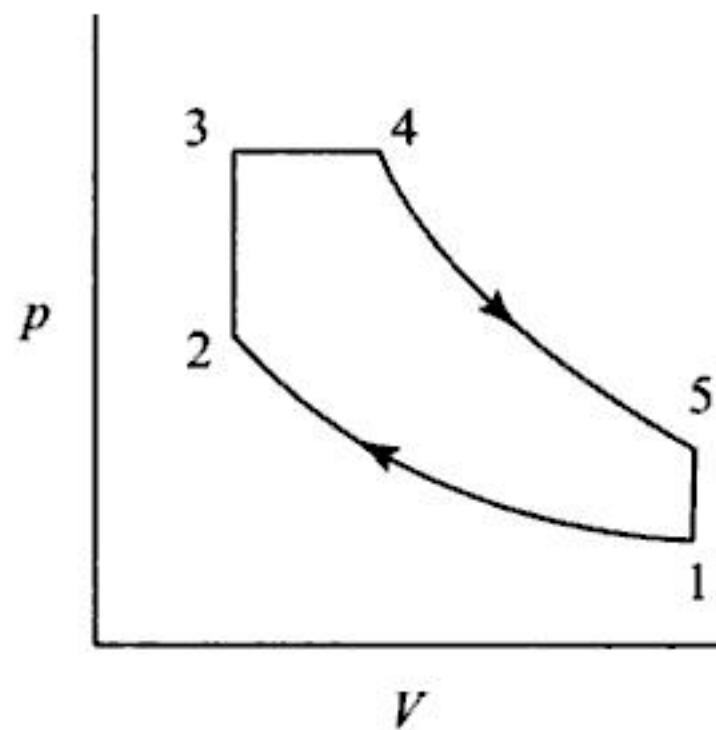


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$$\eta = \frac{1680 - 674.98}{1680} = 59.82\% \quad \underline{\underline{\text{Ans}}}$$

- 3.23 An oil engine works on the Dual cycle, the heat liberated at constant pressure being twice that liberated at constant volume. The compression ratio of the engine is 8 and the expansion ratio is 5.3. But the compression and expansion processes follow the law $pV^{1.3} = C$. The pressure and temperature at the beginning of compression are 1 bar and 27 °C respectively. Assuming $C_p = 1.004$ kJ/kg K and $C_v = 0.717$ kJ/kg K for air, find the air-standard efficiency and the mean effective pressure.



Solution

$$\gamma = \frac{C_p}{C_v} = \frac{1.004}{0.717} = 1.4$$

$$\frac{V_s}{V_c} = r - 1 = 7$$

$$V_s = 7V_c$$

$$r_e = \frac{r}{r_c} = 5.3$$

$$r_c = \frac{8}{5.3} = 1.509$$

$$\text{Mean effective pressure} = \frac{\text{Area 12345}}{V_s}$$

$$\text{Area 12345} = \text{Area under 3-4} +$$

$$\text{Area under 4-5} - \text{Area under 2-1}$$

$$\begin{aligned}
&= p_3 (V_3 - V_4) + \frac{p_4 V_4 - p_5 V_5}{n-1} - \frac{p_2 V_2 - p_1 V_1}{n-1} \\
V_2 &= V_3 = V_c \\
V_1 &= V_5 = r V_c = 8 V_c \\
V_4 &= r_c V_3 = 1.509 V_c \\
\frac{T_2}{T_1} &= r^{(n-1)} = 8^{0.3} = 1.866 \\
T_2 &= 1.866 \times 300 = 559.82 \text{ K} \\
\frac{p_2}{p_1} &= r^n = 8^{1.3} = 14.93 \\
p_2 &= 14.93 \times p_1 = 14.93 \times 10^5 \text{ N/m}^2
\end{aligned}$$

Heat released during constant pressure combustion

$$= 2 \times \text{Heat released during constant volume combustion}$$

$$\begin{aligned}
C_p (T_4 - T_3) &= 2 C_v (T_3 - T_2) \\
1.004 \times (T_4 - T_3) &= 2 \times 0.717 \times (T_3 - T_2) \\
T_4 - T_3 &= 1.428 \times (T_3 - T_2) \\
\frac{T_4}{T_3} &= \frac{V_4}{V_3} = r_c = 1.509 \\
T_4 &= 1.509 T_3 \\
1.509 T_3 - T_3 &= 1.428 \times (T_3 - 559.82) \\
T_3 &= 869.88 \text{ K} \\
T_4 &= 1312.65 \text{ K} \\
\frac{p_3}{p_2} &= \frac{T_3}{T_2} = \frac{869.88}{559.82} = 1.554 \\
p_3 &= 1.554 \times 14.93 \times 10^5 \\
&= 23.20 \times 10^5 \text{ N/m}^2 = p_4
\end{aligned}$$

$$\frac{T_4}{T_5} = r_e^{(n-1)} = 5.3^{0.3} = 1.649$$

$$T_5 = \frac{1312.65}{1.649} = 796.03 \text{ K}$$

$$\frac{p_4}{p_5} = r_e^n = 5.3^{1.3} = 8.741$$

$$p_5 = \frac{p_4}{8.741} = \frac{23.2 \times 10^5}{8.741}$$

$$= 2.654 \times 10^5 \text{ N/m}^2$$

$$\begin{aligned} \text{Area } 12345 &= \left[\frac{23.2 \times 1.509V_c - 2.654 \times 8V_c}{0.3} + \right. \\ &\quad \left. 23.2 \times (1.509V_c - V_c) - \right. \\ &\quad \left. \frac{14.93 \times V_c - 1 \times 8V_c}{0.3} \right] \times 10^5 \end{aligned}$$

$$= 34.63 \times V_c \times 10^5 \text{ N/m}^2$$

$$V_c = \frac{V_1}{8}$$

$$\text{Area } 12345 = p_m \times V_s = p_m \times 7 \times V_c$$

Therefore,

$$p_m = \frac{34.63}{7} = 4.95 \times 10^5 \text{ N/m}^2$$

$$= 4.95 \text{ bar}$$

Ans

$$\eta = \frac{w}{q_s}$$

$$v_1 = \frac{mRT_1}{p_1} = \frac{1 \times 287 \times 300}{1 \times 10^5}$$

$$= 0.861 \text{ m}^3/\text{kg}$$

$$w = 34.63 \times 10^5 \times \frac{v_1}{8}$$

$$= 34.63 \times 10^5 \times \frac{0.861}{8}$$

$$= 3.727 \times 10^5 \text{ J/kg} = 372.7 \text{ kJ/kg}$$

Note : Work done should be calculated only from the area ($\int p dv$) for a polytropic process

$$q_s = q(T_3 - T_2) + C_p(T_4 - T_3)$$

$$= 667.1 \text{ kJ/kg}$$

$$\eta = \frac{372.7}{667.1} \times 100 = 55.9\% \quad \underline{\underline{\text{Ans}}}$$

COMPARISON OF CYCLES

- 3.24 A four-cylinder, four-stroke, spark-ignition engine has a displacement volume of 300 cc per cylinder. The compression ratio of the engine is 10 and operates at a speed of 3000 rev/min. The engine is required to develop an output of 40 kW at this speed. Calculate the cycle efficiency, the necessary rate of heat addition, the mean effective pressure and the maximum temperature of the cycle. Assume that the engine operates on the Otto cycle and that the pressure and temperature at the inlet conditions are 1 bar and 27 °C respectively. If the above engine is a compression-ignition engine operating on the Diesel cycle and receiving heat at the same rate, calculate efficiency, the maximum temperature of the cycle, the cycle efficiency, the power output and the mean effective pressure. Take $C_v = 0.717 \text{ kJ/kg K}$ and $\gamma = 1.4$.

Solution

Consider the Otto cycle, Fig.3.9(a)

$$\eta = 1 - \frac{1}{r^{\gamma-1}}$$

$$= 1 - \frac{1}{10^{0.4}} = 0.602$$

$$= 60.2\% \quad \underline{\underline{\text{Ans}}}$$

$$\eta = \frac{\text{Power output}}{\text{Heat supplied}}$$

$$\text{Heat supplied} = \frac{40}{0.602} = 66.5 \text{ kW}$$

$$= 66.5 \text{ kJ/s} \quad \underline{\underline{\text{Ans}}}$$

$$\text{Number of cycles/sec} = \frac{3000}{2 \times 60} = 25$$

Net work output per cycle from each cylinder

$$= \frac{40}{4 \times 25} = 0.4 \text{ kJ}$$

$$p_m = \frac{W}{V_s} = \frac{0.4 \times 1000}{300 \times 10^{-6}}$$

$$= 13.3 \times 10^5 \text{ N/m}^2 \quad \underline{\underline{\text{Ans}}}$$

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{\gamma-1}$$

$$= 300 \times 10^{0.4} = 753.6 \text{ K}$$

Heat supplied/cylinder/cycle (Q_{2-3})

$$= \frac{66.5}{4 \times 25} = 0.665 \text{ kJ}$$

Now,

$$Q_{2-3} = mC_v (T_3 - T_2)$$

$$v_1 = \frac{RT_1}{p_1} = \frac{287 \times 300}{1 \times 10^5}$$

$$= 0.861 \text{ m}^3/\text{kg}$$

This initial volume of air in the cylinder is

$$V_1 = V_2 + V_s = \left(\frac{V_1}{10} \right) + V_s$$

$$0.9V_1 = V_s$$

$$V_1 = \frac{V_s}{0.9} = \frac{300 \times 10^{-6}}{0.9}$$

$$= 333 \times 10^{-6} \text{ m}^3$$

$$m = \frac{V_1}{v_1} = \frac{333 \times 10^{-6}}{0.861}$$

$$= 0.387 \times 10^{-3} \text{ kg}$$

The temperature rise resulting from heat addition is

$$\begin{aligned}
 T_3 - T_2 &= \frac{Q_{2-3}}{m C_v} \\
 &= \frac{0.665}{0.387 \times 10^{-3} \times 0.717} \\
 &= 2396.6 \text{ K} \\
 T_3 &= T_2 + 2396.6 = 753.6 + 2396.6 \\
 &= 3150.2 \text{ K} = 2877^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

Now let us consider the Diesel cycle

T_2 is the same as in the previous case, i.e.

$$T_2 = 753.6$$

Heat supplied per cycle per cylinder is also same, i.e.

$$\begin{aligned}
 Q_{2-3'} &= 0.665 \text{ kJ} \\
 Q_{2-3'} &= m C_p (T_{3'} - T_2) \\
 T_{3'} - T_2 &= \frac{0.665}{0.387 \times 10^{-3}} \times \frac{1}{1.004} \\
 &= 1711.5 \text{ K} \\
 T_{3'} &= 1711.5 + 753.6 = 2465.1 \text{ K} \\
 &= 2192.1^\circ \text{ C} \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

$$\begin{aligned}
 \text{Cut-off ratio, } r_c &= \frac{V_3}{V_2} = \frac{T_{3'}}{T_2} \\
 &= \frac{2465.1}{753.6} = 3.27
 \end{aligned}$$

$$\begin{aligned}
 \text{Air-standard efficiency} &= 1 - \frac{1}{r^{\gamma-1}} \left[\frac{r_c^\gamma - 1}{\gamma (r_c - 1)} \right] \\
 &= 1 - \frac{1}{10^{0.4}} \times \left[\frac{3.27^{1.4} - 1}{1.4 \times (3.27 - 1)} \right] \\
 &= 1 - 0.398 \times 1.338 = 0.467 \\
 &= 46.7\% \quad \underline{\underline{\text{Ans}}}
 \end{aligned}$$

$$\begin{aligned}\text{Power output} &= \eta \times \text{total rate of heat added} \\ &= 0.467 \times 66.5 = 31.1 \text{ kW}\end{aligned}$$

$$\text{Power output/cylinder} = \frac{31.1}{4} = 7.76 \text{ kW}$$

Work done/cylinder/cycle

$$\begin{aligned}&= \frac{7.76}{25} = 0.3104 \text{ kJ} \\ p_m &= \frac{W}{V_s} = \frac{0.3104 \times 1000}{300 \times 10^{-6}} \\ &= 10.35 \times 10^5 \text{ N/m}^2 \quad \underline{\underline{\text{Ans}}}\end{aligned}$$

As discussed in the text, this problem illustrates that for the same compression ratio and heat input Otto cycle is more efficient.

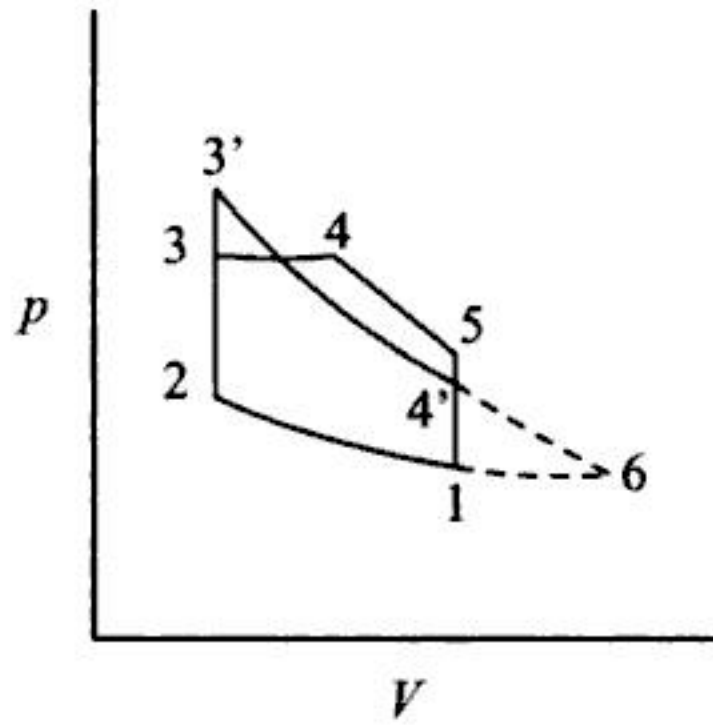
- 3.25 The compression ratio of an engine is 10 and the temperature and pressure at the start of compression is 37 °C and 1 bar. The compression and expansion processes are both isentropic and the heat is rejected at exhaust at constant volume. The amount of heat added during the cycle is 2730 kJ/kg. Determine the mean effective pressure and thermal efficiency of the cycle if (i) the maximum pressure is limited to 70 bar and heat is added at both constant volume and constant pressure and (ii) if all the heat is added at constant volume. In this case how much additional work per kg of charge would be obtained if it were possible to expand isentropically the exhaust gases to their original pressure of 1 bar. Assume that the charge has the same physical properties as that of air.

Solution

$$\begin{aligned}v_1 &= \frac{V_1}{m} = \frac{RT_1}{p_1} \\ &= \frac{287 \times 310}{1 \times 10^5} = 0.89 \text{ m}^3/\text{kg}\end{aligned}$$

Consider the process 1-2

$$\begin{aligned}\frac{T_2}{T_1} &= \left(\frac{V_1}{V_2}\right)^{\gamma-1} = 310 \times 10^{0.4} \\ &= 778.7 \text{ K}\end{aligned}$$



$$p_2 = p_1 \left(\frac{V_1}{V_2} \right)^\gamma = 25.12 \text{ bar}$$

Consider the limited pressure cycle (123451)

$$\begin{aligned} T_3 &= T_2 \frac{p_3}{p_2} \\ &= 778.7 \times \frac{70}{25.12} = 2170 \text{ K} \end{aligned}$$

Heat supplied at constant volume

$$\begin{aligned} &= 0.717 \times (2170 - 778.7) \\ &= 997.56 \text{ kJ/kg} \end{aligned}$$

Heat supplied at constant pressure

$$= 2730 - 997.56 = 1732.44 \text{ kJ/kg}$$

Now,

$$1732.4 = 1.004 \times (T_4 - 2170)$$

$$T_4 = 2170 + \frac{1732.4}{1.004} = 3895.5 \text{ K}$$

$$v_4 = v_3 \frac{T_4}{T_3}$$

$$= \frac{0.89}{10} \times \frac{3895.5}{2170}$$

$$= 0.16 \text{ m}^3/\text{kg}$$

$$\begin{aligned}
 T_5 &= T_4 \left(\frac{v_4}{v_5} \right)^{\gamma-1} \\
 &= 3895.5 \times \left(\frac{0.16}{0.89} \right)^{0.4} = 1961 \text{ K}
 \end{aligned}$$

$$\begin{aligned}
 \text{Heat rejected} &= C_v (T_5 - T_1) \\
 &= 0.717 \times (1961 - 310) = 1184 \text{ kJ/kg}
 \end{aligned}$$

$$\begin{aligned}
 \text{Work done} &= \text{Heat supplied} - \text{Heat rejected} \\
 &= 2730 - 1184 = 1546 \text{ kJ/kg}
 \end{aligned}$$

$$\eta = \frac{w}{q} = \frac{1546}{2730} = 0.57$$

$$= 57\% \quad \text{Ans} \quad \leftarrow$$

$$p_m = \frac{w}{v_1 - v_2} = \frac{w}{v_1 \left(1 - \frac{1}{r} \right)}$$

$$= \frac{1546 \times 10^3}{0.89 \times \left(1 - \frac{1}{10} \right)}$$

$$= 19.32 \times 10^5 \text{ N/m}^2 = 19.32 \text{ bar} \quad \text{Ans} \quad \leftarrow$$

Consider the constant-volume cycle (123'4'1)

$$q_{2-3'} = 2730 \text{ kJ/kg}$$

$$2730 = 0.717 \times (T_{3'} - 778)$$

$$T_{3'} = 4586 \text{ K}$$

$$p_{3'} = T_{3'} \left(\frac{p_2}{T_2} \right)$$

$$= 4586 \times \frac{25.12}{778.7} = 147.9 \text{ bar}$$

$$\frac{T_{3'}}{T_{4'}} = \left(\frac{V_{4'}}{V_{3'}} \right)^{\gamma-1}$$

$$T_{4'} = 4586 \times \left(\frac{1}{10} \right)^{0.4}$$

$$= 1852.7$$



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- 3.4 What is Carnot cycle and what is its importance? How is this cycle reversible?
- 3.5 Draw the Carnot cycle on p - V and T - s diagrams. Derive an expression for its efficiency. Comment on the significance of this result as it related to source and sink temperature.
- 3.6 Define mean effective pressure and comment its application in internal combustion engines.
- 3.7 Name the cycles which have the same efficiency of Carnot cycle. Are these cycles reversible in the sense the Carnot cycle is?
- 3.8 Draw the Stirling cycle on p - V and T - s diagrams and show how the cycle is reversible?
- 3.9 Derive the expression for Stirling cycle efficiency and show that the expression is same as that of Carnot cycle.
- 3.10 If you include the efficiency of the heat exchanger show how the expression is modified.
- 3.11 Draw the p - V and T - s diagram of Ericsson cycle and show how it is made reversible.
- 3.12 Compare Carnot, Stirling and Ericsson cycles operating between the same source and sink temperatures and with equal changes in specific volume.
- 3.13 Draw the Otto cycle on p - V and T - s diagrams mark the various processes.
- 3.14 Derive an expression for the efficiency of Otto cycle and comment on the effect of compression ratio on the efficiency with respect of ratio of specific heats by means of a suitable graph.
- 3.15 Obtain an expression for mean effective pressure of an Otto cycle.
- 3.16 What is the basic difference between an Otto cycle and Diesel cycle? Derive the expression for the efficiency and mean effective pressure of the Diesel cycle.
- 3.17 Show that the efficiency of the Diesel cycle is lower than that of Otto cycle for the same compression ratio. Comment why the higher efficiency of the Otto cycle compared to Diesel cycle for the same compression ratio is only of a academic interest and not practical importance.

- 3.18 Compare the Otto cycle for the same peak pressure and temperature. Illustrate the cycles on p - V and T - s diagrams.
- 3.19 Draw the p - V and T - s diagrams of a Dual cycle. Why this cycle is also called limited pressure or mixed cycle?
- 3.20 Derive the expressions for the efficiency and mean effective pressure of a Dual cycle.
- 3.21 Compare Otto, Diesel and Dual cycles for the
- (i) same compression ratio and heat input
 - (ii) same maximum pressure and heat input
 - (iii) same maximum pressure and temperature
 - (iv) same maximum pressure and work output
- 3.22 Sketch the Lenoir cycle on p - V and T - s diagrams and obtain an expression for its air-standard efficiency.
- 3.23 Compare the Otto cycle and Atkinson cycle. Derive the expression for the efficiency of Atkinson cycle.
- 3.24 Derive an expression for the air-standard efficiency of the Joule cycle in terms of
- (i) compression ratio
 - (ii) pressure ratio.
- 3.25 Where do the following cycles have applications
- (i) Otto cycle
 - (ii) Diesel cycle
 - (iii) Dual cycle
 - (iv) Stirling cycle
 - (v) Ericsson cycle
 - (vi) Atkinson cycle
 - (vii) Lenoir cycle
 - (viii) Joule cycle



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ideal thermal efficiency. Assume $C_p = 1.004 \text{ kJ/kg K}$ and $C_v = 0.717 \text{ kJ/kg K}$. *Ans:* 41.36%

- 3.21 The cycle of an internal combustion engine with isochoric heat supply is performed with the compression ratio equal to 8. Find heat supplied to the cycle and the useful work, if the removed heat is 500 kJ/kg and the working fluid is air.

Ans: (i) 1148.63 kJ/kg (ii) 648.63 kJ/kg

- 3.22 The initial parameters (at the beginning of compression) of the cycle of an internal combustion engine with isobaric heat supply are 0.1 MPa and 80°C . The compression ratio is 16 and the heat supplied is 850 kJ/kg . Calculate the parameters at the characteristic points of the cycle and the thermal efficiency, if the working fluid is air.

Ans: (i) $p_2 = 48.5 \text{ bar}$ (ii) $p_3 = 48.5 \text{ bar}$
 (iii) $p_4 = 2.26 \text{ bar}$ (iv) $T_2 = 1070.1 \text{ K}$
 (v) $T_3 = 1916 \text{ K}$ (vi) $T_4 = 797.73 \text{ K}$
 (vii) $\eta_{th} = 62.5\%$

- 3.23 The pressure ratio $\lambda = 1.5$ in the process of isochoric heat supply for the cycle of an internal combustion engine with a mixed supply of heat $= 1034 \text{ kJ/kg}$ and the compression ratio $= 13$. Find the thermal efficiency and temperature at the characteristic points of the cycle if the initial parameters are 0.09 MPa and 70°C and the working substance is air.

Ans: (i) $\eta_{th} = 57.5\%$ (ii) $T_2 = 956.9 \text{ K}$
 (iii) $T_3 = 1435.5 \text{ K}$ (iv) $T_4 = 2122.86 \text{ K}$
 (v) $T_5 = 890 \text{ K}$

- 3.24 The parameters of the initial state of one kilogram of air in the cycle of an internal combustion engine are 0.095 MPa and 65°C . The compression ratio is 11. Compare the values of the thermal efficiency for isobaric and isochoric heat supply in amounts of 800 kJ , assuming that $k = 1.4$.

Ans: $\eta_{tp} = 55.7\%$, $\eta_{tv} = 61.7\%$

- 3.25 Find the thermal efficiency of the cycle of an internal combustion engine with a mixed heat supply, if the minimum temperature of the cycle is 85°C and the maximum temperature is 1700 K . The compression ratio is 15 and the pressure ratio in the process of heat supply is 1.3. The working fluid is air.

Ans: $\eta_{th} = 65.25\%$



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close to the actual conditions than those used in the air-standard cycle analysis.

4.2 FUEL-AIR CYCLES AND THEIR SIGNIFICANCE

By air-standard cycle analysis, it is understood how the efficiency is improved by increasing the compression ratio. However, analysis cannot bring out the effect of air-fuel ratio on the thermal efficiency because the working medium was assumed to be air. In this chapter, the presence of fuel in the cylinder is taken into account and accordingly the working medium will be a mixture of fuel and air. By fuel-air cycle analysis it will be possible to bring out the effect of fuel-air ratio on thermal efficiency and also study how the peak pressures and temperatures during the cycle vary with respect to fuel-air ratio. In general, influence of many of the engine operating variables on the pressures and temperatures within the engine cylinder may be better understood by the examination of the fuel-air cycles. The fuel-air cycle analysis takes into account the following :

- (i) *The actual composition of the cylinder gases* : The cylinder gases contains fuel, air, water vapour and residual gas. The fuel-air ratio changes during the operation of the engine which changes the relative amounts of CO_2 , water vapour, etc.
- (ii) *The variation in the specific heat with temperature* : Specific heats increase with temperature except for mono-atomic gases. Therefore, the value of γ also changes with temperature.
- (iii) *The effect of dissociation* : The fuel and air do not completely combine chemically at high temperatures (above 1600 K) and this leads to the presence of CO , H_2 , H and O_2 at equilibrium conditions.
- (iv) *The variation in the number of molecules* : The number of molecules present after combustion depend upon fuel-air ratio and upon the pressure and temperature after the combustion.

Besides taking the above factors into consideration, the following assumptions are commonly made :

- (i) There is no chemical change in either fuel or air prior to combustion.
- (ii) Subsequent to combustion, the charge is always in chemical equilibrium.



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the value of C_v because of variable specific heat, which reduces the temperature as already explained.

The process, $2' \rightarrow 3'$ is heat addition with the variation in specific heat. From $3'$, if expansion takes place at constant specific heats, this would result in the process $3' \rightarrow 4''$ whereas actual expansion due to variable specific heat will result in $3' \rightarrow 4'$ and $4'$ is higher than $4''$. The magnitude in the difference between $4'$ and $4''$ is proportional to the reduction in the value of γ .

Consider the process $3'4''$

$$T_{4''} = T_{3'} \left(\frac{v_3}{v_4} \right)^{\gamma-1} \quad (4.7)$$

for the process $3'4'$

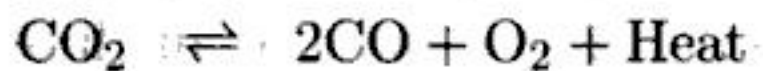
$$T_{4'} = T_{3'} \left(\frac{v_3}{v_4} \right)^{k-1} \quad (4.8)$$

Reduction in the value of k due to variable specific heat results in increase of temperature from $T_{4''}$ to $T_{4'}$.

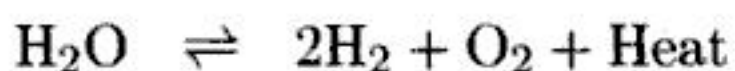
4.5 DISSOCIATION

Dissociation process can be considered as the disintegration of combustion products at high temperature. Dissociation can also be looked as the reverse process to combustion. During dissociation the heat is absorbed whereas during combustion the heat is liberated. In IC engines, mainly dissociation of CO_2 into CO and O_2 occurs, whereas there is a very little dissociation of H_2O .

The dissociation of CO_2 into CO and O_2 starts commencing around 1000°C and the reaction equation can be written as



Similarly, the dissociation of H_2O occurs at temperatures above 1300°C and is written as



The presence of CO and O_2 in the gases tends to prevent dissociation of CO_2 ; this is noticeable in a rich fuel mixture, which, by producing more CO , suppresses dissociation of CO_2 . On the other hand, there is no dissociation in the burnt gases of a lean fuel-air mixture. This is mainly due to the fact that the temperature produced is too low for this phenomenon to occur. Hence, the maximum



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- (i) character of the cycle (due to assumptions)
- (ii) equivalence ratio (actual $F/A \div$ stoichiometric F/A)
- (iii) chemical composition of the fuel

Figure 4.5 shows variation of efficiency with mixture strength of fuel-air cycle relative to that of air cycle showing the gain in efficiency as the mixture becomes leaner. It is seen from Fig.4.5 that the efficiency ratio (fuel-air cycle efficiency/air-standard cycle efficiency) increases as the mixture becomes leaner and leaner tending towards the air-standard cycle efficiency. It is to be noted that this, trend exists at all compression ratios.

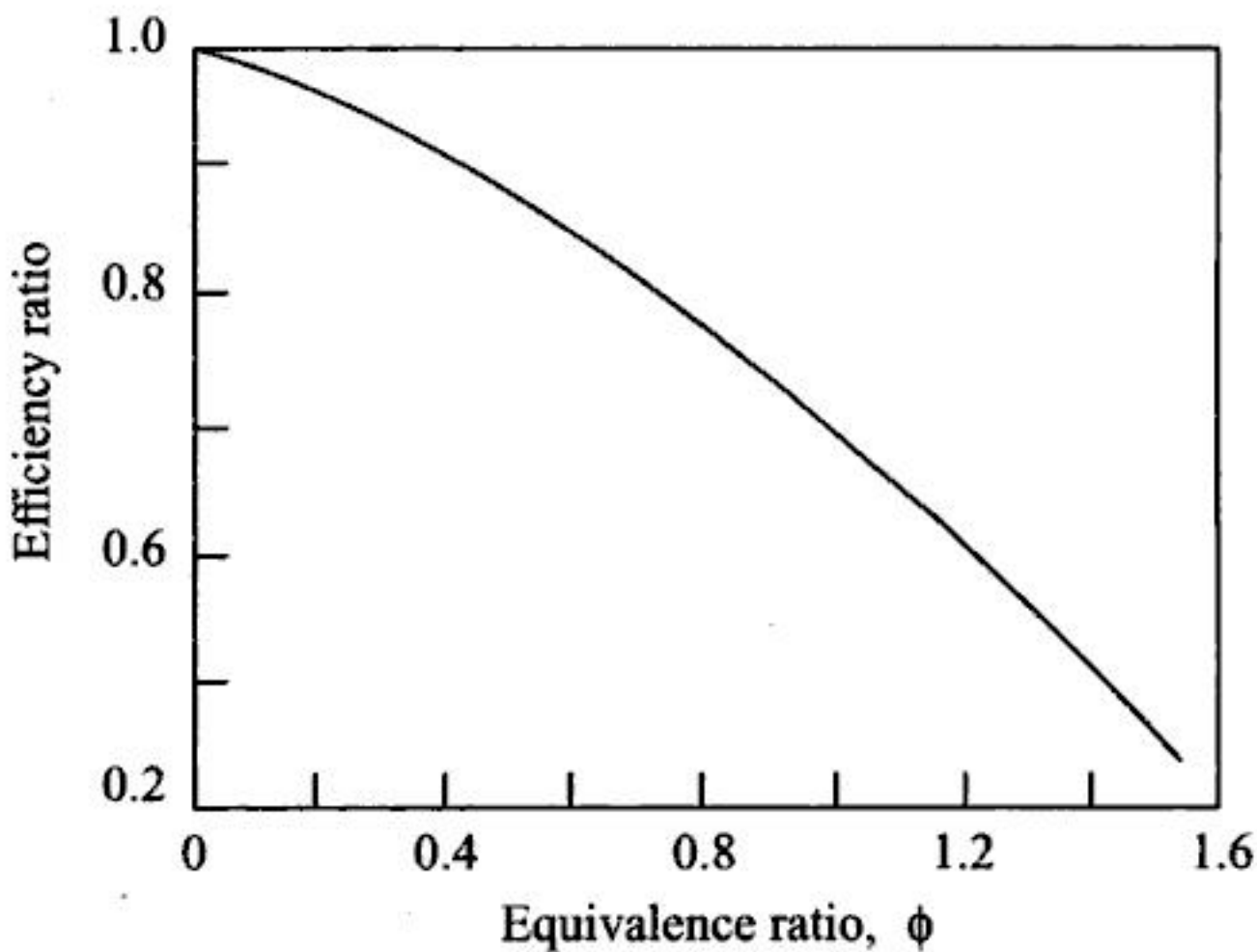


Fig. 4.5 Effect of Relative Fuel-Air ratio on Efficiency Ratio

At very low fuel-air ratio the mixture would tend to behave like a perfect gas with constant specific heat. Cycles with lean to very lean mixtures tend towards air-standard cycles. In such cycles the pressure and temperature rises. Some of the chemical reactions involved tend to be more complete as the pressure increases. These considerations apply to constant-volume as well as constant-pressure cycles.

The simple air-standard cycle analysis cannot predict the variation of thermal efficiency with mixture strength since air is assumed to be the working medium. However, fuel-air cycle analysis suggests that the thermal efficiency will deteriorate as the mixture supplied to an engine is enriched. This is explained by the increasing



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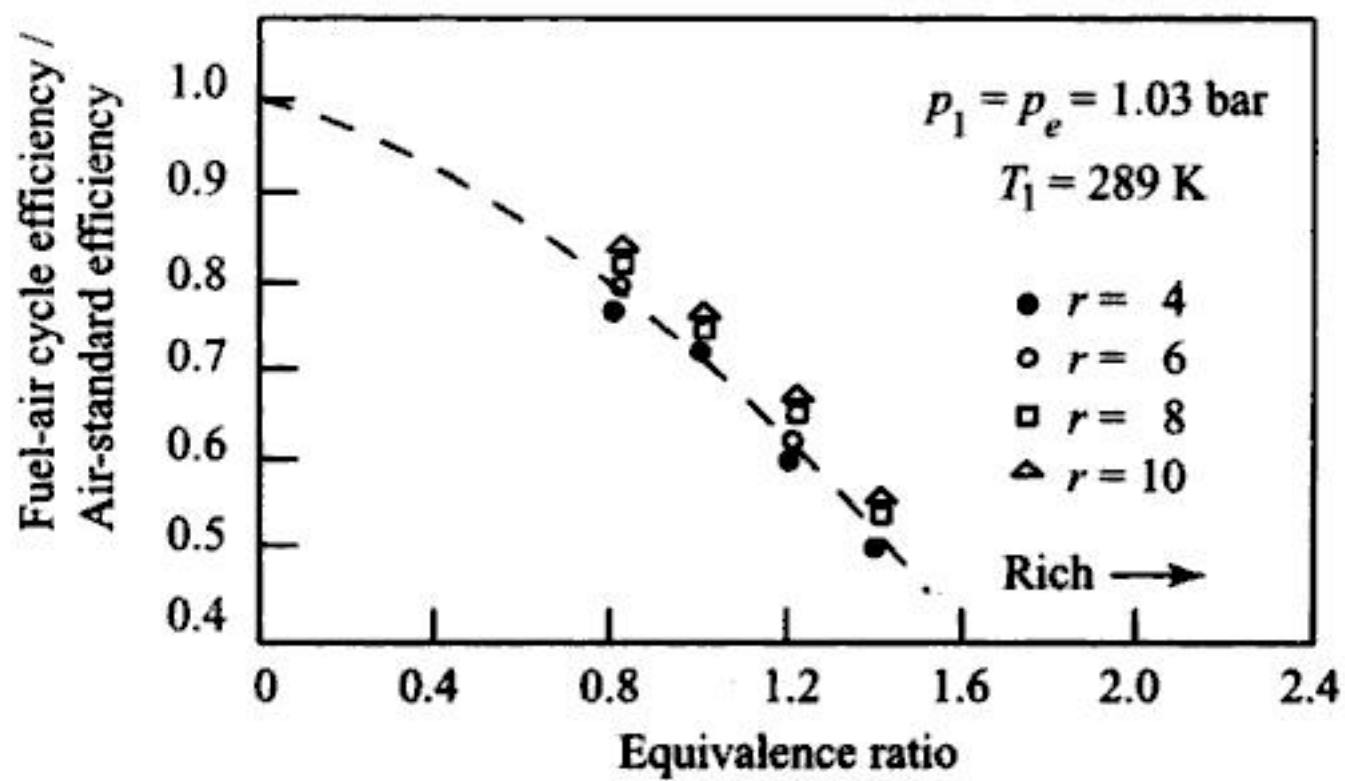


Fig. 4.9 Variation of Efficiency with Mixture Strength for a Constant Volume Fuel-Air Cycle

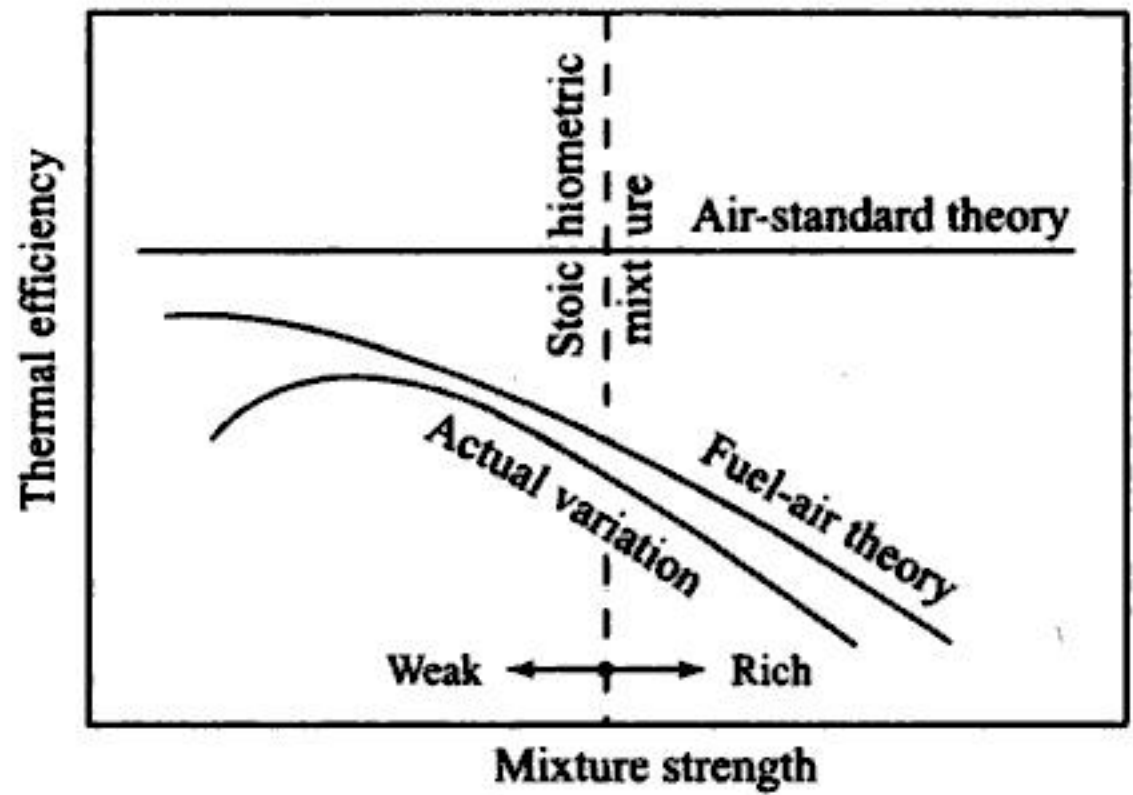


Fig. 4.10 Effect of Mixture Strength on Thermal Efficiency

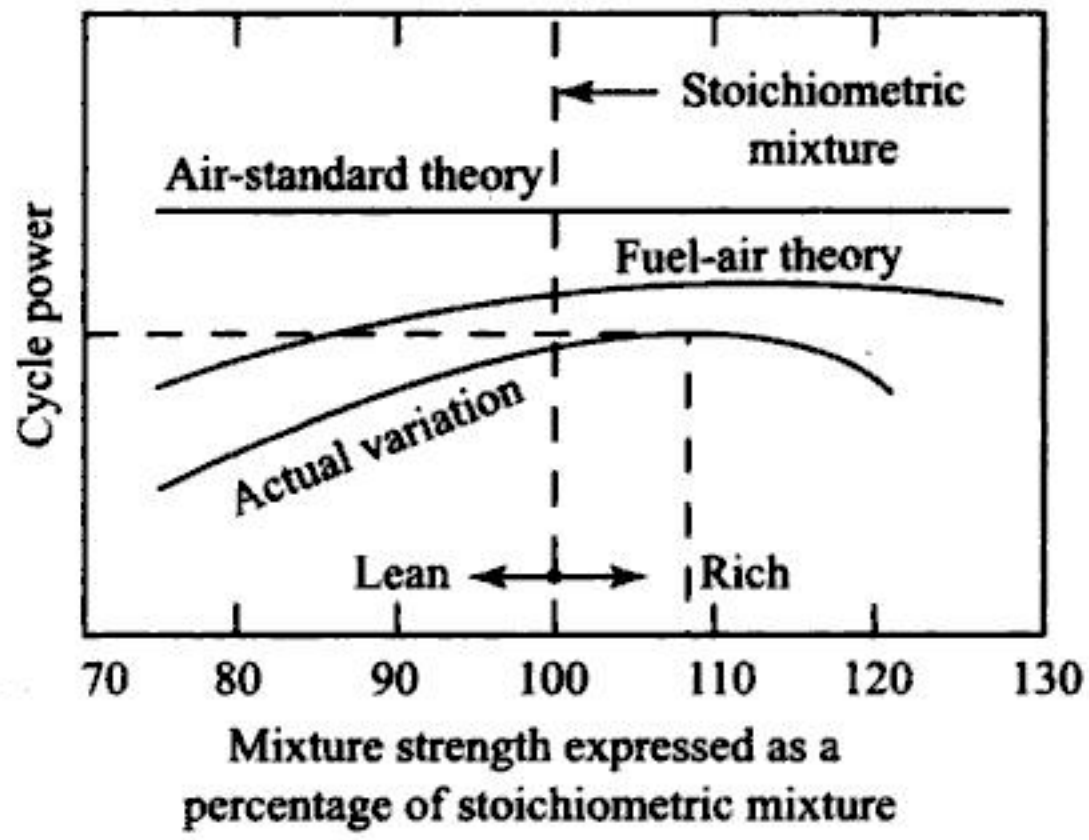


Fig. 4.11 Effect of Fuel-Air Ratio on Power



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Exercise

- 4.1 Find the percentage change in the efficiency of an Otto cycle having a compression ratio of 10, if C_v decreases by 2%.

Ans: 1.22%

- 4.2 Find the percentage increase in the efficiency of a Diesel cycle having a compression ratio of 16 and cut-off ratio is 10% of the swept volume, if C_v decreases by 2%. Take $C_v = 0.717$ and $\gamma = 1.4$.

Ans: 1.23%

- 4.3 The air-fuel ratio of a Diesel engine is 31:1. If the compression ratio is 15:1 and the temperature at the end of compression is 1000 K, find at what percentage of stroke is the combustion complete if the combustion begins at *TDC* and continuous at constant pressure. Calorific value of the fuel is 40000 kJ/kg. Assume the variable specific heat, $C_p = a + bT$, where $a = 1$ and $b = 0.28 \times 10^{-4}$.

Ans: 15.68%

- 4.4 An engine working on the Otto cycle, uses hexane (C_6H_{14}) as fuel. The engine works on chemically correct air-fuel ratio and the compression ratio is 8. Pressure and temperature at the beginning of compression are 1 bar and 77 °C respectively. If the calorific value of the fuel is 43000 kJ/kg and $C_v = 0.717$ kJ/kg K, find the maximum temperature and pressure of the cycle. Assume the compression follows the law $pV^{1.3} = c$.

Ans: (i) 4343.6 K (ii) 99.28 bar

- 4.5 Find the percentage change in efficiency of a dual cycle having compression ratio = 16 and cut-off ratio of 10% of swept volume and if C_v increases by 2%. Given $\frac{T_3}{T_2} = 1.67$.

Ans: 0.68%

- 4.6 It is estimated that for air operating in a given engine the γ decreases by 2% from its original value of 1.4. Find the change in efficiency. The pressure at the end of compression is 18 bar.

Ans: 4.5%



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formula. For example, isomerization is used for the conversion of n-butane into iso-butane for alkylation. Conversion of n-pentane and n-hexane into isoparaffins to improve knock rating of highly volatile gasoline is another example.

- (vi) *Cyclization* joins together the ends of a straight chain molecule to form a ring compound of the naphthene family.
- (vii) *Aromatization* is similar to cyclization, the exception being that the product is an aromatic compound.
- (viii) *Reformation* is a type of cracking process which is used to convert the low antiknock quality stocks into gasolines of higher octane rating (see section 6.6). It does not increase the total gasoline volume.
- (ix) *Blending* is a process of obtaining a product of desired quality by mixing certain products in some suitable proportion.

6.5 IMPORTANT QUALITIES OF ENGINE FUELS

Fuels used in IC engines should possess certain basic qualities which are important for the smooth running of the engines. In this section, the important qualities of fuels for both SI and CI engines are reviewed.

6.5.1 SI Engine Fuels

Gasoline which is mostly used in the present day SI engines is usually a blend of several low boiling paraffins, naphthenes and aromatics in varying proportions. Some of the important qualities of gasoline are discussed below.

- (i) *Volatility* : Volatility is one of the main characteristic properties of gasoline which determines its suitability for use in an SI engine. Since gasoline is a mixture of different hydrocarbons, volatility depends on the fractional composition of the fuel. The usual practice of measuring the fuel volatility is the distillation of the fuel in a special device at atmospheric pressure and in the presence of its own vapour. The fraction that boils off at a definite temperature is measured. The characteristic points are the temperatures at which 10, 40, 50 and 90% of the volume evaporates as well as the temperature at which boiling of the fuel terminates. Figure 6.2 shows the fractional distillation



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6.6.1 Rating of SI Engine Fuels

Resistance to knocking is an extremely important characteristic of fuel for spark-ignition engines. These fuels differ widely in their ability to resist knock depending on their chemical composition. A satisfactory rating method for comparing the antiknock qualities of the various fuels has been established. In addition to the chemical characteristics of hydrocarbons in the fuel, other operating parameters such as fuel-air ratio, ignition timing, dilution, engine speed, shape of the combustion chamber, ambient conditions, compression ratio etc. affect the tendency to knock in the engine cylinder. Therefore, in order to determine the knock resistance characteristic of the fuel, the engine and its operating variables must be fixed at standard values.

According to a standard practice, the antiknock value of an SI engine fuel is determined by comparing its antiknock property with a mixture of two reference fuels, iso-octane (C_8H_{18}) and normal heptane (C_7H_{16}). Iso-octane chemically being a very good antiknock fuel, is arbitrarily assigned a rating of 100 octane number. Normal heptane (C_7H_{16}), on the other hand, has very poor antiknock qualities and is given a rating of 0 octane number. *The Octane number fuel is defined as the percentage, by volume, of iso-octane in a mixture of iso-octane and normal heptane, which exactly matches the knocking intensity of the fuel in a standard engine under a set of standard operating conditions.*

The addition of certain compounds (e.g. tetraethyl lead) to iso-octane produces fuels of greater antiknock quality (above 100 octane number). The antiknock effectiveness of tetraethyl lead, for the same quantity of lead added, decreases as the total content of lead in the fuel increases. Furthermore, each octane number at the higher range of the octane scale will produce greater antiknock effect compared to the same unit at the lower end of the scale. For instance, octane number increase from 92 to 93 produce greater antiknock effect than a similar increase from 32 to 33 octane number. Because of this non-linear variation, a new scale was derived which expresses the approximate relative engine performance and the units of this scale are known as the Performance Numbers, PN . Octane numbers, ON above 100 can be computed by

$$ON(> 100) = 100 + \frac{28.28A}{1.0 + 0.736A + \sqrt{1.0 + 0.736A - 0.035216A^2}}$$

where A is TEL in ml/gal of fuel,



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which control the larger oil fields. As of now many alternate fuels have been used in limited quantities in automobiles. Quite often, fleet vehicles have been used for testing (e.g., taxies, delivery vans, utility company trucks). This allows for comparison with similar gasoline-fuelled vehicles, and simplifies fuelling of these vehicles.

The engines used for alternate fuels are modified engines which were originally designed for gasoline fuelling. They are, therefore, not the optimum design for the other fuels. Only when extensive research and development is done over a period of years will maximum performance and efficiency be realized from these engines. However, the research and development is difficult to justify until the fuels are accepted as viable for large numbers of engines.

Some diesel engines have started appearing on the market which use dual fuel. They use methanol or natural gas and a small amount of diesel fuel that is injected at the proper time to ignite both fuels. Most alternate fuels are very costly at present. This is often because of the quantity used. Many of these fuels will cost much less if the amount of their usage gets to the same order of magnitude as gasoline. The cost of manufacturing, distribution, and marketing all would be less.

Another problem with alternate fuels is the lack of distribution points (service stations) where the fuel is available to the public. The public will be reluctant to purchase an automobile unless there is a large-scale network of service stations available where fuel for that automobile can be purchased. On the other hand, it is difficult to justify building a network of these service stations until there are enough automobiles to make them profitable. Some cities are starting to make available a few distribution points for some of these fuels, like propane, natural gas, LPG and methanol. The transfer from one major fuel type to another will be a slow, costly, and sometimes painful process. In the following section we will discuss the various alternate fuels.

7.2 POSSIBLE ALTERNATIVES

Fuels are classified into three forms, viz. solid, liquid and gaseous. Present day automobiles are using mainly liquid fuels of petroleum origin. However, some use gaseous fuels like CNG and LPG. It is interesting to note that during early days even solid fuels like coal, slurry and charcoal have been tried. In the following sections we will discuss in detail the various alternate fuels for IC engines. Table 7.1 gives the properties of various fuels.



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7.7 GASEOUS FUELS

Gaseous fuels are best suited for IC engines since physical delay is almost nil. However, as fuel displaces equal amount of air the engines may have poor volumetric efficiency. There are quite few gaseous fuels that can be used as alternate fuels. We will discuss them in details in the following sections.

7.7.1 Hydrogen

A number of automobile manufacturers have built with prototype or modified engines which operate on hydrogen fuel. The advantages of hydrogen as an IC engine fuel include:

- Low emissions. Essentially no CO or HC in the exhaust as there is no carbon in the fuel. Most exhaust would be H_2O and N_2 .
- Fuel availability. There are a number of different ways of making hydrogen, including electrolysis of water.
- Fuel leakage to environment is not a pollutant.
- High energy content per volume when stored as a liquid. This would give a large vehicle range for a given fuel tank capacity, but see the following.

The disadvantages of using hydrogen as a fuel include:

- Requirement of heavy, bulky fuel storage both in vehicle and at the service station. Hydrogen can be stored either as a cryogenic liquid or as a compressed gas. If stored as a liquid, it would have to be kept under pressure at a very low temperature. This would require a thermally super-insulated fuel tank. Storing in a gas phase would require a high pressure vessel with limited capacity.
- Difficult to refuel.
- Poor engine volumetric efficiency. Any time a gaseous fuel is used in an engine, the fuel will displace some of the inlet air and poorer volumetric efficiency will result.
- Fuel cost would be high at present-day technology and availability.



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Since modern engines are of high speed type, the time available for mixture formation is very limited. For example, an engine running at 3000 rpm has only about 10 milliseconds (ms) for mixture induction during intake stroke. When the speed becomes 6000 rpm the time available is only 5 ms. Therefore, in order to have high quality carburetion (that is mixture with high vapour content) the velocity of the air stream at the point where the fuel is injected has to be increased. This is achieved by introducing a venturi section in the path of the air. The fuel is discharged from the main metering jet at the minimum cross section of the venturi (called throat).

Other factors which ensure high quality carburetion within a short period are the presence of highly volatile hydrocarbons in the fuel. Therefore, suitable evaporation characteristics of the fuel, indicated by its distillation curve, are necessary for efficient carburetion especially at high engine speeds.

The temperature and pressure of surrounding air has a large influence on efficient carburetion. Higher atmospheric air temperature increases the vaporization of fuel (percentage of fuel vapour increases with increase in mixture temperature) and produces a more homogeneous mixture. An increase in atmospheric temperature, however, leads to a decrease in power output of the engine when the air-fuel ratio is constant due to reduced mass flow into the cylinder or, in other words, reduced volumetric efficiency.

The design of the carburetor, the intake system and the combustion chamber have considerable influence on uniform distribution of mixture to the various cylinders of the engine. Proper design of carburetor elements alone ensures the supply of desired composition of the mixture under different operating conditions of the engine.

8.4 AIR-FUEL MIXTURES

An engine is generally operated at different loads and speeds. For this, proper air-fuel mixture should be supplied to the engine cylinder. Fuel and air are mixed to form three different types of mixtures.

- (i) chemically correct mixture
- (ii) rich mixture and
- (iii) lean mixture

Chemically correct or stoichiometric mixture is one in which there is just enough air for complete combustion of the fuel. For example,



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less or more. As the throttle opening is varied, the air flow varies and creates a certain pressure differential between the float chamber and the venturi throat. The same pressure differential regulates the flow of fuel through the nozzle. Therefore, the velocity of flow of air and fuel vary in a similar manner. At the same time, the density of air decreases as the pressure at the venturi throat decreases with increasing air flow whereas that of the fuel remains unchanged. This results in a simple carburetor producing a progressively rich mixture with increasing throttle opening. The mathematical analysis of the performance of a simple carburetor is given in the next section.

8.9 CALCULATION OF THE AIR-FUEL RATIO

A simple carburetor with the tip of the fuel nozzle h metres above the fuel level in the float chamber is shown in Fig.8.7. It may be noted that the density of air is not the same at the inlet to the carburetor (section A-A, point 1) and the venturi throat (section B-B, point 2). The calculation of exact air mass flow involves taking this change in density or compressibility of air into account.

Applying the steady flow energy equation to sections A-A and B-B and assuming unit mass flow of air, we have,

$$q - w = (h_2 - h_1) + \frac{1}{2}(C_2^2 - C_1^2) \quad (8.1)$$

Here q , w are the heat and work transfers from entrance to throat and h and C stand for enthalpy and velocity respectively.

Assuming an adiabatic flow, we get $q = 0$, $w = 0$ and $C_1 \approx 0$,

$$C_2 = \sqrt{2(h_1 - h_2)} \quad (8.2)$$

Assuming air to behave like ideal gas, we get $h = C_p T$. Hence, Eq.8.2 can be written as,

$$C_2 = \sqrt{2C_p(T_1 - T_2)} \quad (8.3)$$

As the flow process from inlet to the venturi throat can be considered to be isentropic, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \quad (8.4)$$

$$T_1 - T_2 = T_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \right] \quad (8.5)$$



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The calibrated jets have a stamped number which gives the flow in ml/min under a head of 500 mm of pure benzol.

For a venturi of 30 to 35 mm size (having a jet size which is one sixteenth of venturi size) the pressure difference ($p_1 - p_2$) is about 50 mm of Hg. The velocity at throat is about 90 - 100 m/s and the coefficient of discharge for venturi C_{da} is usually 0.85.

8.10 ESSENTIAL PARTS OF A CARBURETOR

A carburetor consists essentially of the following parts, viz.

- (i) fuel strainer
- (ii) float chamber
- (iii) main fuel metering and idling nozzles
- (iv) choke and throttle

The various parts mentioned above are discussed briefly in the following section.

8.10.1 The Fuel Strainer

As the gasoline has to pass through a narrow nozzle exit there is every possibility that the nozzle may get clogged during prolonged operation of the engine. To prevent possible blockage of the nozzle by dust particles, the gasoline is filtered by installing a fuel strainer at the inlet to the float chamber (Fig.8.8). The strainer consists of a fine wire mesh or other type of filtering device, cone shaped or cylindrical shaped. The strainer is usually removable so that it can be taken out and cleaned thoroughly. It is retained in its seat by a strainer plug or a compression spring.

8.10.2 The Float Chamber

The function of a float chamber in a carburetor is to supply the fuel to the nozzle at a constant pressure head. This is possible by maintaining a constant level of the fuel in the float bowl. The float in a carburetor is designed to control the level of fuel in the float chamber. This fuel level must be maintained slightly below the discharge nozzle outlet holes in order to provide the correct amount of fuel flow and to prevent leakage of fuel from the nozzle when the engine is not operating. The arrangement of a float mechanism in relation to the



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increases and the cruising range of operation is established. The desired fuel-air ratio for idling can be regulated by idling adjustment shown in Fig.8.10.

Hot Idling Compensator: Some modern automobiles have this system in the carburetor unit. Under certain extremely hot operating conditions (with increased engine room temperature and also a high carburetor body temperature) there is a tendency for the idling mixture to become too rich. This causes idling instability. The hot idling compensator system (HIC) incorporates bi-metallic valve which admits air directly into the manifold in correct quantity when needed. Thus the mixture richness is adjusted and stable idling is ensured.

8.10.4 The Choke and the Throttle

When the vehicle is kept stationary for a long period during cool winter seasons, may be overnight, starting becomes more difficult. As already explained, at low cranking speeds and intake temperatures a very rich mixture is required to initiate combustion. Some times air-fuel ratio as rich as 9:1 is required. The main reason is that very large fraction of the fuel may remain as liquid suspended in air even in the cylinder. For initiating combustion, fuel-vapour and air in the form of mixture at a ratio that can sustain combustion is required. It may be noted that at very low temperature vapour fraction of the fuel is also very small and this forms combustible mixture to initiate combustion. Hence, a very rich mixture must be supplied. The most popular method of providing such mixture is by the use of choke valve. This is simple butterfly valve located between the entrance to the carburetor and the venturi throat as shown in Fig.8.11. When the choke is partly closed, large pressure drop occurs at the venturi throat that would normally result from the quantity of air passing through the venturi throat. The very large depression at the throat inducts large amount of fuel from the main nozzle and provides a very rich mixture so that the ratio of the evaporated fuel to air in the cylinder is within the combustible limits. Sometimes, the choke valves are spring loaded to ensure that large carburetor depression and excessive choking does not persist after the engine has started, and reached a desired speed. This choke can be made to operate automatically by means of a thermostat so that the choke is closed when engine is cold and goes out of operation when engine warms up after starting. The speed and the output of an engine is controlled by the use of the throttle valve, which is located on the downstream



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the throttle valve. The plunger moves into the cylinder and forces an additional jet of fuel at the venturi throat. When the throttle is partly open, the spring sets the plunger back. There is also an arrangement which ensures that fuel in the pump cylinder is not forced through the jet when valve is slowly opened or leaks past the plunger or some holes into the float chamber.

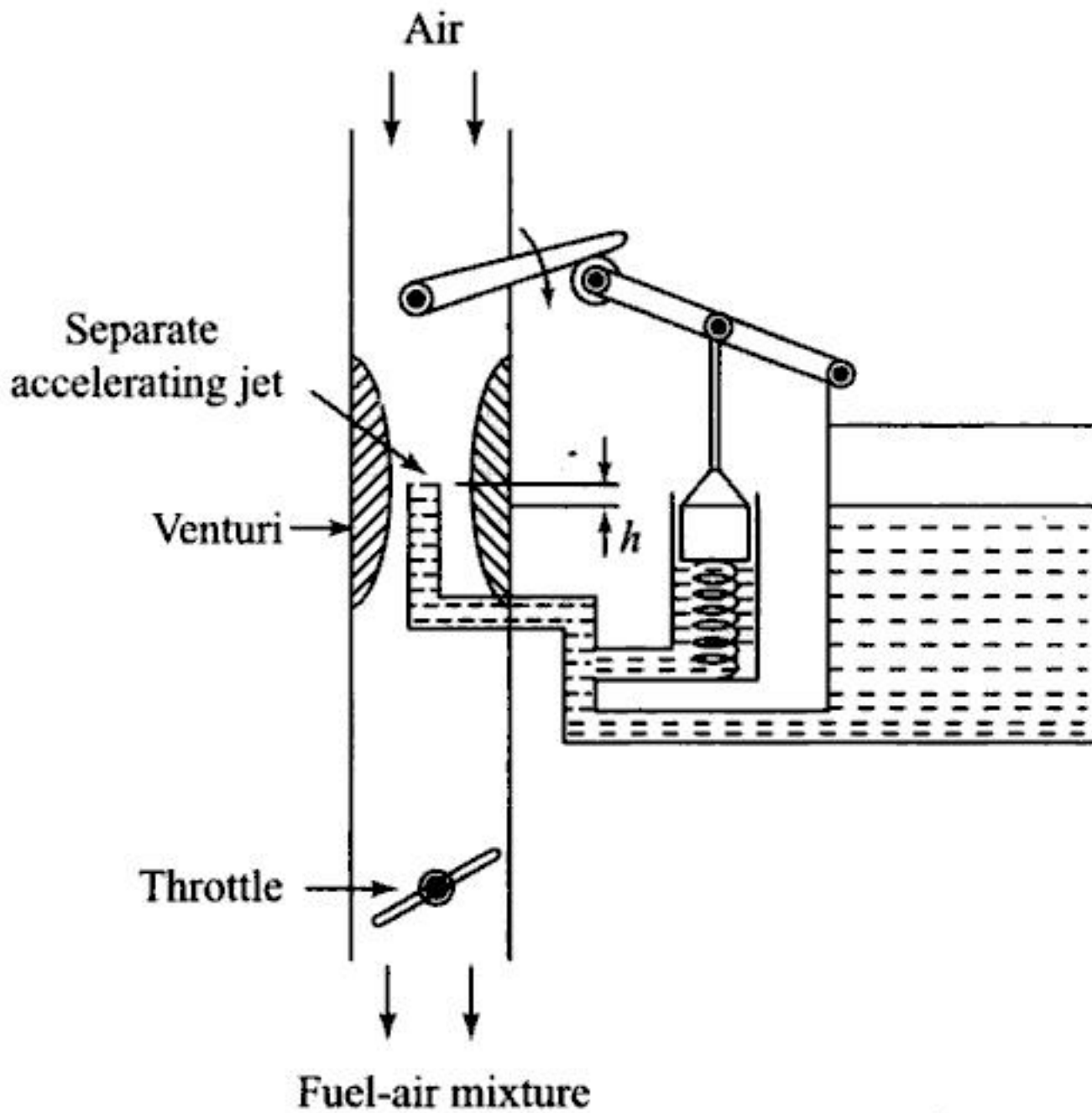


Fig. 8.19 Acceleration Pump System

Mechanical linkage system, in some carburetor, is substituted by an arrangement where by the pump plunger is held up by manifold vacuum. When this vacuum is decreased by rapid opening of the throttle, a spring forces the plunger down pumping the fuel through the jet.

8.12.4 Economizer or Power Enrichment System

At the maximum power range of operation from 80% to 100% load, richer air-fuel ratio of about 12 to 14 is required and at the maximum power, an air-fuel ratio of approximately 12 is expected. An economizer is a valve which remains closed at normal cruise operation and gets opened to supply rich mixture at full throttle operation. It reg-



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Table 9.1 Comparison of Various Fuel-Injection Systems

Job	Air injection system	Solid injection system		
		Individual pump	Common rail	Distributor
Metering	Pump	Pump	Injection valve	Pump
Timing	Fuel cam	Pump cam	Fuel cam	Fuel cam
Injection rate	Spray valve	Pump cam	Spray valve	Fuel cam
Atomization	Spray valve	Spray tip	Spray tip	Spray tip
Distribution	Spray valve	Spray tip	Spray tip	Spray tip

9.4 FUEL FEED PUMP

A schematic sketch of fuel feed pump is shown in Fig.9.4. It is of spring loaded plunger type. The plunger is actuated through a push rod from the cam shaft.

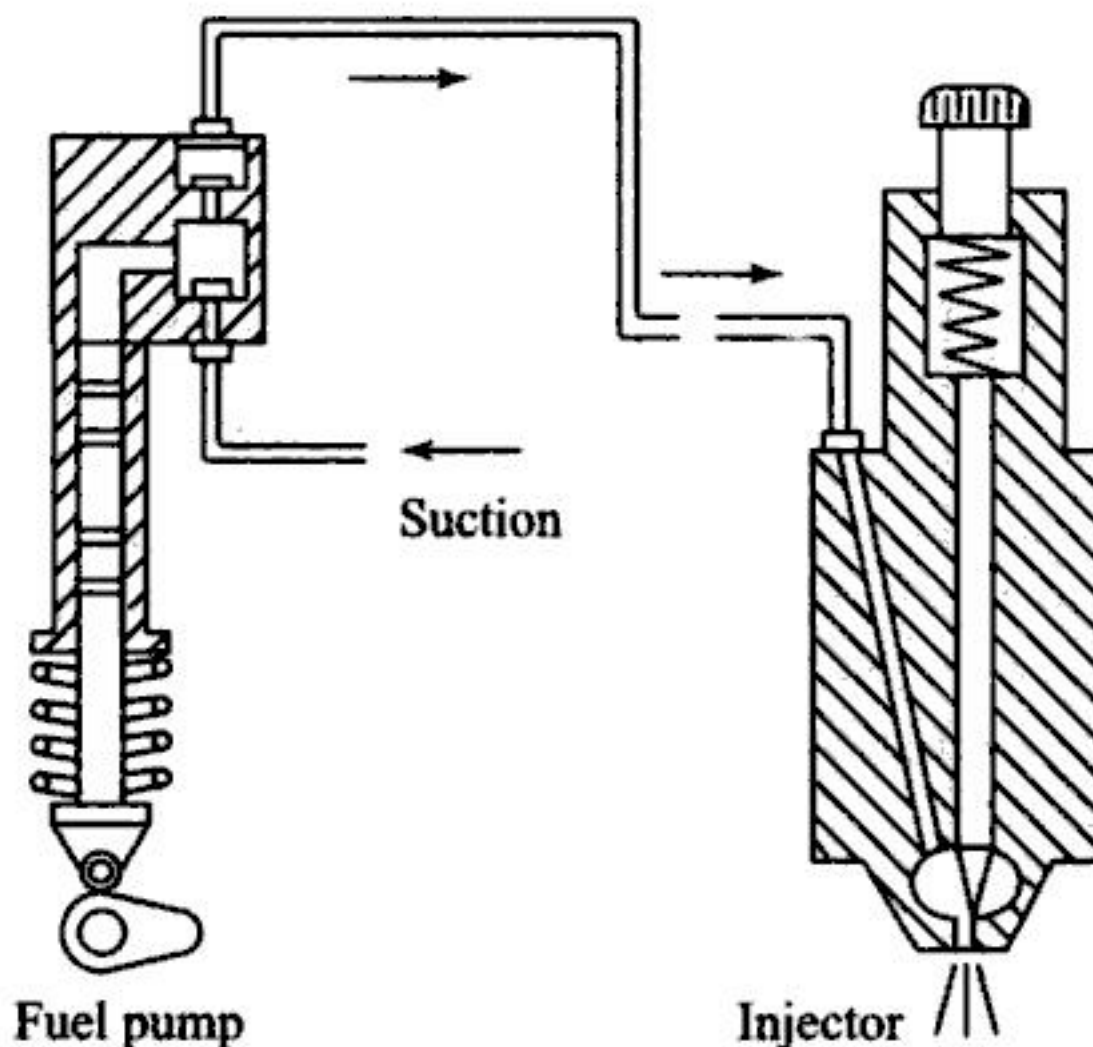


Fig. 9.4 Schematic Diagram of Fuel Feed Pump

At the minimum lift position of the cam the spring force on the plunger creates a suction which causes fuel flow from the main tank into the pump. When the cam is turned to its maximum lift position, the plunger is lifted upwards. At the same time the inlet valve is closed and the fuel is forced through the outlet valve. When the



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11

IGNITION

11.1 INTRODUCTION

In spark-ignition engines, as compression ratio is lower, and the self-ignition temperature of gasoline is higher, for igniting the mixture for the initiation of combustion an ignition system is a must.

The electrical discharge produced between the two electrodes of a spark plug by the ignition system starts the combustion process in a spark-ignition engine. This takes place close to the end of the compression stroke. The high temperature plasma kernel created by the spark, develops into a self sustaining and propagating flame front. In this thin reaction sheet certain exothermic chemical reactions occur. The function of the ignition system is to initiate this flame propagation process. It must be noted that the spark is to be produced in a repeatable manner viz., cycle-by-cycle, over the full range of load and speed of the engine at the appropriate moment in the engine cycle.

By implication, ignition is merely a prerequisite for combustion. Therefore, the study of ignition is a must to understand the phenomenon of combustion so that a criterion may be established to decide whether ignition has occurred. Although the ignition process is intimately connected with the initiation of combustion, it is not associated with the gross behaviour of combustion. Instead, it is a local small-scale phenomenon that takes place within a small zone in the combustion chamber.

In terms of its simplest definition, ignition has no degree, intensively or extensively. Either the combustion of the medium is initiated or it is not. Therefore, it is reasonable to consider ignition from the standpoint of the beginning of the combustion process that it initiates.



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is caused by heat transfer and diffusion of burning fuel molecules from the combustion zone to the adjacent layers of unburnt mixture. The flame front is a narrow zone separating the fresh mixture from the combustion products. The velocity with which the flame front moves, with respect to the unburned mixture in a direction normal to its surface is called the normal flame velocity.

In a homogeneous mixture with an equivalence ratio, ϕ , (*the ratio of the actual fuel-air ratio to the stoichiometric fuel-air ratio*) close to 1.0, the flame speed is normally of the order of 40 cm/s. However, in a spark-ignition engine the maximum flame speed is obtained when ϕ is between 1.1 and 1.2, i.e., when the mixture is slightly richer than stoichiometric.

If the equivalence ratio is outside this range the flame speed drops rapidly to a low value. When the flame speed drops to a very low value, the heat loss from the combustion zone becomes equal to the amount of heat-release due to combustion and the flame gets extinguished. Therefore, it is quite preferable to operate the engine within an equivalence ratio of 1.1 to 1.2 for proper combustion. However, by introducing turbulence and incorporating proper air movement, the flame speed can be increased in mixtures outside the above range.

12.3 HETEROGENEOUS MIXTURE

In a heterogeneous gas mixture, the rate of combustion is determined by the velocity of mutual diffusion of fuel vapours and air and the rate of chemical reaction is of minor importance. Self-ignition or spontaneous ignition of fuel-air mixture, at the high temperature developed due to higher compression ratios, is of primary importance in determining the combustion characteristics.

When the mixture is heterogeneous the combustion can take place in an overall lean mixture since, there are always local zones where ϕ varies between 1.0 and 1.2 corresponding to maximum rate of chemical reaction. Ignition starts in this zone and the flame produced helps to burn the fuel in the adjoining zones where the mixture is leaner. Similarly, in the zones where the mixture is rich the combustion occurs because of the high temperature produced due to combustion initiated in the zones where ϕ is 1.0 to 1.2.

A comprehensive study of combustion in both spark-ignition and compression-ignition engines is given in the following sections.



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knock. These are the temperature, pressure, density of the unburned charge and the time factors. Since, the effect of temperature, pressure and density are closely interrelated, these three are consolidated into one group and the time factors into another group.

12.11.1 Density Factors

Any factor in the design or operation of an engine which tends to reduce the temperature of the unburned charge should reduce the possibility of knocking by reducing the temperature of the end charge for autoignition. Similarly, any factor which reduces the density of the charge tends to reduce knocking by providing lower energy release. Further, the effect of the following parameters which are directly or indirectly connected with temperature, pressure and density factors on the possibility of knocking is discussed below.

Compression Ratio: Compression ratio of an engine is an important factor which determines both the pressure and temperature at the beginning of the combustion process. Increase in compression ratio increases the pressure and temperature of the gases at the end of the compression stroke. This decreases the ignition lag of the end gas and thereby increasing the tendency for knocking. The overall increase in the density of the charge due to higher compression ratio increases the preflame reactions in the end charge thereby increasing the knocking tendency of the engine. The increase in the knocking tendency of the engine with increasing compression ratio is the main reason for limiting the compression ratio to a lower value.

Mass of Inducted Charge: A reduction in the mass of the inducted charge into the cylinder of an engine by throttling or by reducing the amount of supercharging reduces both temperature and density of the charge at the time of ignition. This decreases the tendency of knocking.

Inlet Temperature of the Mixture: Increase in the inlet temperature of the mixture makes the compression temperature higher thereby, increasing the tendency of knocking. Further, volumetric efficiency will be lowered. Hence, a lower inlet temperature is always preferable to reduce knocking. It is important that the temperature should not be so low as to cause starting and vaporization problems in the engine.

Temperature of the Combustion Chamber Walls: Temperature of the combustion chamber walls play a predominant role in knocking. In order to prevent knocking the hot spots in the combustion chamber should be avoided. Since, the spark plug and exhaust



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16.6.1 Air Box Method

The orifice method can be used if pressure pulsations could be damped out by some means. The usual method of damping out pulsations is to fit an air box of suitable volume (500 to 600 times the swept volume in single cylinder engines and less in the case of multi-cylinder engines) to the engine with an orifice placed in the side of the box remote from the engine (Fig.16.19).

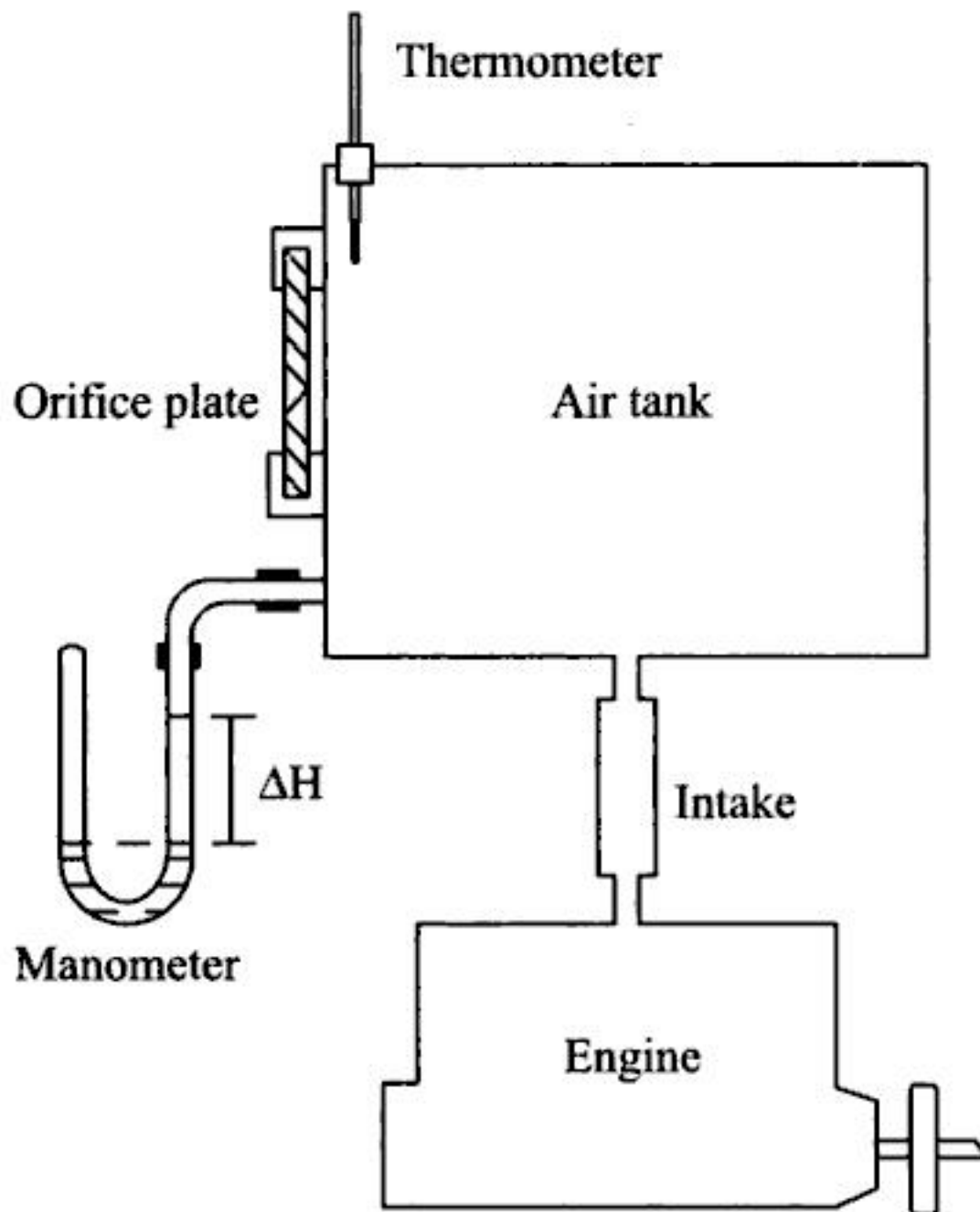


Fig. 16.19 Measurement of Air by Air Box Method

16.6.2 Viscous-Flow Air Meter

The use (Fig.16.20) of viscous flow airmeter gives accurate reading for pulsating flows. This meter uses an element where viscous resistance is the principal source of pressure loss and kinetic effects are small. This gives a linear relationship between pressure difference and flow instead of a square-law. From this it follows that a true mean-flow indication is obtained under pulsating flow conditions.

The viscous element is in the form of a honeycomb passage (a very large number of passages, Reynolds number being less than 200). The passages are triangular of the size approximately $0.5 \times 0.5 \times 75$ mm.



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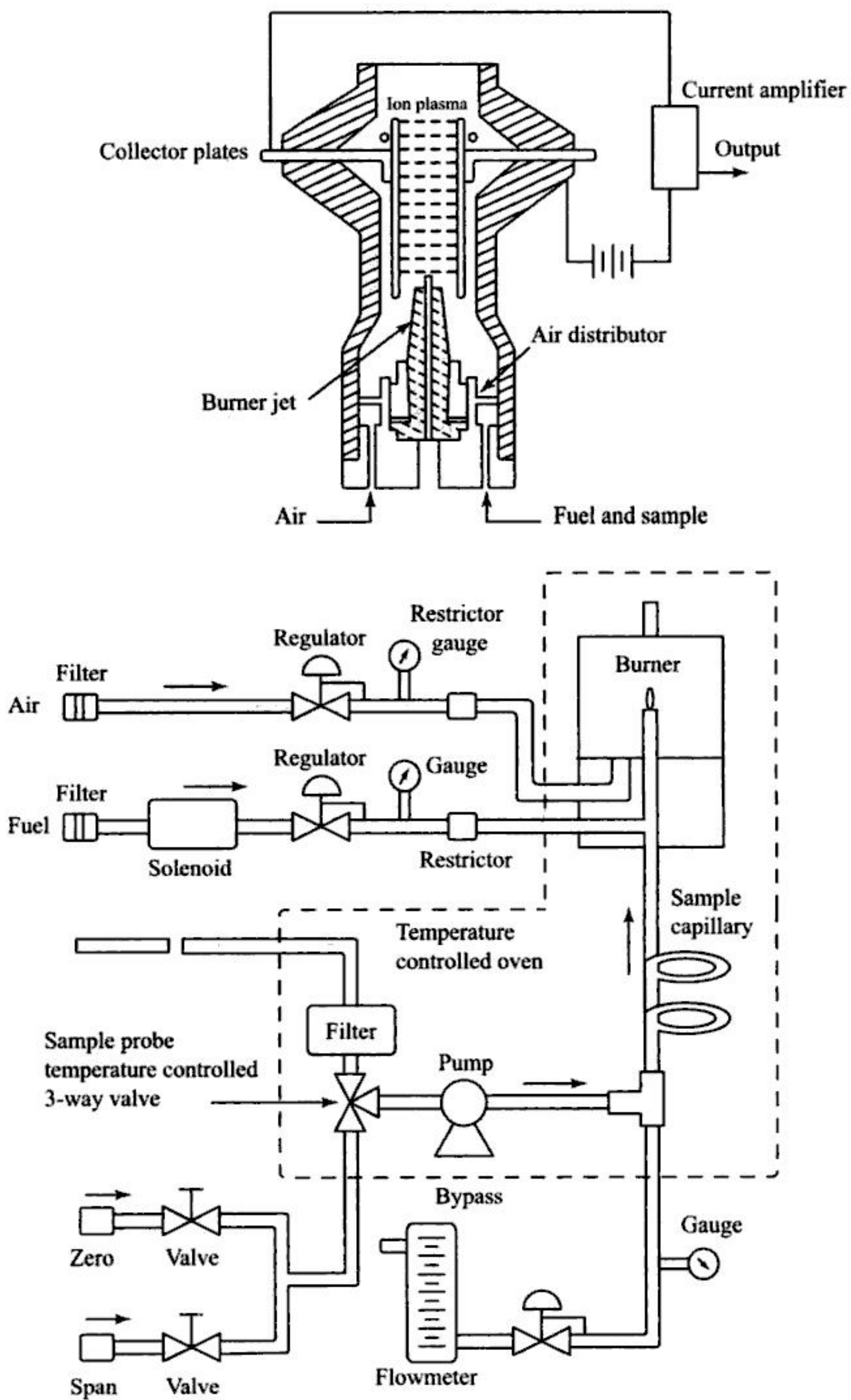


Fig. 16.23 Flame Ionization Method for Measuring Unburnt Hydrocarbons



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displacement volume, V_s , gives the same indicated net work as is actually produced with the varying pressures.

$$p_{im} \times (V_1 - V_2) = \text{Net work of cycle} \quad (17.3)$$

$$p_{im} = \frac{\text{Net work of cycle}}{V_1 - V_2} \quad (17.4)$$

$$= \frac{\text{Area of the indicator diagram}}{\text{Length of the indicator diagram}} \quad (17.5)$$

On an actual engine, the p - V diagram (called the *indicator diagram*) is obtained by a mechanical or electrical instrument attached to the cylinder. The area enclosed by the actual cycle on the indicator card may be measured by a planimeter. The value of the area measured, when divided by the piston displacement, results in the mean ordinate, or indicated mean effective pressure, p_{im} .

17.2.2 Indicated Power (ip)

Power is defined as the rate of doing work. In the analysis of cycles the net work is expressed in kJ/kg of air. This may be converted to power by multiplying by the mass flow rate of air through the engine in kg per unit time. Since, the net work obtained from the p - V diagram is the net work produced in the cylinder as measured by an indicator diagram, the power based there on is termed indicated power, ip .

$$ip = \dot{m}_a \times \text{net work} \quad (17.6)$$

where $\dot{m}_a$ is in kg/s, network is in kJ/kg of air and ip is in kW.

In working with actual engines, it is often desirable to compute ip from a given p_{im} and given engine operating conditions. The necessary formula may be developed from the equation of net work based on the mean effective pressure and piston displacement. From Eq.17.3,

$$\text{Indicated net work/cycle} = p_{im} V_s \quad (17.7)$$

By definition,

$$\text{Indicated power} = \text{Indicated net work} \times \text{cycles/s}$$

$$\begin{aligned} ip &= \frac{p_{im} V_s n K}{1000 \times 60} \\ &= \frac{p_{im} L A n K}{60000} \text{ kW} \end{aligned} \quad (17.8)$$



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17.3.7 Charge Efficiency

The charge efficiency shows how well the piston displacement of a four-stroke engine is utilized. Various factors affecting charge efficiency are:

- (i) the compression ratio.
- (ii) the amount of heat picked up during passage of the charge through intake manifold.
- (iii) the valve timing of the engine.
- (iv) the resistance offered to air-fuel charge during its passage through induction manifold.

17.3.8 Combustion Efficiency

Combustion efficiency is the ratio of heat liberated to the theoretical heat in the fuel. The amount of heat liberated is less than the theoretical value because of incomplete combustion either due to dissociation or due to lack of available oxygen. Combustion efficiency in a well adjusted engine varies from 92% to 97%.

17.4 ENGINE PERFORMANCE CHARACTERISTICS

Engine performance characteristics are a convenient graphical presentation of an engine performance. They are constructed from the data obtained during actual test runs of the engine and are particularly useful in comparing the performance of one engine with that of another. In this section some of the important performance characteristics of the SI engines are discussed.

It is to be noted that there is a certain speed, within the speed range of a particular engine, at which the charge inducted per cylinder per cycle will be the maximum. At this point, the maximum force can therefore be exerted on the piston. For all practical purposes, the torque, or engine capacity to do work, will also be maximum at this point. Thus, *there is a particular engine speed at which the charge per cylinder per cycle is a maximum, and at approximately this same speed, the torque of the engine will be a maximum.*

As the speed of the engine is increased above this speed the quantity of the indicated charge will decrease. However, the power output of the engine increases with speed due to more number of cycles are



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18

ENGINE ELECTRONICS

18.1 INTRODUCTION

Engine development has become a challenge mainly because the user expects improved performance levels with reduced fuel consumption. Further, environmental considerations necessitate to adopt the use of methods which lower emissions. Achieving the twin goal of low emissions with good performance and drivability is a difficult task. It is in this context that electronic engine management has gained importance. The principal aims of an engine developer is to achieve:

- (i) High reliability and durability with lowest possible initial cost
- (ii) High power output and torque
- (iii) Low levels of gaseous and particulate emissions
- (iv) Low fuel consumption
- (v) Low noise levels and vibrations

Over the years lot of developments in electronics have taken place. Therefore, in gasoline engines control of the following parameters is easily monitored by using sensors and actuators linked to a electronic control module. The important parameters that can be controlled are:

- (i) Air-fuel ratio
- (ii) Mixture distribution between cylinders
- (iii) Ignition timing



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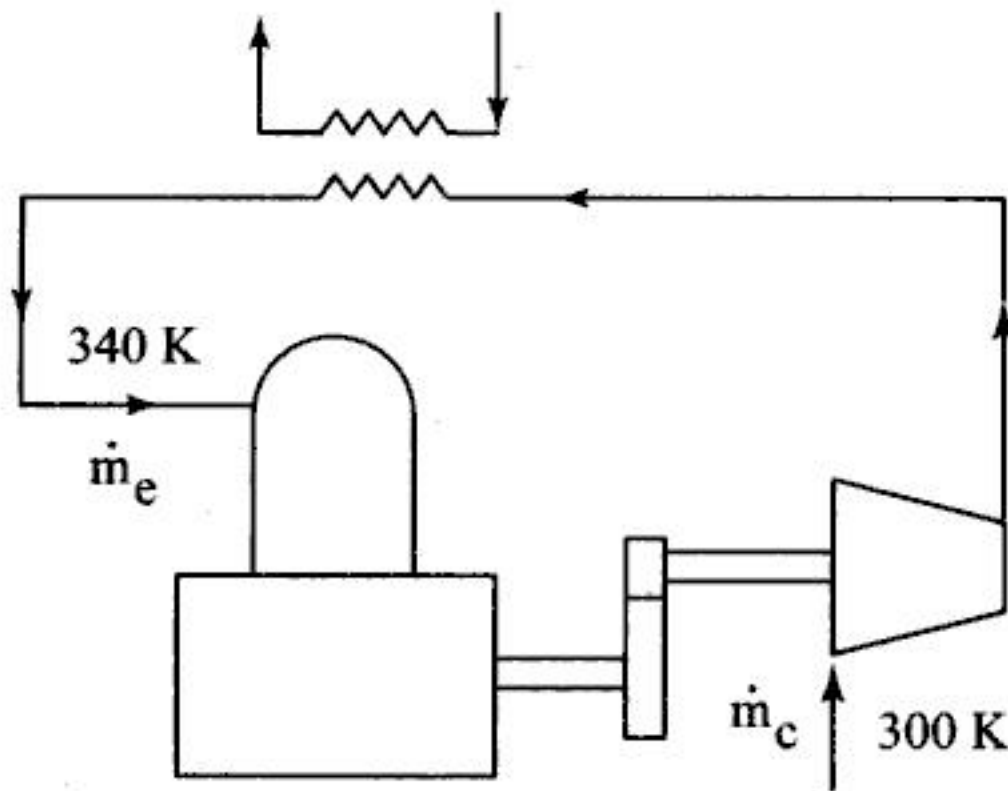
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based on intake manifold pressure and temperature. The other details about the engine are: bp is 50 kW, speed is 3000 rpm and mechanical efficiency η_m is 85%.

Determine

- (i) $imep$ of the engine
- (ii) air consumption rate of the engine
- (iii) the air handling capacity of the compressor in kg/min.

Take isentropic efficiency the compressor to be 85%.

Solution

$$ip = \frac{bp}{\eta_m} = \frac{50}{0.85} = 58.82 \text{ kW}$$

$$ip \text{ per cylinder} = \frac{58.82}{4} = 14.7 \text{ kW}$$

$$imep = \frac{ip}{LAN}$$

$$= \frac{14.7 \times 60000}{0.1 \times \frac{\pi}{4} \times 0.1^2 \times \frac{3000}{2}} \times 10^{-5}$$

$$= 7.49 \text{ bar}$$

Ans

$$\text{Swept volume} = \frac{\pi}{4} D^2 L = \frac{\pi}{4} \times 0.1^2 \times 0.1$$

$$= 7.85 \times 10^{-4} \text{ m}^3$$



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