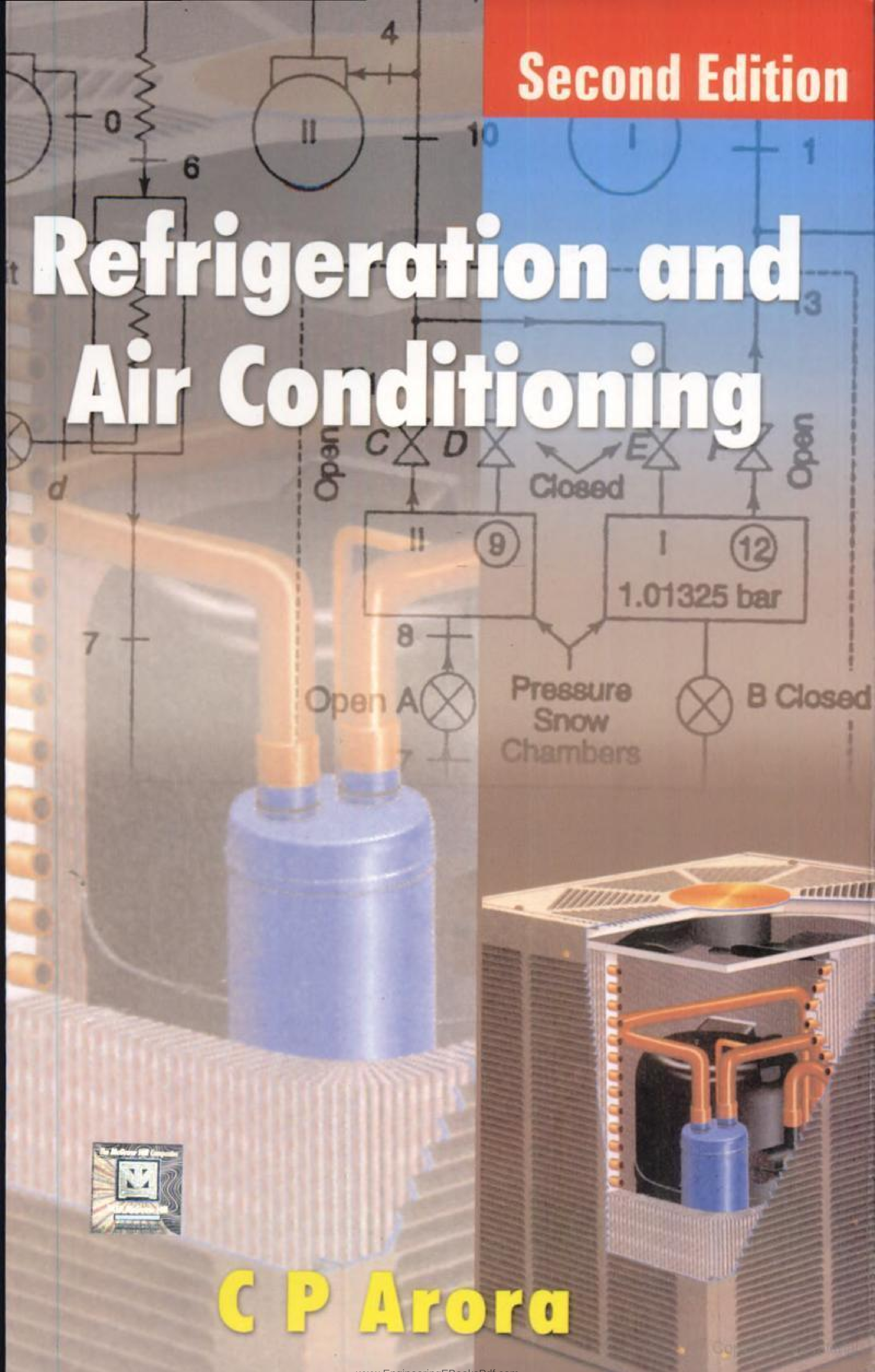
Second Edition

Refrigeration and Air Conditioning



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List of Principal Symbols

Capital letters

A	Area
A_F	Face area
C	Velocity, thermal conductance, concentration (in mass transfer), clearance factor, heat capacity rate
D	Diameter, diffusion coefficient, mass of vapour distilled from generator
E	Emissive power
F	Force, genometric factor, rich solution circulation
G	Mass velocity
H	Enthalpy, head
I	Solar radiation intensity
I_D	Intensity of direct solar radiation
I_d	Identity of diffuse solar radiation
K	Dynamic loss coefficient
L	Fin width, length, air mass
M	Molecular weight, stability criterion in finite difference approximation
N	Number of tubes
P	Perimeter, power requirement
Q	Heat transfer rate
Q_L	Latent heat transfer
Q_s	Sensible heat transfer
Q_v	Volume flow rate
R	Gas constant, thermal resistance
S	Entropy
T	Absolute temperature
U	Internal energy, overall heat transfer coefficient
V	Volume
V_p	Piston displacement
W	Work, moisture content of material
X	Bypass factor

Small letters

a	Velocity of sound, absorptivity
d	Solar declination angle
f	Heat transfer coefficient, friction factor, specific rich solution circulation
g	Acceleration due to gravity
h	Specific enthalpy, heat transfer coefficient, hour angle
h_M	Mass transfer coefficient
k	Thermal conductivity
k_d	Diffusion coefficient
k_ω	Diffusion coefficient based on specific humidity
l	Fin height, tube length
m	Mass, polytropic index of expansion
n	Polytropic index of compression, number of moles, recirculation number
p	Pressure
Δp	Pressure loss
p_S	Static pressure
p_T	Total pressure
p_V	Velocity pressure
q	Heat flux, heat transfer per unit mass
r	Radius, compression ratio, reflectivity
s	Specific entropy
t	Celsius temperature
t_e	Sol-air temperature
Δt_E	Equivalent temperature difference
u	Specific internal energy, tangential velocity
v	Specific volume
w	Specific work, moisture removal
x	Distance, dryness fraction, liquid phase mole fraction
y	Vapour phase mole fraction
z	Height above datum

Greek letters

α	Thermal diffusivity, wall solar azimuth angle
β	Coefficient of thermal expansion, solar altitude
γ	Adiabatic index, solar azimuth angle
δ	Joule Thomson coefficient
$\mathcal{E}$	Coefficient of performance
ε	Emissivity, heat exchanger effectiveness
λ	Decrement factor
η	Efficiency
η_p	Polytropic efficiency
ϕ	Flow coefficient, relative humidity, time lag
σ	Stefan-Boltzman constant, surface tension
μ	Dynamic viscosity, head coefficient, degree of saturation

ν	Kinematic viscosity
ρ	Density
ψ	Zenith angle
ξ	Concentration by weight
τ	Time, transmissivity
θ	Angle of incidence, excess temperature
ω	Specific humidity, angular velocity
$\mathcal{H}_{u}$	Lockhart-Martinelli parameter for two phase turbulent flow

Dimensionless numbers

Bi	Biot number
Bo	Boiling number
Co	Condensation number
Fo	Fourier number
Gr	Grashof number
$\mathbf{K_f}$	Load factor in boiling
Le	Lewis number
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
Sc	Schmidt number
Sh	Sherwood number
St	Stanton number
θ	Trouton number

Subscripts

A	Absorber
C	Convective
I	Infiltration
R	Radiative
S	Apparatus dew point, wetted surface
TP	two phase
a	Ambient, poor solution, dry air
b	Black body
c	Cold, clearance, condensate, critical
d	Dynamic loss, diffusion, vapour from generator, dew point, discharge
e	Entrainment
f	Friction, saturated liquid, fin, fouling, fluid
fg	Vaporization
g	Glass, saturated vapour, air-side
h	Generator, hot
i	Inside, initial
is	Isentropic
k	Heat rejection

m	Log mean
max	Maximum
min	Minimum
n	Nozzle, normal to surface
o	Outside, heat absorption or refrigeration, molar, stagnation
r	Radial, refrigerant-side, rich solution
rel	Relative
s	Suction, at normal boiling point, saturation, saturated solid
sat	Saturation
sd	Shading
sg	Sublimation
t	Total, based on extended surface side area
u	Tangential
v	Vapour, volumetric
w	Wall, water
x	x-direction
∞	Free stream

Superscripts

*	Per ton refrigeration, thermodynamic wet bulb
′	Wet bulb
L	Saturated liquid mixture
V	Saturated vapour mixture

Chapter 1

Introduction

1.1 A BRIEF HISTORY OF REFRIGERATION

The methods of production of cold by mechanical processes are quite recent. Long back in 1748, William Coolen of Glasgow University produced refrigeration by creating partial vacuum over *ethyl ether*. But, he could not implement his experience in practice. The first development took place in 1834 when Perkins proposed a hand-operated compressor machine working on ether. Then in 1851 came Gorrie's air refrigeration machine, and in 1856 Linde developed a machine working on ammonia.

The pace of development was slow in the beginning when steam engines were the only prime movers known to run the compressors. With the advent of electric motors and consequent higher speeds of the compressors, the scope of appli-cations of refrigeration widened. The pace of development was considerably quickened in the 1920 decade when du Pont put in the market a family of new working substances, the fluoro-chloro derivatives of methane, ethane, etc.—popularly known as chloro fluorocarbons or CFCs—under the name of *Freons*. Recent developments involve finding alternatives or substitutes for Freons, since it has been found that chlorine atoms in Freons are responsible for the depletion of ozone layer in the upper atmosphere. Another noteworthy development was that of the ammonia-water vapour absorption machine by Carre. These developments account for the major commercial and industrial applications in the field of refrigeration.

A phenomenon called *Peltier* effect was discovered in 1834 which is still not commercialized. Advances in *cryogenics*, a field of very low temperature refrigeration, were registered with the liquefaction of oxygen by Pictet in 1877. Dewar made the famous Dewar flask in 1898 to store liquids at cryogenic temperatures. Then followed the liquefaction of other permanent gases including helium in 1908 by Onnes which led to the discovery of the phenomenon of *superconductivity*. Finally in 1926, Giaque and Debye independently proposed adiabatic demagnetization of a paramagnetic salt to reach temperatures near absolute zero.

Two of the most common commercial refrigeration applications, viz., a window-type air conditioner and a domestic refrigerator, have been described in the following pages.

1.1.1 Room Air Conditioner

Figure 1.1 shows the schematic diagram of a typical window-type room air conditioner, which works according to the principle described below:

Consider that a room is maintained at constant temperature of 25°C. In the air conditioner, the air from the room is drawn by a fan and is made to pass over a *cooling coil*, the surface of which is maintained, say, at a temperature of 10°C. After passing over the coil, the air is cooled (for example, to 15°C) before being supplied to the room. After picking up the room heat, the air is again returned to the cooling coil at 25°C.

Now, in the cooling coil, a liquid working substance called a *refrigerant*, such as $CHClF_2$ (monochloro-difluoro methane), also called *Freon 22* by trade name, or simply *Refrigerant 22* (R 22), enters at a temperature of, say, 5°C and evaporates, thus absorbing its latent heat of vaporization from the room air. This equipment in which the refrigerant evaporates is called an *evaporator*.

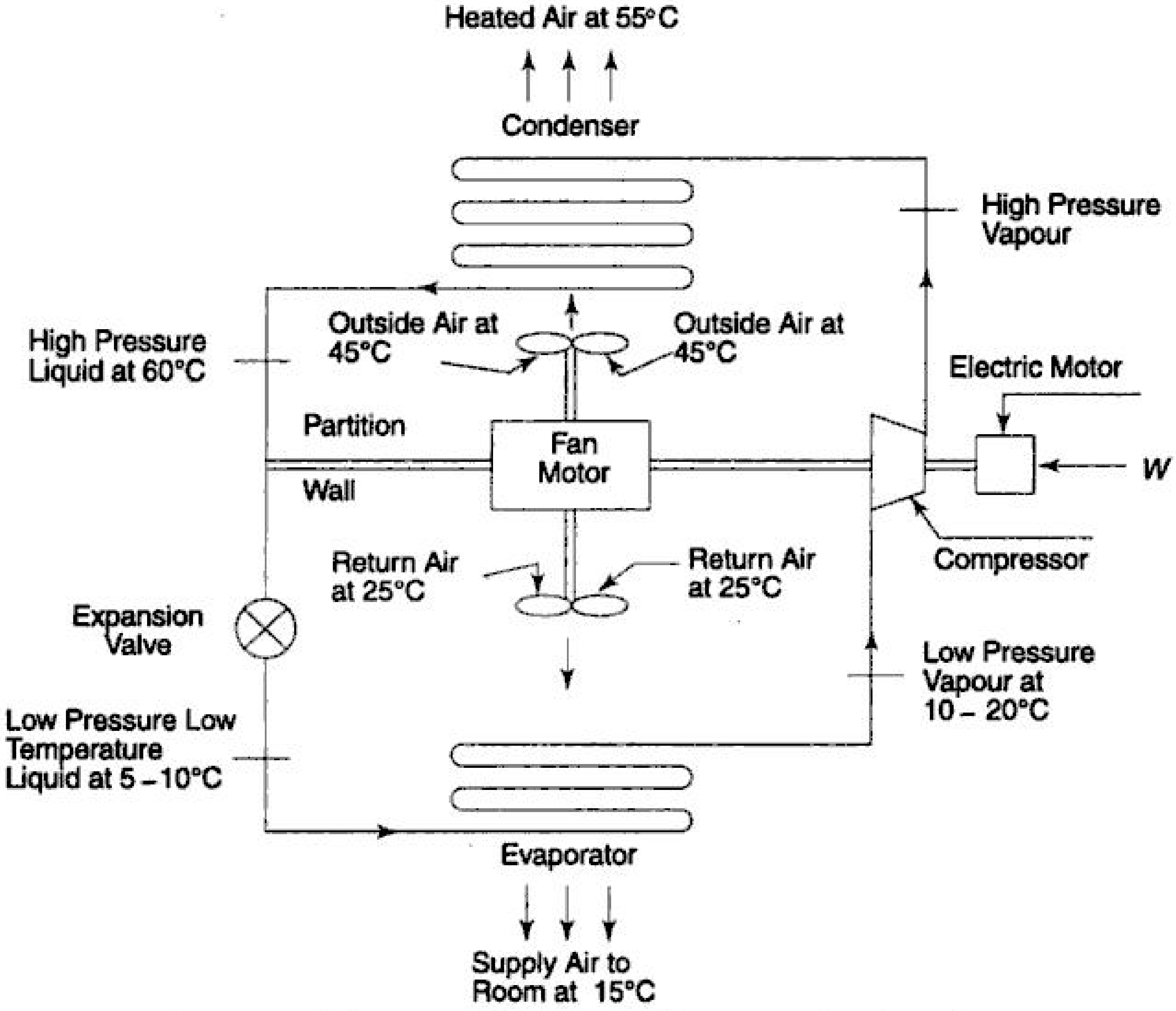


Fig. 1.1 Schematic Diagram of a Room Air Conditioner

After evaporation, the refrigerant becomes vapour. To enable it to condense back and to release the heat—which it has absorbed from the room while passing through the evaporator—its pressure is raised by a *compressor*. Following this, the high pressure vapour enters the *condenser*. In the condenser, the outside atmospheric air, say, at a temperature of 45°C in summer, is circulated by a fan. After picking up the latent heat of condensation from the condensing refrigerant, the air is let out into the environment, say, at a temperature of 55°C. The condensation of refrigerant may occur, for example, at a temperature of 60°C.

After condensation, the high pressure liquid refrigerant is reduced to the low pressure of the evaporator by passing it through a pressure reducing device called the *expansion device*, and thus the cycle of operation is completed. A partition wall separates the high temperature side of the condenser from the low temperature side of the evaporator.

The principle of working of large air conditioning plants is also the same, except that the condenser is *water cooled* instead of being *air cooled*.

1.1.2 Domestic Refrigerator

The working principle of a domestic refrigerator is exactly the same as that of an air conditioner. A schematic diagram of the refrigerator is shown in Fig. 1.2. Like the air conditioner, it also consists of the following four basic components:

(i) Evaporator; (ii) Compressor; (iii) Condenser; (iv) Expansion device.

But there are some design features which are typical of a refrigerator. For example, the evaporator is located in the *freezer* compartment of the refrigerator. It forms the coldest part of the cabinet with a temperature of about –15°C, while the refrigerant evaporates inside the evaporator tubes at –25°C. Just below the

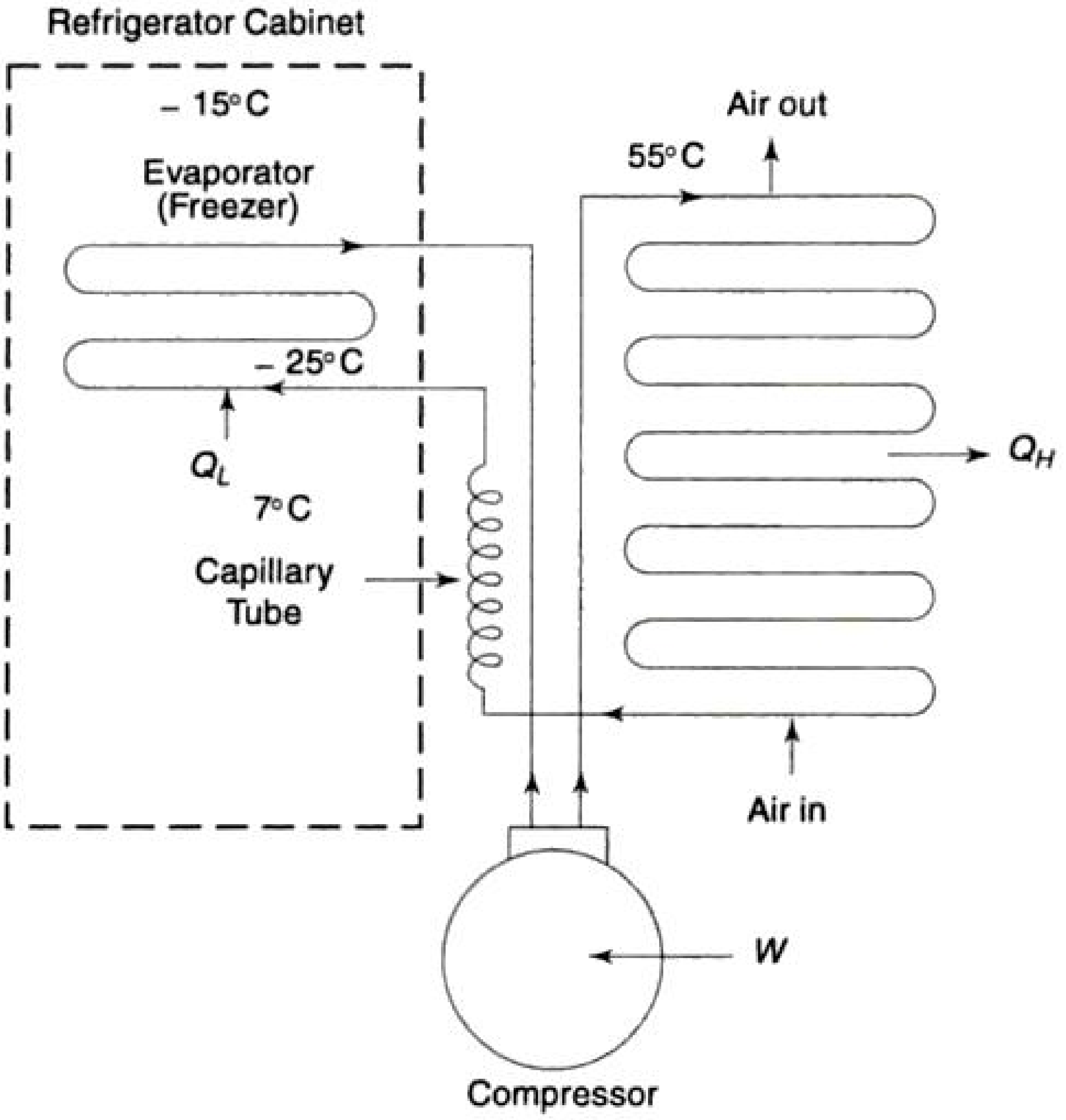


Fig. 1.2 Schematic Diagram of a Domestic Refrigerator

freezer, there is a *chiller tray*. Further below are compartments with progressively higher temperatures. The bottom-most compartment which is meant for vegetables is the least cold one. The cold air being heavier flows down from the freezer to the bottom of the refrigerator. The warm air being lighter rises from the vegetable compartment to the freezer, gets cooled and flows down again. Thus, a *natural convection current* is set up which maintains a temperature gradient between the top and the bottom of the refrigerator. The temperature maintained in the freezer is about –15°C, whereas the mean inside temperature of the cabinet is 7°C.

The design of the condenser is also a little different. It is usually a *wire and tube* or *plate and tube* type mounted at the back of the refrigerator. There is no fan. The refrigerant vapour is condensed with the help of surrounding air which rises above by natural convection as it gets heated after receiving the latent heat of condensation from the refrigerant. The standard condensing temperature is 55°C.

Note ✍: *In both the room air conditioner as well as the refrigerator a long narrow bore tube, called the capillary tube, is employed as the expansion device.*

In the modern no-frost refrigerators, the evaporator is located outside the freezer compartment. The cold air is made to flow by forced convection by a fan.

Working Substances in Refrigerating Machines Presently, the working substance being used in air conditioners is R22, and that in refrigerators is R12. R12 is a CFC (chloro-fluoro carbon). Because of the ozone-layer depletion problem, alternatives such as the following are being used in place of R12.

1. Refrigerant 290 or R290, viz., Propane (C_3H_8).
2. Refrigerant 134a or R134a, viz., Tetra-fluoroethane ($C_2H_2F_4$)
3. Refrigerant 600a or R 600a, viz., Isobutane (C_4H_{10}).

1.2 SYSTEME INTERNATIONAL D'UNITES (SI UNITS)

SI or the International System of Units is the purest form and an extension and refinement of the traditional metric system. In SI, the main departure from the traditional metric system is in the use of *Newton* as the unit of force.

There are six basic SI units as given in Table 1.1. The units of other thermodynamic quantities may be derived from these basic units.

Table 1.1 Basic SI Units

Quantity	*Unit*	*Symbol*
Length	metre	m
Mass	kilogram	kg
Time	second	s
Temperature	kelvin	K
Electric current	ampere	A
Luminous intensity	candela	cd

The unit of temperature is kelvin which measures the absolute temperature given by

$$T = t + 273.15$$

where t is the Celsius temperature in °C.

1.2.1 Unit of Force

Force F is proportional to mass m and acceleration a, so that

$$F = C(m)\,(a) \tag{1.1}$$

where C is a proportionality constant. The SI unit of force, viz., Newton denoted by the symbol N is derived from unit values taking the proportionality constant as unity. Thus, one newton is

$$1\text{N} = (1\text{ kg})\left(1\,\frac{\text{m}}{\text{s}^2}\right) = 1\,\frac{\text{kg}\cdot\text{m}}{\text{s}^2}$$

The MKS unit of force, kgf, defined by Eq. (1.1) is

$$1\text{ kgf} = \frac{1}{9.80665}\,(1\text{ kg})\left(9.80665\,\frac{\text{m}}{\text{s}^2}\right) = 1\text{ kgf}$$

which represents a unit weight or the gravitational force on one kilogram mass. In the above definition, the value of the constant C is taken as equal to the reciprocal of the standard gravitational acceleration so that one kilogram mass has one kilogram weight.

It can be seen that

$$1\text{ kgf} = (1\text{ kg})\left(9.80665\,\frac{\text{m}}{\text{s}^2}\right) = 9.80665\text{ N}$$

Also, we known that

$$1\text{ lbf} = 0.453592\text{ kgf}$$

1.2.2 Unit of Pressure

The SI unit of pressure p can also be derived from its definition as force per unit area. Thus

$$|p| = \frac{[F]}{[A]} = \text{N/m}^2$$

The unit is also called pascal and is denoted by the symbol Pa.

Another common SI unit of pressure is bar which is equivalent to a pressure of 10^5 N/m^2 or 0.1 MN/m^2 or 100 kN/m^2. Its conversion to MKS and FPS units is as follows

$$1\text{ bar} = \frac{10^5/9.80665\text{ kgf}}{10^4\text{ cm}^2} = 1.0197\text{ kgf/cm}^2\text{ or ata}$$

$$= \frac{1.02(2.54)^2}{0.453592} = 14.5\text{ lbf/in}^2$$

It can be seen that one *standard atmosphere* is given by

$$1 \text{ atm} = 1.033 \text{ kgf/cm}^2 = 14.696 \text{ lbf/in}^2$$
$$= \frac{1.033}{1.0197} = 1.01325 \text{ bar}$$
$$= 760 \text{ mm Hg or } 760 \text{ torr}$$

Accordingly,

$$1 \text{ torr} = 1 \text{ mm Hg} = \frac{1}{760} \text{ atm} = 133 \text{ N/m}^2$$

The conversion of one *technical atmosphere*, i.e. ata is obtained as:

$$1 \text{ ata} = 1 \text{ kgf/cm}^2 = (9.80665)(10^4) = 980665 \text{ N/m}^2$$
$$= 0.980665 \text{ bar}$$
$$= (0.980665)(14.5) = 14.22 \text{ lbf/in}^2$$
$$= \frac{98066.5}{133} = 736 \text{ torr or mm Hg}$$

The conversion of other units of pressure are

$$1 \text{ cm H}_2\text{O} = \left(\frac{10^4 \times 1}{1000}\right) \text{kg} \left(9.80665 \frac{\text{m}}{\text{s}^2}\right) = 98.1 \text{ N/m}^2$$

$$1 \text{ in Hg} = (25.4 \text{ mm}) \left(133 \frac{\text{N/m}^2}{\text{mm Hg}}\right) = 3390 \text{ N/m}^2$$

1.2.3 Unit of Energy (Work and Heat)

The unit of work or energy is obtained from the product of force and distance moved. The SI unit of work is *Newton metre* denoted by Nm or Joule denoted by J. Thus

$$1 \text{ Nm} = 1\text{J} = (1 \text{ kg m/s}^2)(1 \text{ m}) = 1 \text{ kg. m}^2/\text{s}^2$$

Since both heat and work are energy, the SI unit of heat is the same as the unit of work, viz., joule. The conversion of the MKS unit of heat, viz., kcal, is obtained from its mechanical equivalent of heat which is 427 kcal/kgfm. Thus:

$$1 \text{ kcal} = 427 \text{ kgf m} = (427)(9.80665 \text{ N})\text{m}$$
$$= 4186.8 \text{ Nm or J}$$
$$= 4.1868 \text{ kJ}$$

Also $$1 \text{ kcal} = (1 \text{ kg of water})(1°\text{C})$$
$$= \frac{1}{0.453} \times \frac{9}{5} \text{ lb °F} = 3.968 \text{ Btu}$$

Hence $$1 \text{ kcal} = 4.1868 \text{ kJ} = 3.968 \text{ Btu}$$
$$1 \text{ kJ} = 0.948 \text{ Btu} = 0.239 \text{ kcal}$$
$$1 \text{ Btu} = 0.252 \text{ kcal} = 1.055 \text{ kJ}$$

1.2.4 Unit of Power

The SI unit of power is *watt*, denoted by the symbol W. It is defined as the rate of doing 1 Nm of work per second. Thus

$$1 \text{ W} = 1 \text{ J/S} = 1 \text{ Nm/s}$$

It may also be noted that watt also represents the electrical unit of work defined by

$$1 \text{ W} = 1 \text{ (volt)} \times 1 \text{ (ampere)} = 1 \text{ J/s}$$

The conversion of the horsepower unit can also be obtained

$$\underset{\text{(imperial)}}{1 \text{ hp}} = 550 \frac{\text{ft.1bf}}{\text{s}} = \frac{(550 \times 0.3048 \text{ m})(0.453592 \times 9.80665 \text{ N})}{\text{s}}$$

$$= 746 \text{ Nm/s or J/s or W}$$

$$\underset{\text{(metric)}}{1 \text{ hp}} = 75 \frac{\text{kgf.m}}{\text{s}} = (75 \times 9.80665 \text{ N}) \frac{\text{m}}{\text{s}}$$

$$= 736 \text{ Nm/s or J/s or W}$$

Further, the units of energy can be derived from those of power. Thus

$$1 \text{ J} = 1 \text{ Ws}$$

$$1 \text{ kWH} = 3{,}600{,}000 \text{ J} = 3{,}600 \text{ kJ} = 860 \text{ kcal} = 3{,}410 \text{ Btu}$$

$$\underset{\text{(imperial)}}{1 \text{ hp/hr}} = 746 \times 3{,}600 \text{ J} = 2{,}680 \text{ kJ} = 641 \text{ kcal} = 2{,}540 \text{ Btu}$$

$$\underset{\text{(metric)}}{1 \text{ hp/hr}} = 736 \times 3{,}600 \text{ J} = 2{,}650 \text{ kJ} = 632 \text{ kcal} = 2{,}510 \text{ Btu}$$

1.2.5 Unit of Enthalpy

The interconversion of units of enthalpy are as follows

$$1 \text{ kJ/kg} = 0.239 \text{ kcal/kg} = 0.42 \text{ Btu/lb}$$

$$1 \text{ kcal/kg} = 4.19 \text{ kJ/kg} = 1.8 \text{ Btu/lb}$$

$$1 \text{ Btu/lb} = 0.556 \text{ kcal/kg} = 2.33 \text{ kJ/kg}$$

Note ✍: *The definition of enthalpy (H) (and specific enthalpy (h)) is obtained by the application of the First Law of Thermodynamics to a thermodynamic process.*

1.2.6 Units of Entropy and Specific Heat

These are expressed as

$$1 \text{ kJ/kg.K} = 0.239 \text{ kcal/kg°C or Btu/lb°F}$$

$$1 \text{ kcal/kg°C} = 1 \text{ Btu/lb°F} = 4.1868 \text{ kJ/kg.K}$$

Note ✍: *The definition of entropy (S) (and specific entropy (s)) is obtained by the application of the Second Law of Thermodynamics to a thermodynamic process.*

1.2.7 Unit of Refrigerating Capacity

The standard unit of refrigeration in vogue is *ton refrigeration* or simply ton denoted by the symbol TR. It is equivalent to the production of cold at the rate at which heat is to be removed from one US tonne of water at 32°F to freeze it to ice at 32°F in one day or 24 hours. Thus

$$1\ \text{TR} = \frac{1 \times 2{,}000\ \text{lb} \times 144\ \text{Btu/lb}}{24\ \text{hr}}$$
$$= 12{,}000\ \text{Btu/hr} = 200\ \text{Btu/min}$$

where the latent heat of fusion of ice has been taken as 144 Btu/lb. The term one ton refrigeration is a carry over from the time ice was used for cooling. In general 1 TR always means 12,000 Btu of heat removal per hour, irrespective of the working substance used and the operating conditions, viz., temperature of refrigeration and heat rejection. This unit of refrigeration is currently in use in the USA, the UK and India. In many countries, the standard MKS unit of kcal/hr is used.

It can be seen that

$$1\ \text{TR} = 12{,}000\ \text{Btu/hr}$$
$$= \frac{12{,}000}{3.968} = 3{,}024.2\ \text{kcal/hr}$$
$$= 50.4\ \text{kcal/min} \approx 50\ \text{kcal/min}$$

Also, since 1 Btu = 1.055 kJ, the conversion of ton into equivalent SI unit is:

$$1\ \text{TR} = 12{,}000 \times 1.055 = 12{,}660\ \text{kJ/hour}$$
$$= 211\ \text{kJ/min} = 3.5167\ \text{kW}$$

Example 1.1 The performance test of an air conditioning unit rated as 140.7 kW (40 TR) seems to be indicating poor cooling. The test on heat rejection to atmosphere in its condenser shows the following:

Cooling water flow rate:	4 L/s
Water temperatures:	In 30°C: Out 40°C
Power input to motor:	48 kW (95% efficiency)

Calculate the actual refrigerating capacity of the unit.

Solution Heat rejected in condenser

$$\dot{Q}_{\text{condenser}} = \dot{m}_w C_w \Delta t_w$$
$$= 4\ (4.1868)\ (40 - 30) = 167.5\ \text{kW}$$

Work input

$$\dot{W} = 48\ (0.95) = 45.6\ \text{kW}$$

Refrigeration capacity (by energy balance)

$$\dot{Q}_{\text{refrigeration}} = \dot{Q}_{\text{condenser}} - \dot{W}$$
$$= 167.5 - 45.6 = 121.9 \text{ kW } (34.7 \text{ TR})$$

The unit is operating below its rated capacity of 40 TR.

1.3 THERMODYNAMIC SYSTEMS, STATE, PROPERTIES, PROCESSES, HEAT AND WORK

Thermodynamic systems are of two types. They are either *closed* or *open* as illustrated in Fig. 1.3. A closed system is one across whose *boundary* only heat Q and work W flow. In an open system the working fluid also crosses the *control surface* drawn around the system. Everything outside the system is *surroundings*. The system plus surroundings combine to make the *universe*.

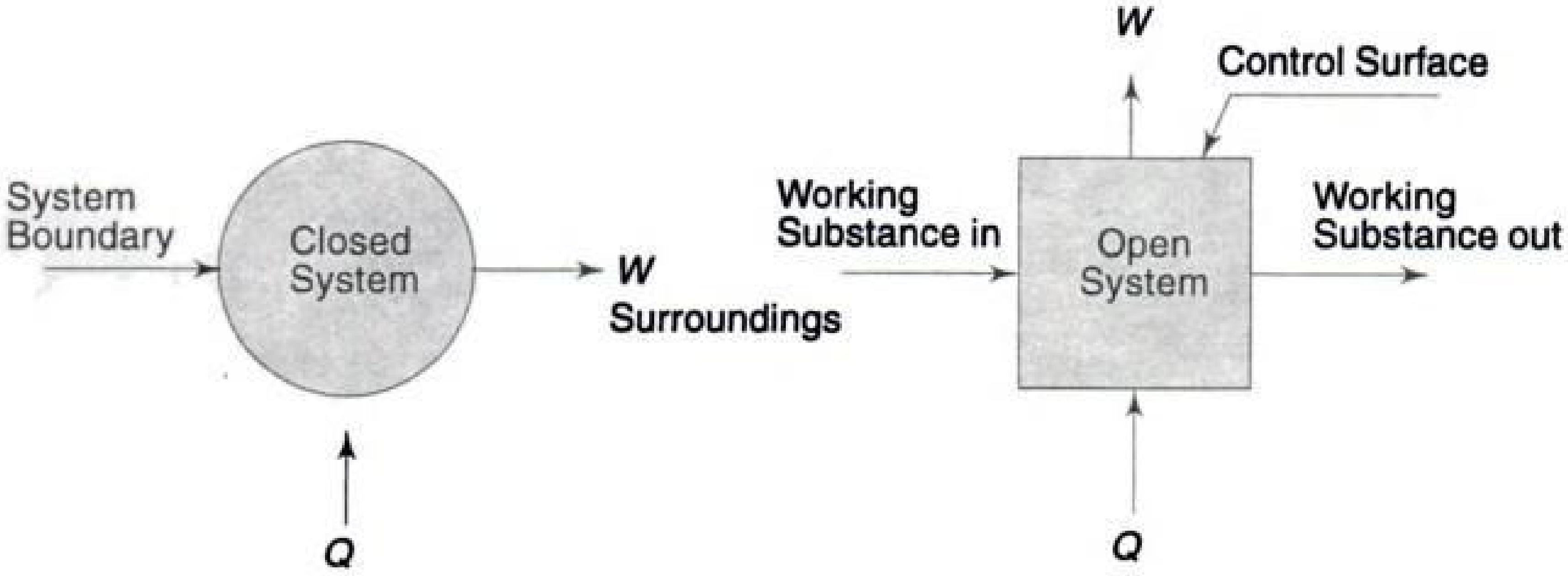


Fig. 1.3 Closed and Open Systems

The state of a thermodynamic system is characterised by its *properties*. The change of state of the working substance represents a *thermodynamic process*. Thermodynamic processes occurring in a closed system are called *non-flow processes*. Likewise, thermodynamic processes occurring in an open system are called *flow processes*.

Further, the processes that can be reversed such that the system and environment, both, can be restored to the initial state are called *reversible processes*. The processes which, when reversed, will not restore *both* the system and environment to the initial state are called *irreversible processes*.

The properties are either *intensive or extensive*. Intensive properties do not depend on the size of the system. These are, e.g., pressure p and temperature T. The extensive properties depend on the size of the system, e.g., volume V, internal energy U, enthalpy H, entropy S, etc. Their numerical values per unit mass of the working substance are called the *specific properties* denoted by lower case symbols, viz., v, u, h, s, etc. The specific properties are intensive properties.

A *thermodynamic process* is accompanied with *heat* and *work* interactions between the system and the surroundings. The heat added to the system is considered as positive, and that rejected by the system as negative. The sign convention for work is the opposite. The work done by the system is positive and the work done *on* the system is negative.

The heat and work interactions per unit mass of the *working substance* in the system are denoted as q and w.

Note ✍: *The work done in a reversible process in a simple compressible system is given by*

$$W = \int p\,dV$$

Note that in an irreversible process, the work is not given by $\int p\,dV$.

1.4 FIRST LAW OF THERMODYNAMICS

The first law of thermodynamics is mathematically stated as follows:

$$\oint \delta Q = \oint \delta W \tag{1.2}$$

Accordingly, during a *thermodynamic cycle*, viz., a cyclic process the system undergoes, the cyclic integral of heat added is equal to the cyclic integral of work done. Equation (1.2) can also be written for a cycle as

$$\oint (\delta Q - \delta W) = 0$$

Equation (1.3) below is a corollary of the first law. It shows that there exists a property U, named *internal energy* of the system/substance, such that a change in its value is equal to the difference in heat entering and work leaving the system. Accordingly, for a process in a closed system, the first law can be written as:

$$\delta Q = \delta U + \delta W \tag{1.3}$$

For the change of state of a system from initial state 1 to final state 2, this becomes

$$Q = U_2 - U_1 + W$$

Another property named *enthalpy H* can also be defined now as a combination of properties U, p and V,

$$H = U + pV,\ h = u + pv$$

For a reversible process, since $\delta W = p\mathrm{d}V$, the first law can also be written as

$$\delta Q = \mathrm{d}U + p\mathrm{d}V,\ \delta q = \mathrm{d}u + p\mathrm{d}v \tag{1.4a}$$

$$\delta Q = \mathrm{d}H - V\mathrm{d}p,\ \delta q = \mathrm{d}h - v\,\mathrm{d}p \tag{1.4b}$$

The first law can be applied to a process in an open system. Figure 1.4 represents an open system undergoing a *steady-state steady-flow* (SSSF) *process*. For the process, the first law takes the form of a *steady-flow energy equation* as in Eq. (1.5)

$$\dot{Q} = \dot{m}[(u_2 - u_1) + (p_2 v_2 - p_1 v_1) + \frac{1}{2}(C_2^2 - C_1^2) + g(z_2 - z_1)] + \dot{W}$$

$$= \dot{m}[(h_2 - h_1) + \frac{1}{2}(C_2^2 - C_1^2) + g(z_2 - z_1)] + \dot{W} \tag{1.5}$$

Here, in addition to change in internal energy, changes in kinetic and potential energies are also considered since these are significant. In addition, work, equal to $(p_2 v_2 - p_1 v_1)$, to make the fluid enter and leave the system called the *flow work* is also considered.

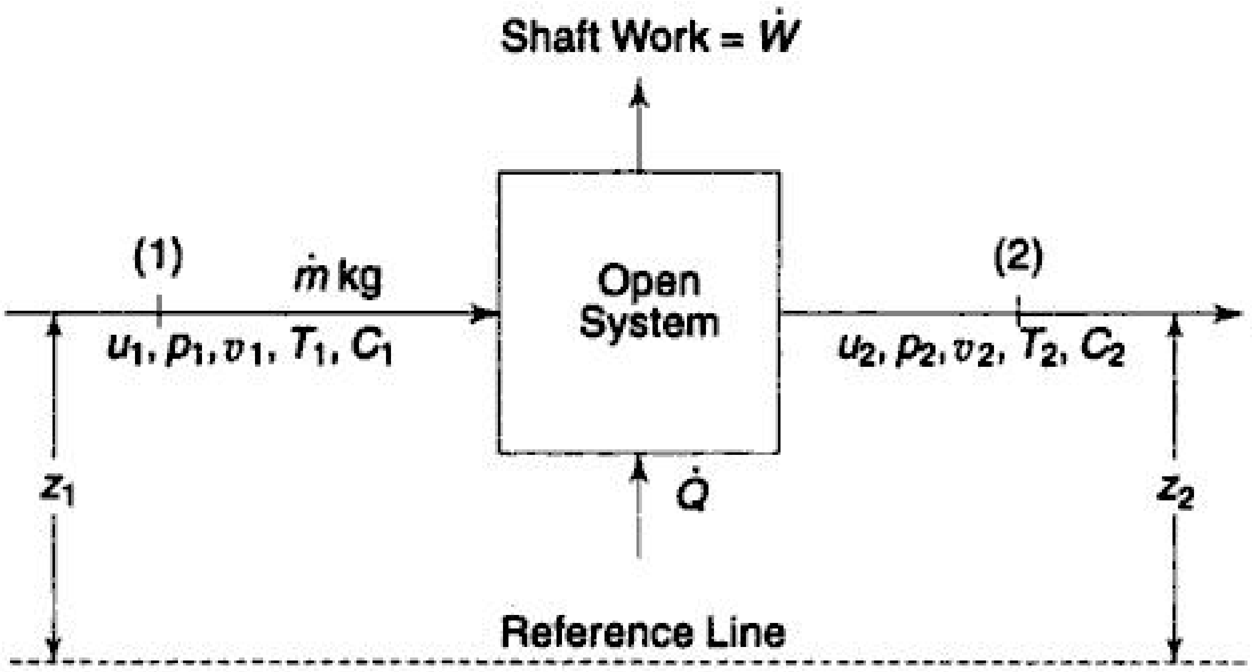


Fig. 1.4 Representation of a Steady-State Steady-Flow Process

Writing Eq. (1.5) on the basis of a unit mass entering and leaving the system, we have Eq. (1.6)

$$q + h_1 + \frac{C_1^2}{2} + gz_1 = h_2 + \frac{C_2^2}{2} + gz_2 + w \tag{1.6}$$

1.5 SECOND LAW OF THERMODYNAMICS

The second law of thermodynamics can be mathematically state for a thermodynamic cycle in the form of *Clausius Inequality* as given in Eq. (1.7)

$$\oint \frac{\delta Q}{T} \leq 0 \tag{1.7}$$

The equality holds for a reversible cycle, and the inequality for an irreversible cycle.

Just as the application of first law to a thermodynamic process led to the establishment of a new property, named internal energy (U), the application of the second law to a process leads to the establishment of another new property named *entropy* (S), defined as follows in Eq. (1.8)

$$\mathrm{d}S \equiv \left(\frac{\delta Q}{T}\right)_{\text{rev}} \tag{1.8}$$

Thus, for a reversible process, between two given states, from initial state 1 to final state 2 in a closed system, or inlet state 1 to exist state 2 in an open system, the change in entropy is given by

$$S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T}\right)_{\text{rev}}, \; s_2 - s_1 = \int_1^2 \left(\frac{\delta q}{T}\right)_{\text{rev}}$$

It is found by applying Clausius inequality that for an irreversible process

$$S_2 - S_1 > \int_1^2 \left(\frac{\delta Q}{T}\right), \; s_2 - s_1 > \int_1^2 \left(\frac{\delta q}{T}\right)$$

Note ✍: *For a reversible process in a compressible system work done* $W = \int_1^2 p\,dV$. *Hence, the area under the curve on P-V diagram gives work done in the process. Similarly, for a reversible process, heat transfer* $Q = \int_1^2 T\,dS$. *Hence, the area under the curve on T-S diagram gives heat transfer during the process.*

1.6 NON-FLOW PROCESSES

Processes in a closed system are referred to as *non-flow processes*. Since the velocities are small, and hence dissipation due to friction is negligible, most non-flow processes are considered as reversible.

In a *reversible constant volume process*, $W = \int p\mathrm{d}V = 0$.

Hence, from first law, $Q = U_2 - U_1$.

In a *reversible constant pressure process*, $W = \int p\mathrm{d}V = p(V_2 - V_1)$, and from first law, $Q = (U_2 - U_1) + p(V_2 - V_1) = H_2 - H_1$.

Also, from second law and property relation, $Q = \int T\mathrm{d}S$

$$= T\,(S_2 - S_1) = H_2 - H_1.$$

In an *isothermal process*, $Q = T(S_2 - S_1) = (U_2 - U_1) + W$.

In an *adiabatic process*, $Q = 0$ and $W = -(U_2 - U_1)$. In a *reversible adiabatic process*, in addition, we have from second law, $Q = \int T\mathrm{d}S = 0$. Hence, $S_2 = S_1$. A reversible adiabatic process is, therefore, an *isentropic process*.

A general process can be represented by the *polytropic relation*

$$p_1\,V_1^n = p_2\,V_2^n = pV^n = \text{Constant}$$

in which n is the *polytropic index*. For the polytropic process, we have Eq. (1.9) for work

$$W = \int p\mathrm{d}V = \frac{n}{1-n}(\,p_2V_2 - p_1V_1)$$

$$= \frac{n}{1-n} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{1.9}$$

1.7 STEADY-FLOW PROCESSES

The steady-flow energy equation is applicable to flow processes, viz. processes in an open system. In most flow processes, the irreversibility due to viscous friction cannot be neglected on account of significant velocities encountered. It is, therefore, necessary to calculate the unknown quantity by first assuming the process as reversible, and then multiplying or dividing the result by a *process efficiency*.

Boiling and Condensation

$$q = - h_2 - h_1 + \frac{1}{2}(C_2^2 - C_1^2)$$
$$\cong h_2 - h_1$$

as the velocities are small, and the change in kinetic energy can be neglected. If there is a significant pressure drop as in *direct-expansion evaporators* in refrigeration, the process is not completely reversible.

Isothermal Process Neglecting kinetic and potential energies

$$q - w = h_2 - h_1$$

Adiabatic Process In nozzles and diffusers, there is no heat transfer or work done. In these devices, there is interconversion between kinetic energy and enthalpy. Thus we have

$$\frac{1}{2}(C_2^2 - C_1^2) = h_1 - h_2$$

In turbines and compressor, assuming no heat transfer and neglecting changes in kinetic and potential energies, we obtain for work

$$w = h_1 - h_2$$

To account for irreversibility in processes, the efficiencies are defined as follows:

Nozzle and Turbine Efficiency

$$\eta = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

Diffuser and Compressor Efficiency

$$\eta = \frac{h_1 - h_{2s}}{h_1 - h_2}$$

In these equations, h_2 represents the enthalphy at the end in the actual process and h_{2s} represents the enthalpy if the process is executed reversibly, viz., isentropically between the pressure limits p_1 and p_2.

Throttling Process It is an irreversible adiabatic process. It is employed to reduce the pressure of a fluid by introducing a restriction in the flow passage as illustrated in Fig. 1.5.

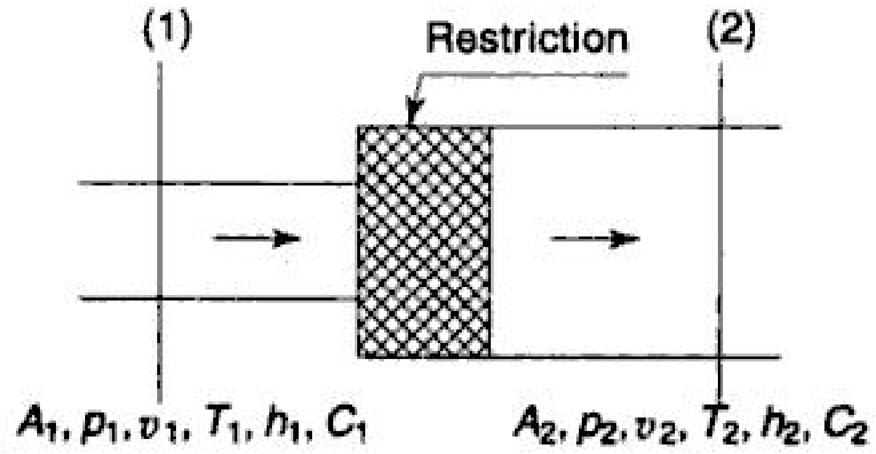


Fig. 1.5 Representation of a Throttling Process

As the process is adiabatic and no external work is done, we have

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

Since $p_2 < p_1$, $V_2 > V_1$. Hence if $A_1 = A_2$, $C_2 > C_1$. Making $A_2 > A_1$ such that $C_2 = C_1$, or since kinetic energy change is negligible, we find that in a throttling process

$$h_1 = h_2$$

Accordingly, throttling is also referred to as an *isenthalpic-expansion process*.

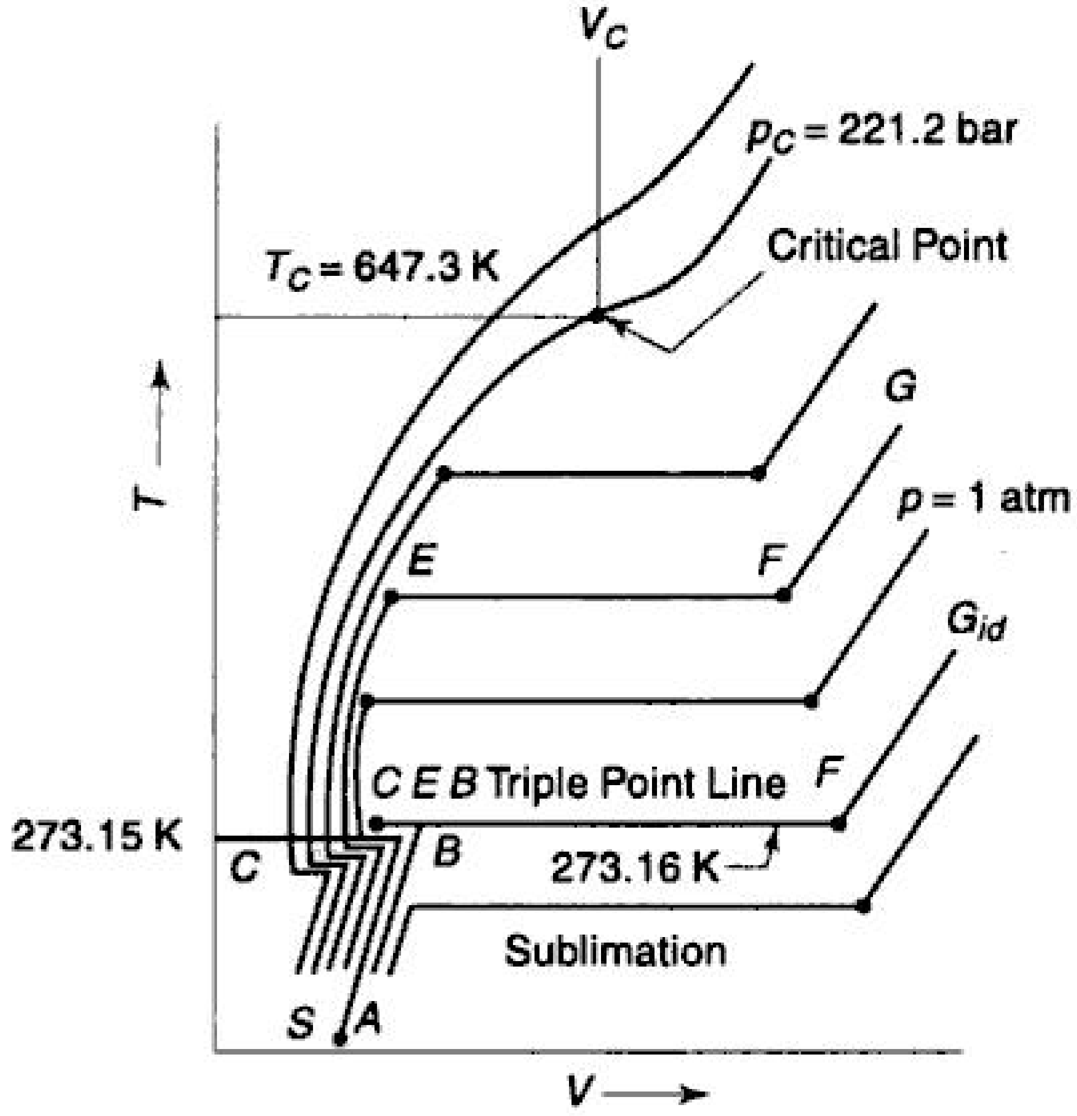
1.8 THERMODYNAMIC STATE OF A PURE SUBSTANCE

The working substance used in refrigerating machines is called a *refrigerant*. A refrigerant is usually a pure substance, though research is on to use mixtures also.

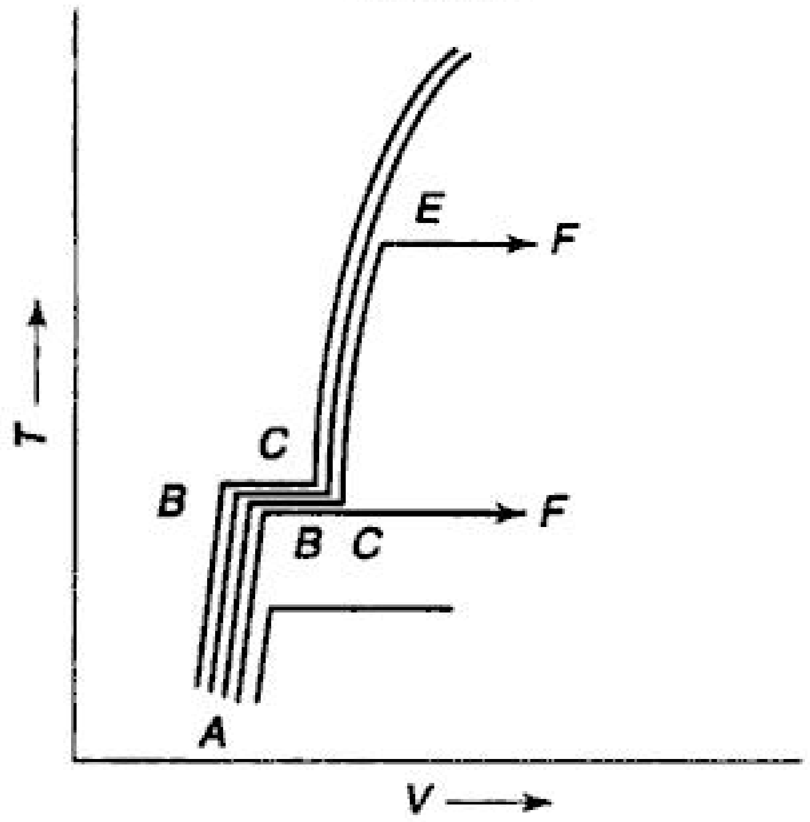
A knowledge of two independent properties is required to determine the thermodynamic state of a pure substance. In the case of equilibrium between two phases of a pure substance, only one independent property is required to define the state.

The equilibrium between phases, and the state of a pure substance are best described by the *phase diagrams* such as the $p - v$ and $T - s$ diagrams shown in Figs 1.7(a) and 1.7(b). The continuous lines on these diagrams represent the locii of equilibrium/saturation states in which phase change occurs, e.g., s and f_1 between solid and liquid, and f_2 and g between liquid and vapour. The temperature for change from solid state s to liquid state f_1 or vice versa, is the *melting/freezing/fusion temperature*. The temperature for charge from liquid state f_2 to vapour state g, and vice versa, is the *boiling/condensation temperature*. Similarly the temperature for change between solid state *s* and vapour state *g* is the ***sublimation/ablimation temperature.***

Consider a unit mass of solid at *S* below its melting point at 1 atm pressure. Let heat be supplied to it, and let us follow the events that occur at constant pressure as shown in Fig. 1.6(a) for water and Figs 1.6(b), 1.7(a) and 1.7(b) for a normal substance. We observe the following.



(a) Water



(b) Normal Substance

Fig. 1.6 Temperature-Specific Volume Phase Diagrams for Water and Normal Substances

(i) Temperature rises until point *s* is reached.

(ii) Further heating results in melting of solid at constant (melting point) temperature until point f_1 is reached.

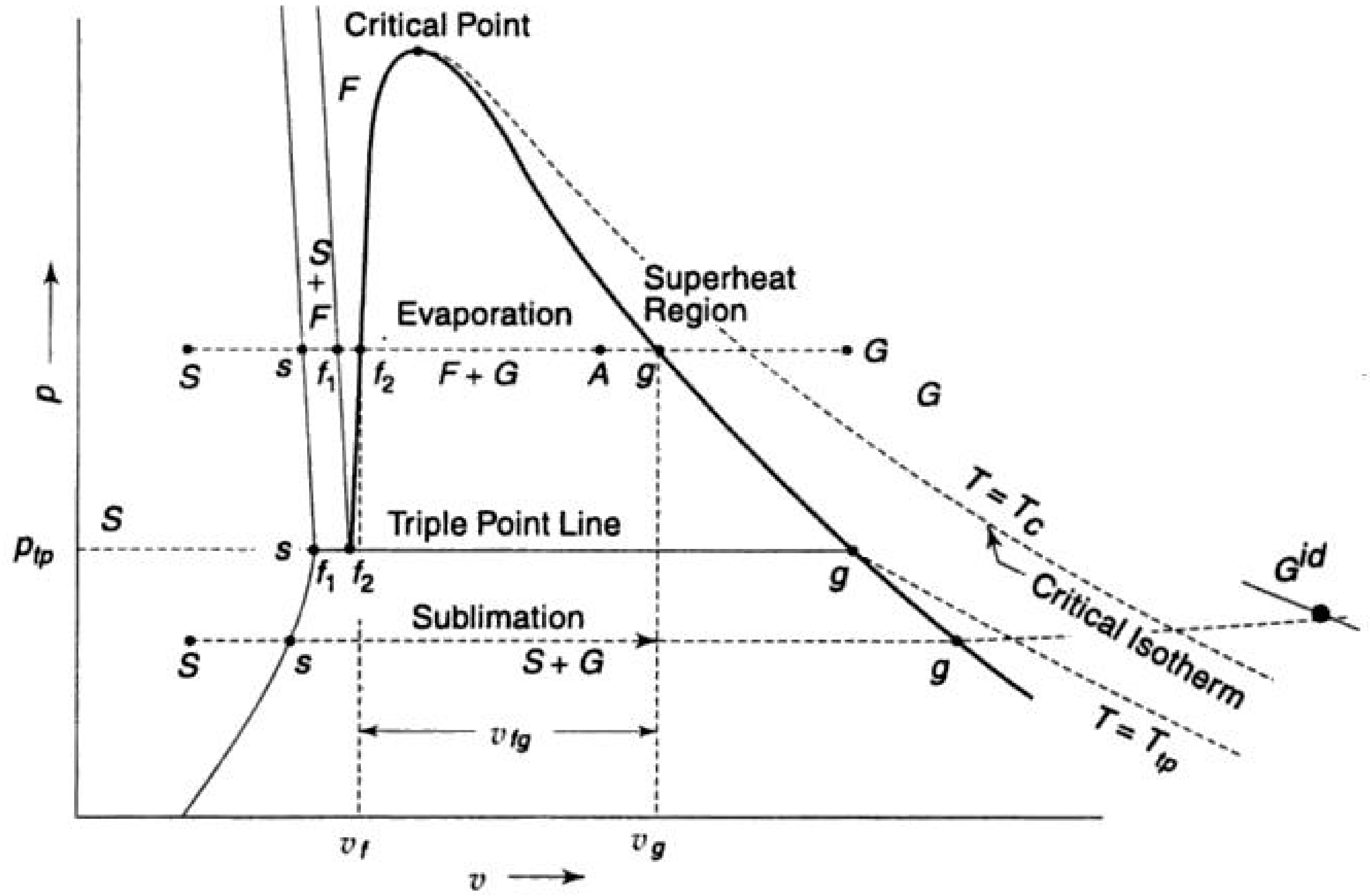


Fig. 1.7(a) *p-v* phase Diagram for a Normal Substance

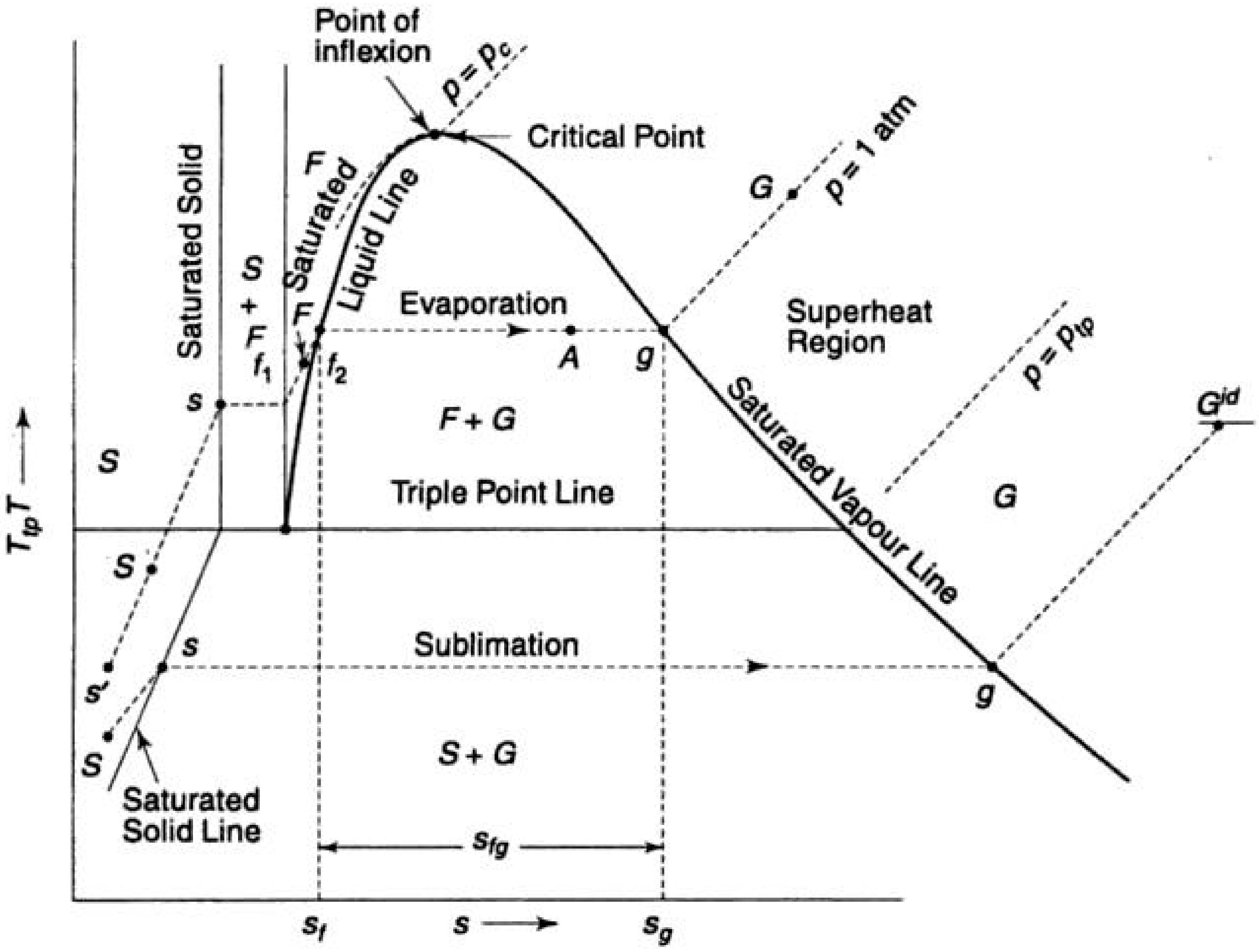


Fig.1.7(b) *T-s* Phase Diagram for a Normal Substance

(iii) Change of phase from solid to liquid is complete at f_1. Heating of liquid now results in rise in temperature until point f_2 is reached.

(iv) Further heating at f_2 results in vaporization of liquid at constant (boiling point) temperature until point g is reached.

(v) Change of phase from liquid to vapour is complete at g. Heating of vapour/gas at g results in rise in temperature again until, say point G above the boiling point temperature is reached.

States, f_1, f_2 and g are saturation states. f_1 represents saturated liquid state in equilibrium with saturated solid state s, and f_2 represents saturated liquid state in equilibrium will saturated vapour stage g. Now onwards, the subscripts 1 and 2 from f_1 and f_2 will be dropped, and the context will tell which one we are referring to.

Note that there is a large change in volume equal to $v_{fg} = v_g - v_f$ during vaporization from f_2 to g. Similarly, we have latent heat of vaporization $h_{fg} = h_g - h_f$, and entropy of vaporization $s_{fg} = s_g - s_f$.

State S below the melting point temperature $T_s = T_{f_1}$ is a *subcooled solid state*. State G above the boiling point temperature $T_{f_2} = T_g$ is a *superheated vapour state*. Any liquid state F between f_1 and f_2 below the boiling point temperature is a *subcooled liquid state*. Any state A in the liquid plus vapour $(F + G)$ region represents a mixture of vapour at g and liquid at f_2. The position of A is governed by the quality or dryness fraction x of vapour and $(1 - x)$ of liquid.

Consider now the reverse process of cooling of superheated vapour at G. The processes followed will be *desuperheating* from G to g, *condensation* from g to f_2, *subcooling* from f_2 to f_1, *freezing* from f_1 to s, and subcooling from s to S.

Consider now that the heating of solid is carried out at lower pressures. As the pressure is lowered, there is a marked decrease in boiling point temperature, and an increase in volume and enthalpy accompanying vaporization. This continues until *triple point pressure* p_{tp} and temperature T_{tp} are reached. On the *triple point line*, all the saturation states s, f_1, f_2 and g lie. Thus, all the three phases exist in equilibrium at the triple point. The triple point for water is at 273.16 K (0.01°C), and 0.006112 bar.

Below the triple point pressure, saturated solid at s on heating directly changes into saturated vapour at g. This is called *sublimation*. Similarly, saturated vapour at g on cooling directly changes to saturated solid at s. This is called *ablimation* or *freeze-condensation*.

Consider now the heating at pressures above atmospheric. In general, at higher pressures, the boiling point increases, and there is a marked decrease in v_{fg}, h_{fg} and s_{fg}. At a certain high pressure, v_{fg}, h_{fg} and s_{fg} become zero. This is referred to as the *critical point*. The properties at this point are denoted as critical pressure p_c, critical temperature T_c and critical volume v_c. The same for water are:

$$p_c = 221.2 \text{ bar}, \; T_c = 647.3 \text{ K}, \; v_c = 0.00317 \text{ m}^3/\text{kg}$$

At pressures above critical, there is no definite transition between liquid and vapour phases.

Note that the gaseous state G^{id} at low pressures (tending to zero) and high temperatures represents the *ideal* or *perfect gas* state following the equation $pv = RT$.

1.9 HEAT EXCHANGE PROCESSES

Heat is normally absorbed or rejected by a working substance at a constant pressure. It has been seen that when a working substance exchanges heat, then either

(i) the temperature of the substance changes and the substance remains in a single phase, or

(ii) the temperature of the substance remains constant but a phase change occurs.

The heat transferred without a phase change results in a temperature rise and is called *sensible heat*, and that transferred resulting in a change of phase at constant temperature is called *latent heat*.

Sensible heat is measured by the expression

$$Q_S = mC_p\Delta T$$

where m is the mass, C_p is the specific heat at constant pressure, and ΔT is the temperature change of the working substance. In general, chilled water or salt solutions of either NaCl or $CaCl_2$ called *brines* are used as carriers of refrigeration for the absorption of heat. Chilled water is used for air conditioning in central air-conditioning plants. Brines have freezing point lower than 0°C and are, therefore, used as coolants in applications below 0°C refrigeration temperature, such as in cold storages, ice plants, skating rinks, etc.

Latent heat exchange processes correspond to those of melting, evaporation and sublimation, and vice versa, viz., fusion, condensation and desublimation and the heat transferred is measured by

$$Q_L = m\Delta h$$

where Δh is the latent heat for the corresponding process, e.g., h_{fg} for latent heat of vaporization, h_{sg} for sublimation and h_{sf} for fusion. The subscripts s, f and g denote, solid, liquid and gas respectively. Evaporation is the most commonly used method in refrigeration for the absorption of heat. Sublimation is used in a process called *freeze drying*. In this, the product is first frozen and cooled until it reaches below its triple point temperature. It is then placed in a chamber which is evacuated to a pressure sufficiently below the triple point pressure. Heat is then supplied to the product and in a *freeze-condenser* the sublimated vapours are condensed at a temperature of about – 40°C. The food thus dried generally retains its original flavour and value and can be reconstituted into its original condition by the addition of water. Freeze drying is used in the manufacture of certain high-value food products and medicines.

Wide use is made of the melting of ice in homes, stores, transports, etc., for the preservation of products at refrigerated temperatures. The main drawbacks of ice are the temperature limitation of 0°C and the spoilage caused by the melted water. These drawbacks can be avoided by the use of solid carbon-dioxide, also called *dry ice*, which sublimates at atmospheric pressure. Dry ice is rather widely used in refrigerated air transport. In recent times, liquid nitrogen has also become quite popular in transport refrigeration.

Example 1.2 100 kg of ice at –5°C is placed in a bunker to cool some vegetables. 24 hours later, the ice has melted into water at 10°C. What is the average rate of cooling in kJ/h and TR provided by the ice? Given:

Specific heat of ice, C_{p_s} = 1.94 kJ/kg.K
Specific heat of water, C_{p_f} = 4.1868 kJ/kg.K
Latent heat of fusion of ice at 0°C, h_{sf} = 335 kJ/kg.

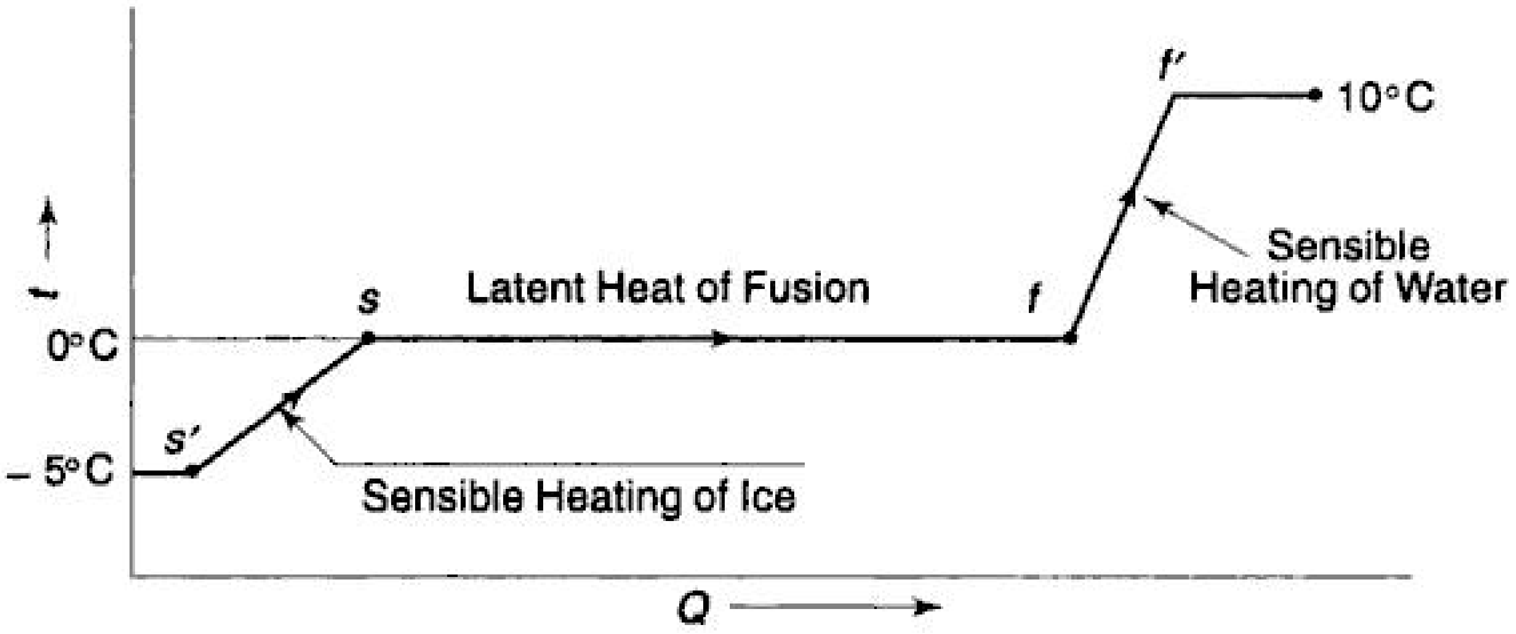


Fig. 1.8 Figure for Example 1.2

Solution The change of temperature with heat supplied is shown in Fig. 1.8.

$$Q = m\,[C_{p_s}(t_s - t_{s'}) + h_{sf} + C_{p_f}(t_{f'} - t_f)]$$
$$= 100\,[1.94\,(0 + 5) + 355 + 4.1868\,(10 - 0)] = 38{,}660 \text{ kJ}$$

$$\text{Capacity} = \frac{38{,}660}{24} = 1611 \text{ kJ/h}$$
$$= \frac{1611}{12{,}660} = 0.127 \text{ TR}$$

1.10 PRODUCTION OF LOW TEMPERATURES

The various principles and processes involved in the production of low temperatures are as follows:

(i) Throttling expansion of a liquid with flashing
(ii) Reversible adiabatic expansion of a gas
(iii) Irreversible adiabatic expansion (throttling) of a real gas
(iv) Thermoelectric cooling
(v) Adiabatic demagnetization

1.10.1 Expansion of a Liquid with Flashing

Consider the throttling of a saturated liquid initially at 1 at pressure p_1 as shown in Fig. 1.9. The state after expansion is at 2. The process is accompanied by an increase in entropy along with a drop in pressure due to which the volume increases and a part of the liquid is vaporized, thus cooling the remaining liquid. The expansion, with the vaporization of a part of the liquid causing the lowering of its temperature, is called *flashing*. This method is most commonly used for obtaining low temperatures. It will be seen that for the process

$$h_1 = h_2 = h_{f_2} + x_2 h_{fg_2} \qquad v_2 = v_{f_2} + x_2 v_{fg_2}$$

It may be noted that the temperature will similarly drop from T_1 to T_2 if the liquid had isentropically expanded from 1 to 2′. But the isentropic expansion of a liquid is not employed in refrigeration for reasons explained in Sec. 3.1.

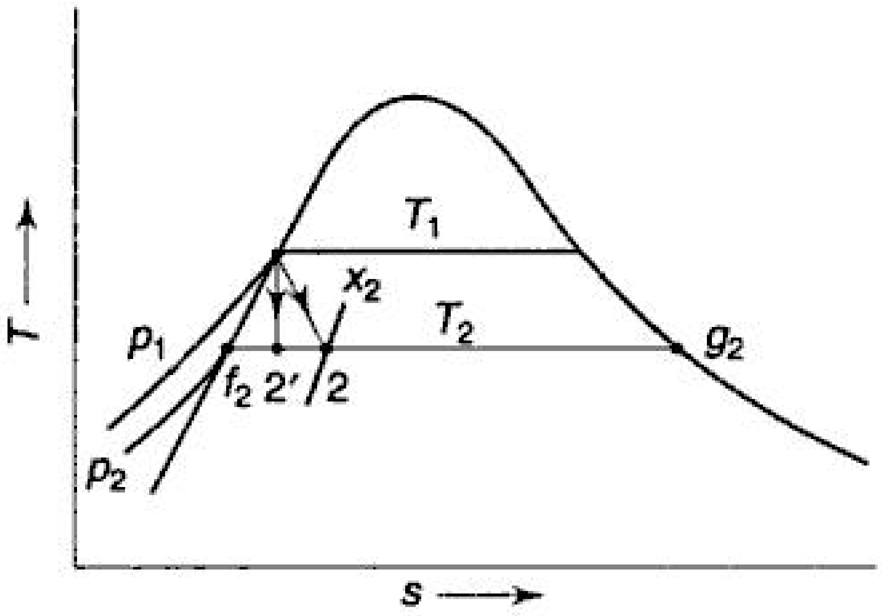


Fig. 1.9 Reversible and Irreversible Adiabatic Expansion of a Liquid

1.10.2 Reversible Adiabatic Expansion of a Gas

This method is used with permanent gases such as air. The gas is initially compressed and cooled and then expanded reversibly in an adiabatic reciprocating or turbo-type expander, thus doing external work. The reversible adiabatic expansion of an ideal gas from p_1 to p_2 follows the path $pV^\gamma = C$. The work done is given by the shaded area shown in Fig. 1.10. The expression for working is obtained by adding the flow work components to the non-flow work. Thus, we have for specific work

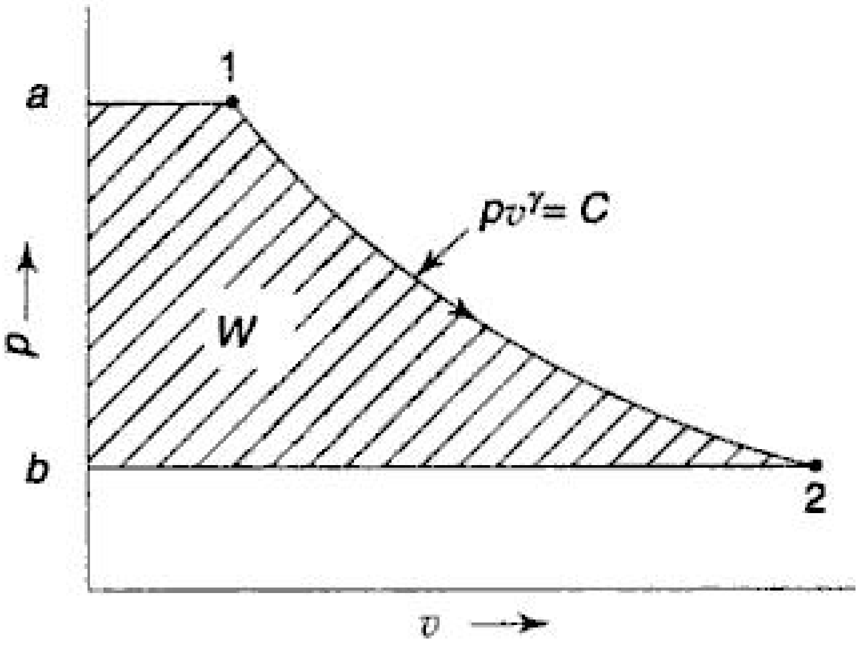


Fig. 1.10 Work done in a Reversible Adiabatic-Expansion Process

$$w = \oint p\,dv = \int_a^1 p\,dv + \int_1^2 p\,dv + \int_2^b p\,dv + \int_b^a p\,dv = -\oint_1^2 v\,dp$$

which gives on integration for a perfect gas

$$w = -\int v_1 \left(\frac{p_1}{p} \right)^{1/\gamma} \mathrm{d}p = \frac{\gamma}{\gamma - 1} \left(p_1 v_1 - p_2 v_2 \right)$$

Also since

$$q = 0$$

we obtain from the steady-state steady-flow energy equation

$$h_1 - h_2 = w = -\int v \mathrm{d}p = \frac{\gamma}{\gamma - 1} (p_1 v_1 - p_2 v_2)$$

which for an ideal gas, on putting $\frac{\gamma R}{\gamma - 1} = C_p$, becomes

$$w = h_1 - h_2 = C_p \left(T_1 - T_2 \right)$$

The temperature after expansion is given by

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

1.10.3 Irreversible Adiabatic Expansion (Throttling) of a Real Gas

Herein again, the initially compressed and cooled gas is expanded, but irreversibly, through a throttle device with restriction to flow. It has been shown that for this process

$$h_1 = h_2$$

and for a perfect gas $T_1 = T_2$, since enthalpy is a function of temperature only.

Real gases, however, show a departure from ideal gas behaviour and usually produce a substantial decrease in temperature under certain conditions, viz., initial low temperature and high pressure of the gas. The decrease in temperature dT corresponding to a drop in pressure dp, defined by

$$\left(\frac{\partial T}{\partial p} \right)_h = \delta \quad \text{or} \quad \mu_J$$

is known as the *Joule-Thomson coefficient*. For an ideal gas this coefficient is zero. For a real gas, we have:

For cooling	δ is positive	Initial state of gas at high pressure and low temperature
For no change in temperature	$\delta = 0$	Ideal gas behaviour
For heating	δ is negative	Initial state of gas at high temperature.

1.10.4 Thermoelectric Cooling

In 1834, Jean Peltier, a French watchmaker and an amateur scientist, discovered that cooling is produced at one junction of two dissimilar metals, if a current is passed through them. Simultaneously, heat is produced at the other junction (Fig. 1.11). The Peltier heats absorbed at the cold end and rejected at the hot end are given by

$$Q_c = \Pi_c I$$
$$Q_h = \Pi_h I$$

where Π_c and Π_h are the *Peltier coefficients* which are functions of temperatures T_c and T_h of the cold and hot ends respectively. It may be seen that if T_h is maintained at ambient temperature, T_c will be lower than the ambient temperature. It is also to be noted that which of the junctions or ends will become cold or hot depends on the direction of flow of the current.

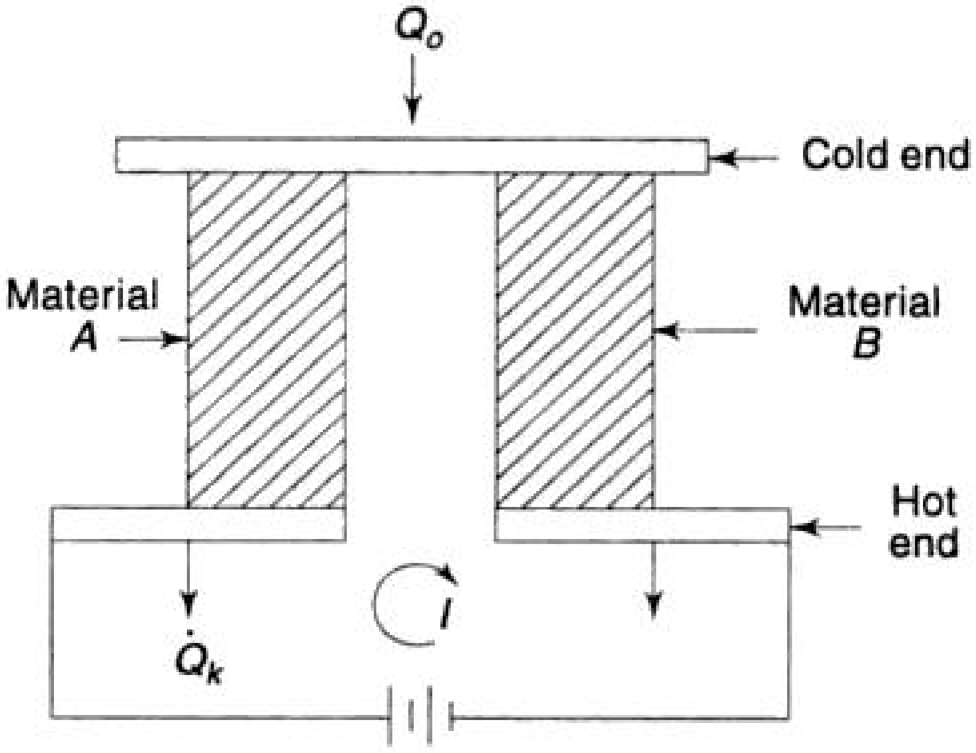


Fig. 1.11 Peltier Effect

The phenomenon is called the *Peltier effect*. The actual cooling produced at the cold end would, however, be reduced due to irreversible processes of Joule and Fourier heat transfers. The energy balance of the system is

$$\dot{Q}_k - \dot{Q}_o = EI$$

where I is the current, E is the emf applied and $\dot{Q}_o$ and $\dot{Q}_k$ are the actual amounts of heat flows at the cold and hot ends respectively.

1.10.5 Adiabatic Demagnetization

In 1926, Giaque and Deby independently proposed that cooling could be produced by the adiabatic demagnetization of a paramagnetic salt. Figure 1.12 provides a schematic representation of such a process.

The paramagnetic salt is suspended in a tube containing low pressure gaseous helium as exchange gas to provide thermal communication with the surrounding bath of liquid helium. The liquid helium bath and the salt are first cooled to about 1 K by pumping helium to the lowest practical pressure. A magnetic field is applied, causing magnetization of the salt. The heat produced is removed by the helium bath such that the temperature of the bath again approach 1 K. Next, the exchange gas is removed by pumping and the magnetic field is turned off. The temperature of the salt then decreases as a result of *adiabatic demagnetization*.

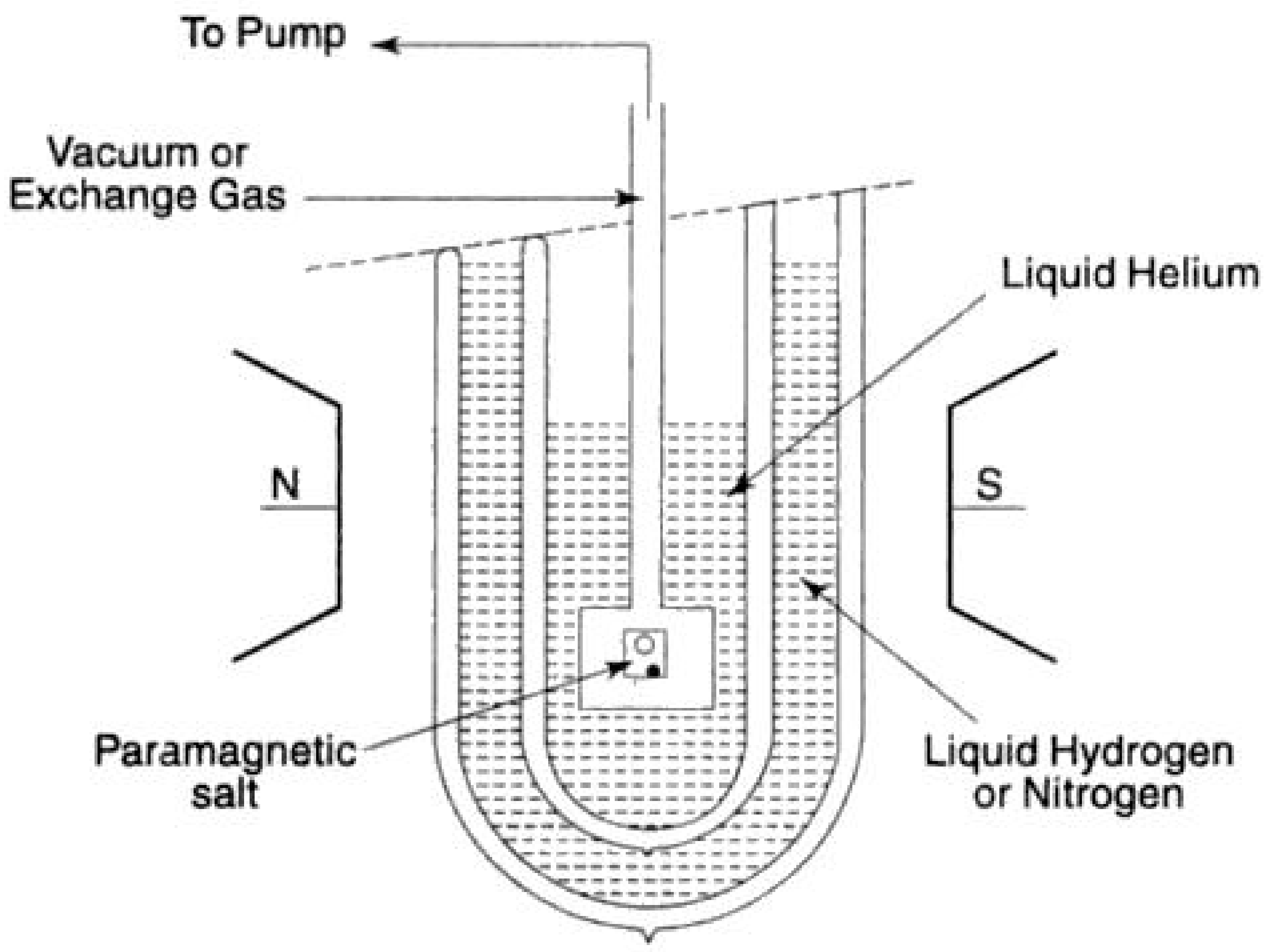


Fig. 1.12 Arrangement for Adiabatic Demagnetization of a Paramagnetic Salt

1.11 SATURATION PRESSURE VERSUS SATURATION TEMPERATURE RELATIONSHIP

The whole matter of phase equilibrium is best summarised in phase diagram in Fig. 1.13 between saturation pressure p^{sat} and saturation temperature T^{sat} of a substance. Any point on a line in this diagram represents two phases in equilib-

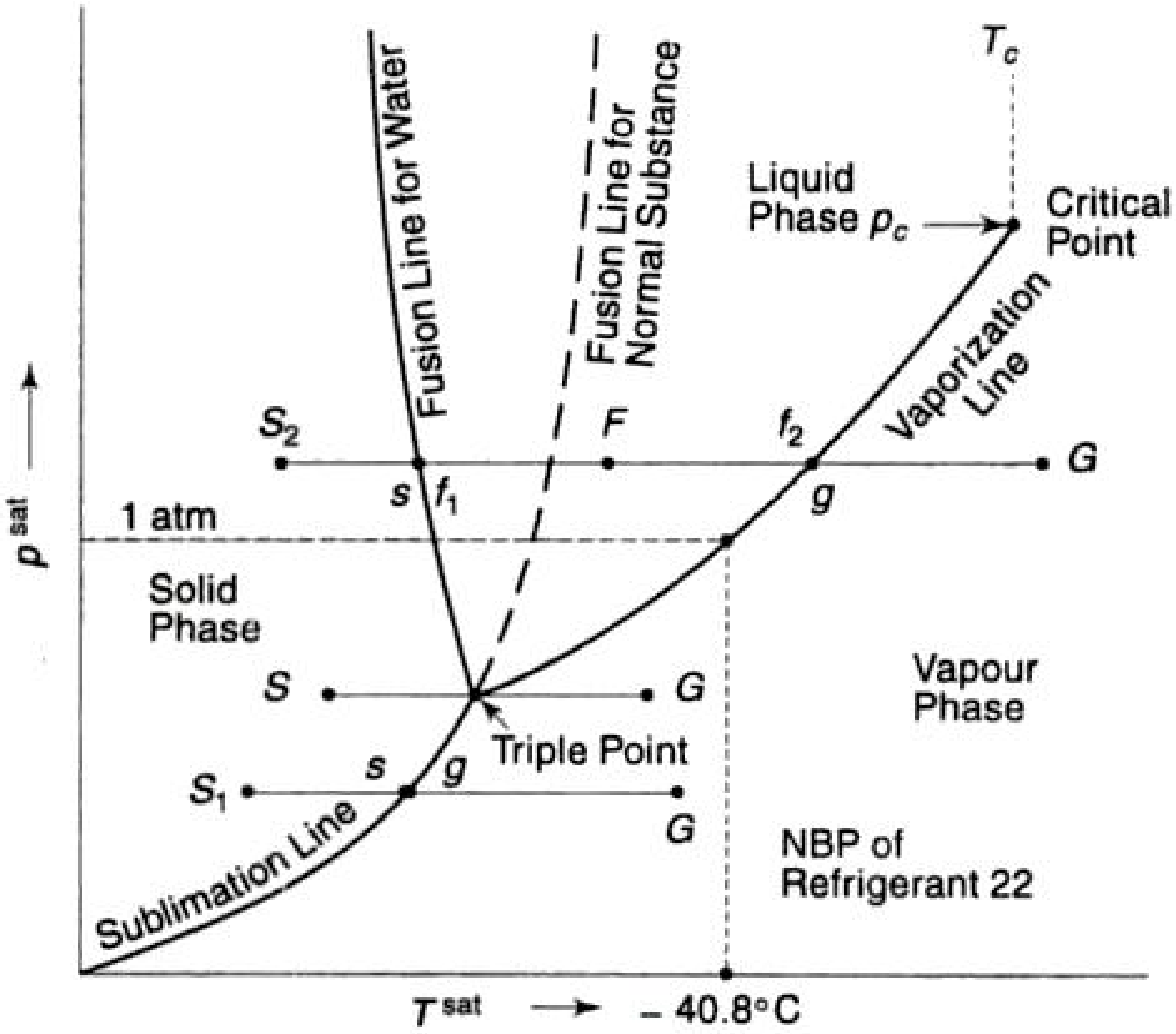


Fig. 1.13 Saturation Pressure Versus Saturation Temperature Phase Diagram of a Pure Substance

rium state such as s and f_1 on *fusion line*, f_2 and g on *vaporization line* and s and g on *sublimation line*. The triple point is one unique point where all saturation states s, f_1, f_2 and g conjoin. The vaporization line begins at the triple point, and ends at the critical point. It is called the *vapour pressure curve*.

The saturation temperature at 1 atm pressure is the *normal boiling point* of the substance. The figure shows the N.B.P. of one of the most commonly used refrigerants, viz., Refrigerant 22 ($CHClF_2$) as –40.8°C. Equation recommended by Reynolds[14] represents the saturation pressure versus saturation temperature data of Refrigerant 22 very accurately.

$$\ln p^{\text{sat}} = 71.55415 - \frac{4{,}818.96}{T^{\text{sat}}} - 7.861 \ln T^{\text{sat}} + 9.0807 \times 10^{-3}\, T^{\text{sat}} + 0.445747\, \frac{\left(381.17 - T^{\text{sat}}\right)}{T^{\text{sat}}} \ln \left(381.17 - T^{\text{sat}}\right) \qquad (1.10)$$

Herein, the pressure is in Pa.

Example 1.3 Find saturation pressure of Refrigerant 22 at 40°C (313 K).

Solution Substituting values in Eq. (1.10), we obtain

$$\ln p^{\text{sat}} = 71.55415 - 15.396 - 45.1711 + 2.8423 + 0.4099 = 14.236$$

$$\Rightarrow \qquad p^{\text{sat}} = 1,527,660 \text{ Pa}$$

Note ✍: *ln p^{sat} versus $1/T^{\text{sat}}$ relationship can be simplified; and can be expressed in the form of Antonie equation as follows in Eq. (1.11)*

$$\ln p^{\text{sat}} = A - \frac{B}{T^{\text{sat}}} \qquad (1.11)$$

which is a straight line, and is quite accurate.

Example 1.4 Using the vapour pressure data of Freon 22 and propane at 40°C and –30°C given below, find the values of the constants of the Antonie equation for the two refrigerants, and verify their validity for pressure at 5°C.

t^{sat}	T^{sat}	p^{sat}	
		Freon 22	*Propane*
°C	K	Pa	
40	313.15	1,533,500	1,366,400
5	278.15	584,000	547,750
–30	243.15	163,500	166,400

Solution Substituting values and solving, we get the Antonie equations:
For Freon 22

$$\ln p^{\text{sat}} = 22.01864 - \frac{2.43492 \times 10^3}{T^{\text{sat}}}$$

Putting $T^{\text{sat}} = 278.15$ K, we get

$$p^{\text{sat}} = 5.765 \times 10^5 \text{ Pa}$$

The error in comparison to 5.84×10^5 Pa is 1.28% only.

For Propane

$$\ln p^{\text{sat}} = 21.50251 - \frac{2.309424 \times 10^3}{T^{\text{sat}}}$$

At $T^{\text{sat}} = 278.15$ K, $p^{\text{sat}} = 5.402 \times 10^5$ Pa

The error in comparison to 547,750 Pa is 1.38% only.

Note ✍: *If a number of data points are used, and least squares method is employed for finding A and B, the accuracy would still be better.*

1.12 THE GASEOUS PHASE: EQUATION OF STATE

Pressure p, specific volume v and temperature T are the three measurable properties of a substance. The equation expressing their relationship in the gaseous phase is called the *equation of state*. This functional relationship between p-v-T of a gas can be either theoretical, or generalized or an empirical equation fitted from experimental data.

The simplest theoretical equation of state is the *ideal* or *perfect gas* equation representing behaviour of a gas at low pressures (*tending to zero*) and high temperatures such as at point G^{id} in Figs 1.6 and 1.7. This equation is

$$v^{id} = \frac{RT}{p}$$

where $R = \overline{R}/M$ is the *gas constant* for the particular gas, $\overline{R}$ is the *universal gas constant* having the value 8.3143 kJ/kmol K, and M is the molecular mass of the substance.

In addition, the internal energy and enthalpy of an ideal gas are functions of temperature only. These are given by,

$$du = C_{v_o}\, dT\,,\ u_2 - u_1 = C_{v_o}(T_2 - T_1)$$
$$dh = C_{p_o}\, dT,\ h_2 - h_1 = C_{p_o}(T_2 - T_1)$$

where C_{v_o} and C_{p_o} are *zero-pressure constant* volume and *zero-pressure constant pressure* specific heats. It can be shown that

$$C_{p_o} - C_{v_o} = R$$

and the ratio of specific heats C_{p_o}/C_{v_o} is denoted as γ. Their values for air as a perfect gas are:

$$C_{p_o} = 1.005 \text{ kJ/kg. K},\ C_{v_o} = 0.718 \text{ kJ/kg.K},\ R = 0.287 \text{ kJ/kg. K}$$
$$\gamma = 1.4,\ M = 28.966$$

For a *real gas*, the actual volume can be expressed by the general relation

$$v = v^{\text{real}} = z\,\frac{RT}{p}$$

where $z = v^{\text{real}}/v^{id} = pv/RT$ is called the *compressibility* of the gas.

Many different equations of state have been proposed to represent the real volume of gases. In 1949, Redlich-Kwong[12] (R-K) proposed the equation as follows in Eq. (1.12)

$$p = \frac{\bar{R}\,T}{\bar{v} - b} - \frac{a}{\sqrt{T}\,(\bar{v}^2 + \bar{v}\,b)} \tag{1.12}$$

in which $\bar{v}$ is the *molar volume* (volume of M kg of gas). Its constants a and b found from conditions of critical isotherm at the critical point are expressed in terms of critical constants T_c and p_c as follows:

$$a = 0.42748\,\frac{R^2\,T_c^{5/2}}{p_c},\; b = 0.08664\,\frac{R\,T_c}{p_c}$$

Often constants a and b are replaced by constants A and B written as:

$$A^2 = \frac{a}{R^2\,T^{2.5}},\; B = \frac{b}{RT},\; \text{and } h = \frac{b}{v} = \frac{b}{zRT/p} = \frac{Bp}{z}$$

so that the R-K equation can be written in terms of z in place of v as follows in Eq. (1.13)

$$\begin{aligned} z &= \frac{1}{1-h} - \frac{a}{b\,RT^{1.5}}\left(\frac{h}{1+h}\right) \\ &= \frac{1}{1-h} - \frac{A^2}{B}\left(\frac{h}{1+h}\right) = \frac{pv}{RT} \end{aligned} \tag{1.13}$$

A modification of the R-K equation is Peng-Robinson[11] (P-R) equation as given in Eq. (1.14)

$$p = \frac{RT}{v-b} - \frac{a}{v^2 + 2bv - v^2} \tag{1.14}$$

where
$$a = \frac{0.45724\,R^2T_c^2}{p_c}\left[1 + f(\omega)\left\{1 - \left(\frac{T}{T_c}\right)^{1/2}\right\}\right]^2$$

$$b = \frac{0.0778\,RT_c}{p_c}$$

$$f(\omega) = 0.37464 + 1.54226\,\omega - 0.26992\,\omega^2$$

$$\omega = \textit{Accentric factor} = -\,1.0 - \log\,(p_r^{\text{sat}})_{T_r = 0.7}$$

$$T_r = \text{Reduced temperature} = \frac{T}{T_c}$$

$$p_r = \text{Reduced pressure} = \frac{p}{p_c}$$

Example 1.6 **Calculation of Latent Heat of Vaporization of R22.**
From the following handbook data for R22,

t °C	p bar	v_f L/kg	v_g m^3/kg
–4	4.358	0.77	0.0536
0	4.976	0.778	0.0471
+4	5.657	0.787	0.0416

calculate the latent heat of vaporization of R22 of 0°C.

Solution

$$\text{At } 0°\text{C}, \quad \left(\frac{\Delta p}{\Delta T}\right)^{\text{sat}} = \frac{129.9 \text{ kPa}}{8°\text{C}} = 16.237 \text{ kPa/°C}$$

$$(h_{fg})0°\text{C} = \left(\frac{\Delta p}{\Delta T}\right)^{\text{sat}} T(v_g - v_f)$$

$$= 16.237\ (273.16)\ (0.0471 - 0.00077)$$

$$= 205.48 \text{ kJ/kg}$$

Note ✍: *The value of dp^{sat}/dT^{sat} at 0°C can also be found by differentiation from Eq. (1.10).*

1.14 PROPERTY RELATIONS

$\delta Q = \mathrm{d}U + \delta W$ is the first law equation which is applicable to *all process* in a closed system. For the particular case of reversible processes in a closed system, since $\delta W = p\mathrm{d}V$ and $\delta Q = T\mathrm{d}S$, this equation takes the form

$$T\mathrm{d}S = \mathrm{d}U + p\mathrm{d}V \tag{1.17a}$$

$$T\mathrm{d}S = \mathrm{d}H - V\mathrm{d}p \tag{1.17b}$$

By dividing both sides by mass m, these can also be written on the basis of a unit mass of the substance as given below in Eqs (1.18a) and (1.18b)

$$T\mathrm{d}s = \mathrm{d}u + p\mathrm{d}v \tag{1.18a}$$

$$T\mathrm{d}s = \mathrm{d}h - v\mathrm{d}p \tag{1.18b}$$

These are the two well-known T–ds equations. Although derived for a reversible process in a closed system, these are actually in terms of properties of a system/substance. Hence, these are applicable to all processes whether reversible or irreversible, and whether in a closed system or in an open system. These equations are referred to as *property relations*, and are used to evaluate the changes in entropy in terms of changes in other properties employing the relationships as illustrated in Eqs. (1.19a) and (1.19b)

$$\mathrm{d}s = \frac{1}{T}\mathrm{d}u + \frac{p}{T}\mathrm{d}v \tag{1.19a}$$

$$ds = \frac{1}{T}dh - \frac{v}{T}dp \tag{1.19b}$$

1.15 THERMODYNAMIC PROPERTIES OF REFRIGERANTS[2]

For establishing the thermodynamic properties of refrigerants, and for that matter, for any pure substance, the following minimum experimental data/correlations are required:

(i) p^{sat} versus T^{sat}, or T^{sat} versus p^{sat}.
(ii) p-v-T data or equation of state for gaseous phase.
(iii) Liquid density ρ or specific volume v_f.
(iv) Liquid specific heat C_f.
(v) Zero-pressure (ideal gas) constant pressure specific heat C_{p_o} or constant volume specific heat C_{v_o} $(= C_{p_o} - R)$ of the gaseous phase.

Then the latent heat is calculated from the Clapeyron equation. The only other properties that need to be calculated are internal energy, enthalpy and entropy. For the purpose, the relations given in Eqs (1.20), (1.21), (1.22), and (1.23) for changes in u, h and s at constant temperature in the gaseous phase are used:

$$(u_2 - u_1)_T = \int_1^2 \left[T\left(\frac{\partial p}{\partial T}\right)_v - p \right] dv \tag{1.20}$$

$$(h_2 - h_1)_T = \int_1^2 \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right] dp \tag{1.21}$$

$$(s_2 - s_1)_T = -\int_1^2 \left(\frac{\partial v}{\partial T}\right)_v dp \tag{1.22}$$

$$(s_2 - s_1)_T = \int_1^2 \left(\frac{\partial p}{\partial T}\right)_v dv \tag{1.23}$$

Note that Eq. (1.20) for internal energy change requires a p-explicit equation of state, where as Eq. (1.21) for enthalpy change requires a v-explicit equation of state. If only a p-explicit equation of state is available, then enthalpy change can be found from internal energy change using the relationship:

$$(h_2 - h_1)_T = (u_2 - u_1)_T + (p_2 v_2 - p_1 v_1)_T \tag{1.24}$$

1.15.1 Enthalpy Calculations

Figure 1.14 illustrates the method of calculation of enthalpy with the help of pressure-enthalpy ($p - h$) diagram. Since there is no absolute value of enthalpy, and only differences in enthalpies of state points are required in calculations, a *reference state* has to be chosen for the purpose to which an arbitrary value of enthalpy is assigned. In Fig. 1.14, the reference state chosen is that of saturated liquid at point 1. In the case of water, point 1, usually, is the saturated liquid state at its triple point (0.01°C) to which a value of $h_1 = h_{f_1} = 0$ kJ/kg is assigned. In the case of refrigerants, the reference state chosen is that of saturated liquid at 0°C. And since the refrigerants work at temperatures below 0°C also, in order to avoid negative values of enthalpies in calculations, the value of enthalpy assigned at the reference temperature is, usually, $h_1 = h_{f_1} = 200$ kJ/kg. Note that the pressure at 1 is $(p^{\text{sat}})_{0°\text{C}}$.

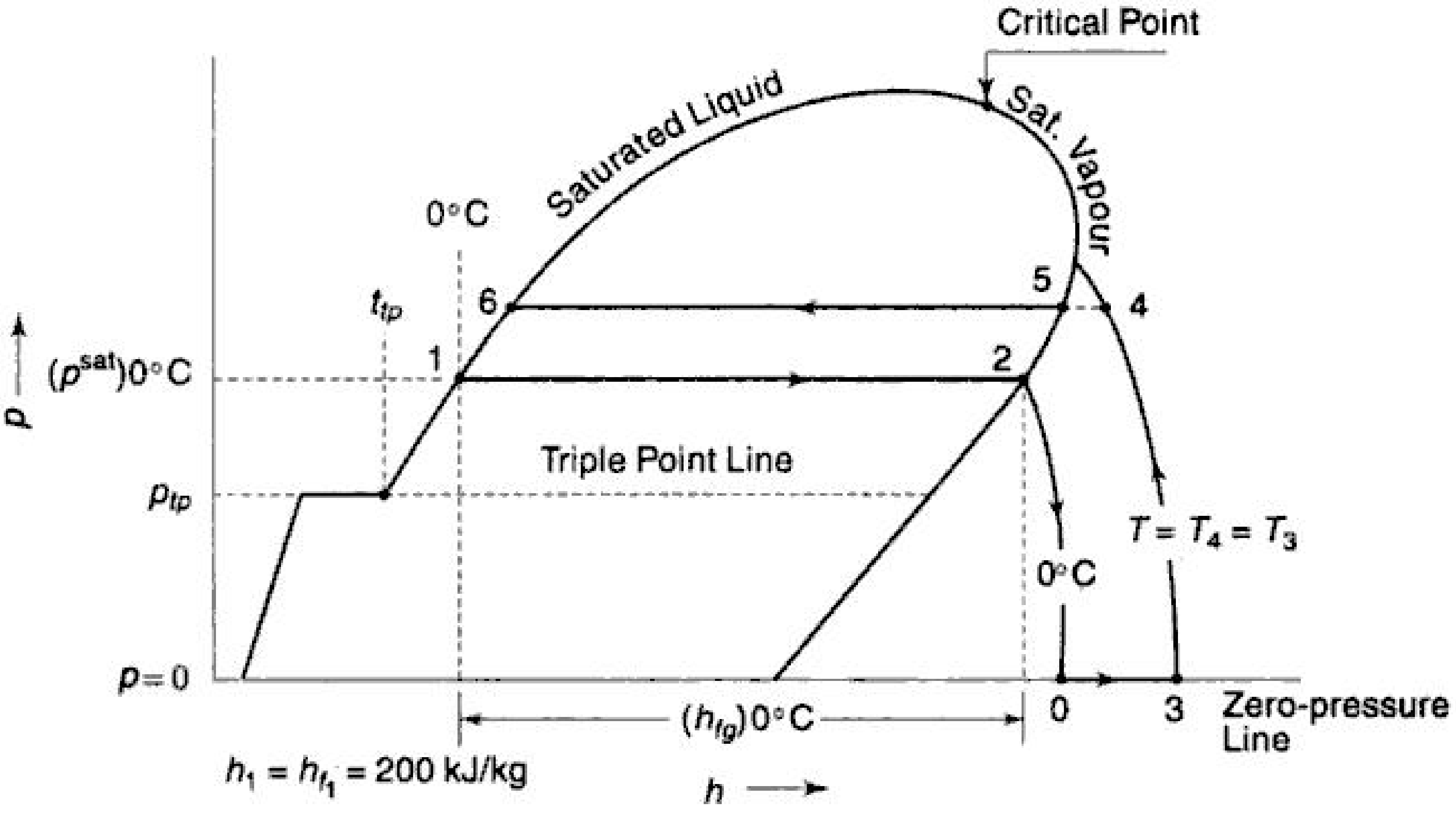


Fig. 1.14 Figure Demonstrating Method of Calculation of Enthalpy

Now the enthalpy of saturated vapour at 2 at the reference temperature and pressure is given by

$$h_2 = h_1 + (h_{fg})_{0°\text{C}} = 200 + (h_{fg})_{0°\text{C}}$$

Then, for the calculation of the enthalpy of vapour at any state 4 at temperature T_4 and pressure p_4, the path followed is from 2 to 0, 0 to 3 and 3 to 4. State 0 is at the same temperature as 2 but at zero pressure. Similarly, state 3 is at the same temperature as 4 but at zero pressure. Both states 0 and 3 are, therefore, ideal gas states at temperatures 0°C and T respectively. It is seen that

$$h_4 = h_2 + (h_0 - h_2) + (h_3 - h_0) + (h_4 - h_3)$$

where

$$h_3 - h_0 = \int_{T_3}^{T_4} C_{p_o}\, \mathrm{d}T$$

Now, $(h_4 - h_3)$ represents the change in enthalpy at constant temperature $T_4 = T_3$ as a result of change in pressure from 0 to $p_4 = p$. Similarly, $(h_2 - h_0)$ represents the change in enthalpy at 0°C as a result of change in pressure from 0 to p_2. These represent differences in enthalpies of real gas and ideal gas at the same temperature. The difference $h^R = (h - h^{id})_T$ is termed *residual enthalpy*, and can be found either from Eq. (1.21) or Eq. (1.24) by integrating between pressure limits of $p = 0$ to p, or between volume limits if $v = \infty$ (at $p = 0$) to v.

The enthalpy of saturated vapour at any point 5 can be similarly calculated. Then, the enthalpy of saturated liquid at 6 can be found from

$$h_6 = h_5 - (h_{fg})_T$$

or from

$$h_6 = h_1 + \int_{T_1}^{T_6} c_f \, \mathrm{d}T$$

Example 1.7 Calculation of Enthalpy of Vapour Using R-K Equation of State

Using Redlich-Kwong equation of state, calculate the enthalpy of superheated vapour of propane at 2 MPa and 350 *K*. Take $h_f = 0$ at 200 K ($p^{\text{sat}} = 19.97$ kPa) as the reference state. Critical data for propane are:

$$T_c = 369.8 \text{ K}, p_c = 4.236 \text{ MPa}, v_c = 0.005066 \text{ m}^3/\text{kg}$$

Assume zero-pressure constant pressure specific heat of propane as constant as

$$C_{p_o} = 1.6794 \text{ kJ/kg.K}$$

Solution Refer to Fig. 1.14. Reference state 1 is at 200 K, and not 0°C. From Clapeyron equation, we find:

$$(h_{fg})_{200\text{ K}} = 456.24 \text{ kJ/kg}$$

Hence, $$h_2 = h_1 + (h_{fg})_{200\text{ K}} = 456.24 \text{ kJ/kg}$$

Gas constant for propane

$$R = \frac{\overline{R}}{M} = \frac{8.3143}{44} = 0.18855 \text{ kJ/kg.K}$$

Constants of Redlich Kwong equation

$$a = 0.42748\, R^2 \frac{T_c^{5/2}}{p_c} = 0.42748 \frac{(188.55)^2\,(369.8)^{5/2}}{4.236 \times 10^6}$$
$$= 9416$$

$$b = 0.08664 \frac{RT_c}{p_c} = 0.08664 \frac{(188.55)\,(369.8)}{4.236 \times 10^6}$$
$$= 1.426 \times 10^{-3}$$

Substituting values in R-K equation, we get

$$v_2 = 1.865 \text{ m}^3/\text{kg}$$
$$v_4 = 0.02534 \text{ m}^3/\text{kg}$$

$$z_2 = \frac{p_2 v_2}{RT_2} = 0.9876$$

$$z_4 = \frac{p_4 v_4}{R\,T_4} = 0.7677$$

Redlich-Kwong[12] derived the following expression for residual enthalpy

$$\frac{h^R}{RT} = \frac{h - h^{id}}{RT} = -\frac{3}{2}\frac{A^2}{B}\ln\left[1 + \frac{Bp}{z}\right] + (z - 1) \tag{1.25}$$

The constants at points 2 and 4 are:

$$A_2^2 = \frac{a}{R^2\,T_2^{2.5}} = 4.692 \times 10^{-7}$$

$$A_4^2 = \frac{a}{R^2\,T_2^{2.5}} = 1.1582 \times 10^{-7}$$

$$B_2 = \frac{b}{RT_2} = 3.782 \times 10^{-8}$$

$$B_4 = \frac{b}{RT_5} = 2.16 \times 10^{-8}$$

Residual enthalpies at points 2 and 4 are:

$$\left(\frac{h - h^{id}}{RT}\right)_2 = -\frac{3}{2}\cdot\frac{4.692 \times 10^{-7}}{3.782 \times 10^{-8}}\ln\left[1 + \frac{3.782 \times 10^{-8} \times 19.97 \times 10^3}{0.9876}\right]$$

$$+ (0.9876 - 1) = -0.02663$$

$$\Rightarrow \qquad (h - h^{id})_2 = -(0.18855)\,(200)\,(0.02663) = -1.0 \text{ kJ/kg}$$

$$\left(\frac{h - h^{id}}{RT}\right)_4 = -\frac{3}{2}\cdot\frac{4.692 \times 10^{-7}}{3.782 \times 10^{-8}}\ln\left[1 + \frac{3.782 \times 10^{-8} \times 2 \times 10^6}{0.7677}\right]$$

$$+ (0.7677 - 1) = -0.6723$$

$$\Rightarrow \qquad (h - h^{id})_4 = -(0.18855)\,(350)\,(0.6723) = -44.35 \text{ kJ/kg}$$

Change in enthalpy at zero pressure

$$h_3 - h_0 = \int_{200}^{350} C_{p_o}\,dT = 1.6794\,(350 - 200) = 251.91 \text{ kJ/kg}$$

Enthalpy of superheated propane vapour at 4

$$\begin{aligned} h_4 &= h_2 + (h_0 - h_2) + (h_3 - h_0) + (h_4 - h_3) \\ &= h_2 - (h - h^{id})_2 + (h_3 - h_0) + (h - h^{id})_4 \\ &= 456.24 + 1.0 + 251.91 - 44.35 \\ &= 664.8 \text{ kJ/kg} \end{aligned}$$

Note ✍: *The value from table is 641.6 kJ/kg. The error is due to the following reasons:*

(i) Redlich -Kwong equation has been used in place of an equation fitting actual p-v-T data.

(ii) A constant and approximate value of Cp_o has been used.

1.15.2 Enthalpy from Residual Internal Energy

If only a *p*-explicit equation is available, the enthalpy change at constant temperature between points 2 and 0, and 4 and 3 can be calculated by using Eq. (1.24). For the purpose, we have for *residual internal energy* from Eq. (1.20)

$$u^R = (u - u^{id})_T = \int_{v=\infty}^{v} \left(T \left(\frac{\partial p}{\partial T} \right)_v - p \right) \mathrm{d}v_T$$

$$= \int_{v}^{v=\infty} \left[p - T \left(\frac{\partial p}{\partial T} \right)_v \right] \mathrm{d}v_T$$

Using this expression, one can find u_0 and then h_0 from Eq. (1.24) at $t_0 = 0°\mathrm{C}$ ($T_0 = 273.15$ K) and $p_0 = 0$. Then, one can find the enthalpy at 4 using the relationship

$$h_4 = h_0 + (h_3 - h_0) + (h_4 - h_3)$$

$$= h_0 + \int_{T_o}^{T=T_3} C_{p0} \mathrm{d}T + \left|(u - u^{id})_T\right|_{v=\infty}^{v=v_4} + p_4 v_4 - p_3 v_3 \qquad (1.25a)$$

Putting $C_{p_o} = C_{v_o} + R$, and $p_3 v_3 = RT_3$ (ideal gas state at T_3), we have

$$h_4 = h_0 - RT_0 + p_4 v_4 + \int_{T_o}^{T=T_3} C_{v_o} \, \mathrm{d}T + \left|(u - u^{id})_T\right|_{v=\infty}^{v=v_4} \qquad (1.25b)$$

Note ✍: *Often, state 0 with enthalpy h_0 is employed as the new reference state for further calculation of enthalpies, and entropies in the entire vapour region.*

Example 1.8 **Residual Internal Energy from Modified Martin-Hou Equation**

Derive an expression for $(u - u^{id})_T$ from Eq. (1.15).

Solution

We have Eqs (1.26), (1.27), (1.28) by differentiating partially with respect to *T*

$$\left(\frac{\partial p}{\partial T} \right)_v = \frac{R}{v-b} + \sum_{i=2}^{5} \frac{1}{(v-b)^i} \left[B_i - C_i \frac{k}{T_c} e^{-kT/T_c} \right] \qquad (1.26)$$

$$\left[p - \left(\frac{\partial p}{\partial T} \right)_v T \right] = \cancel{\frac{RT}{v-b}} + \sum_{i=2}^{5} \frac{A_i + B_i/T + C_i e^{-kT/T_C}}{(v-b)^i}$$

$$- \cancel{\frac{RT}{v-b}} + \sum_{i=2}^{5} \frac{B_i/T - \frac{C_i kT}{T_c} e^{-kT/T_c}}{(v-b)^i} \qquad (1.27)$$

$$\int_{v}^{v=\infty}\left[p-\left(\frac{\partial p}{\partial T}\right)_v T\right]\mathrm{d}v = \int_{v}^{v=\infty}\sum_{i=2}^{5}\frac{A_i + C_i\, e^{-kT/T_c}\left(1+\dfrac{kT}{T_c}\right)}{(v-b)^i}$$

$$= \sum_{i=2}^{5} -\left[\frac{A_i + C_i\, e^{-kT/T_c}\left(1+\dfrac{kT}{T_c}\right)}{(-i+1)\,(v-b)^{i-1}}\right] \tag{1.28}$$

Example 1.9 Reference State Enthalpy of Freon 22 Vapour

Find the value of ideal gas enthalpy h_0 at $T_0 = 273.15$ K and $p_0 = 0$ of Freon 22 vapour using the modified Martin-Hou equation. T_c for Freon 22 is 369 K.

Solution

Refer to Fig. 1.14.

First we find $p_2 = (p^{\text{sat}})_{273.15\text{ K}} = 4.976 \times 10^5$ Pa from Eq. (1.10)

Next we find $v_2 = 0.0471$ m^3/kg at p_2 and T_2 from Eq. (1.15)

Then, $$h_2 = h_{f_1} + (h_{fg})_{273.15\text{ K}} = 200 + 205.48 = 405.48 \text{ kJ/kg}$$

$$u_2 = h_2 - p_2 v_2$$

$$= 405.48 - 497.6\,(0.0471) = 382.04 \text{ kJ/kg}$$

Now, substituting values of A_i, C_i, k, T_c, and b in Eq. (1.28), we have for $T_2 = 273.15$ K and $v_2 = 0.0471$ m^3/kg,

$$u_2 - u_2^{id} = u_2 - u_0 = -\,7020 \text{ J/kg} = -\,7.02 \text{ kJ/kg}$$

$$\Rightarrow \quad u_0 = u_2 + 7.02 = 389.06 \text{ kJ/kg}$$

$$\Rightarrow \quad h_0 = u_0 + R\,T_0 = 389.06 + 0.09615\,(273.15)$$

$$= 415.32 \text{ kJ/kg}$$

Example 1.10 Enthalpy of Freon 22 Vapour

Taking the value of reference state enthalpy h_0 as found in Example 1.9, calculate the enthalpy of Freon 22 superheated vapour at 1650 kPa and 95°C (368.15 K).

Solution

From Eq. (1.15), $v = 0.0187$ m^3/kg at 1650 kPa and 368.15 K.

Substituting values of constants, and v and T, we get:

$$\left|\left(u - u^{id}\right)_{368.15\,K}\right|_{v=\infty}^{v=0.0187} = -\,27{,}440 \text{ J/kg} = -\,27.44 \text{ kJ/kg}$$

The expression for C_{v_o} for Freon 22 from Appendix is as follows in Eq. (1.29)

$$C_{v_o} = C_1 + C_2\,T + C_3\,T^2 - \frac{C_4}{T}, \text{ J/kg.K} \tag{1.29}$$

where $C_1 = 117.767818,\ C_2 = 1.6997296$

$C_3 = -\,8.83043292 \times 10^{-4},\ C_4 = 3.32541759 \times 10^5$

$$\int_{273.15}^{368.15} C_{v_o}\, dT = 70{,}750 \text{ J/kg} = 70.75 \text{ kJ/kg}$$

$$h = h_0 - RT_0 + pv + \int_{T_o}^{T} C_{v_o}\, dT + \left|\left(u - u^{id}\right)_T\right|_{v=\infty}^{v} \tag{1.30}$$

$$= 415.32 - 0.09615\,(273.15) + 1650(0.0187) + 70.75 - 27.44$$

$$= 463.22 \text{ kJ/kg}$$

1.15.3 Entropy Calculations

The method of calculation of entropy is similarly illustrated in Fig. 1.15. The value of entropy assigned to the reference state of saturated liquid at 0°C is usually $s_1 = s_{f_1} = 1.0$ kJ/kg. K. Then, the entropy at 2 is

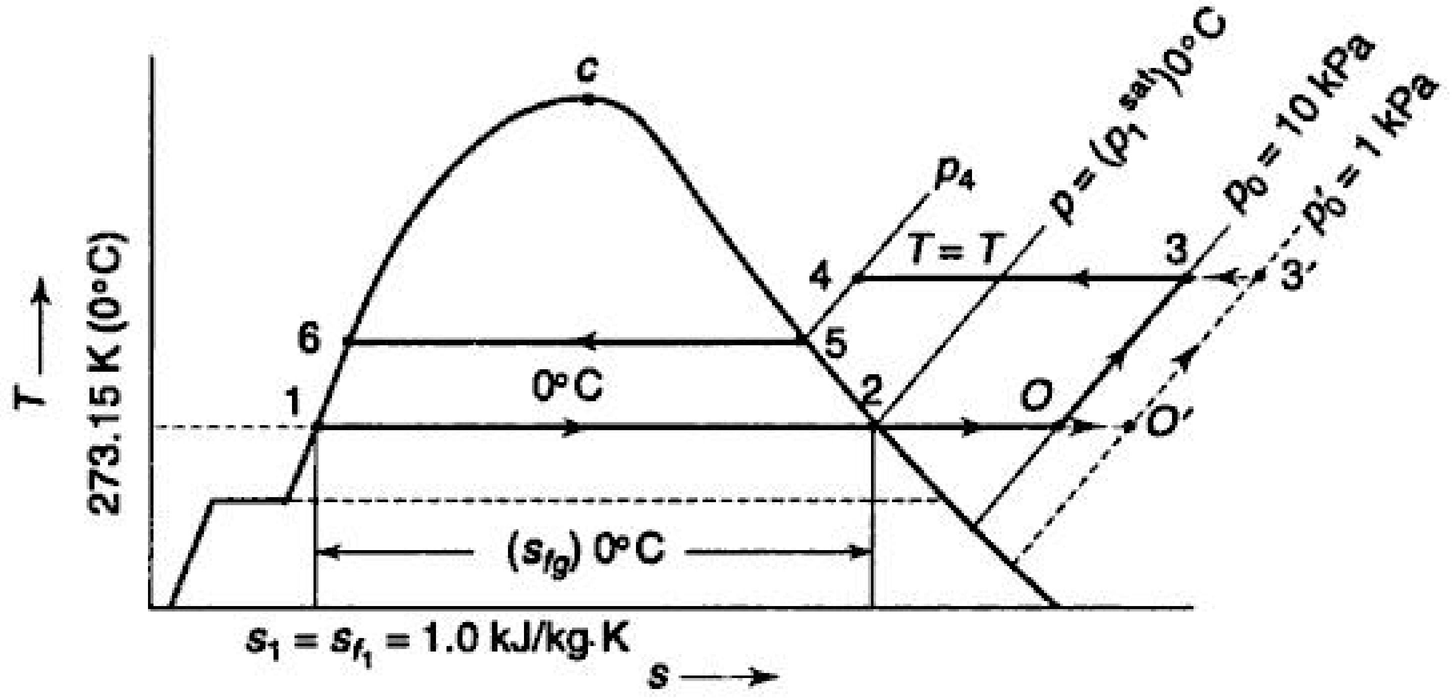


Fig. 1.15 Figure Demonstrating Method of Calculation of Entropy

$$s_2 = s_1 + (s_{fg})_{0°C} = 1.0 + \frac{(h_{fg})_{0°C}}{273.15} \text{ kJ/kg. K}$$

And the entropy of vapour at any point 4 is given by

$$s_4 = s_2 + (s_0 - s_2) + (s_3 - s_0) + (s_4 - s_0)$$

$$= s_2 - \left|\left(s - s^{id}\right)_{0°C}\right|_{p_o}^{p_2} + \int_{T_o}^{T_3} C_{p_o}\,\frac{dT}{T} + \left|\left(s - s^{id}\right)_{T_4}\right|_{p_o}^{p_4}$$

We note that in the calculation of residual entropy $s^R = (s - s^{id})$, the lower limit of integration is not $p_o = 0$ but p_o equal to some very low pressure, say 10 kPa, at which the gas behaves as an ideal gas. Correspondingly, the volume at 0 is $v_0^{id} = RT_0/p_0$, and the volume at 3 is $v_3^{id} = RT_3/p_0$. This is done because of a mathematical anamoly since $s \to \infty$ as $p \to 0$. The path followed for entropy calculation is shown in Fig. 1.15. In case a lower reference pressure is chosen such as $p_0' = 1$, 0.1, 0.01, etc., kPa, this will not affect the values at 4 since the new path followed will now be $2 - 0'$, $0' - 3'$, and $3' - 4$.

Example 1.11

Residual Entropy from Modified Martin-Hou Equation

Derive an expression for residual entropy from Eq. (1.15).

Solution

$$s^R = (s - s^{id})_T = \int_{v^{id}}^{v} \left(\frac{\partial p}{\partial T}\right)_v dv$$

$$= \int_{v^{id}}^{v} \frac{R}{v-b}\, dv + \int_{v^{id}}^{v} \sum_{i=2}^{5} \frac{B_i - C_i \dfrac{k}{T_c} e^{-kT/T_c}}{(v-b)^i}$$

$$= R \ln \frac{p_o v}{RT} + \sum_{i=2}^{5} \frac{B_i - C_i \dfrac{k}{T_c} e^{-kT/T_c}}{(-i+1)(v-b)^{i-1}} \tag{1.31}$$

Note ✍: *In the integration of the first term $(v - b)$ has been approximated to v. The term containing $1/(v^{id} - b)^{i-1}$, obtained after the integration of the second term, is approximated to zero as $v^{id} >> v$.*

Example 1.12 **Reference State Entropy of Freon 22 Vapour**

Find the reference state entropy s_0 of Freon 22 vapour at 273.15 K and 10 kPa using modified Martin-Hou equation.

Solution

Ideal gas volume at 273.15 K

$$v_0^{id} = \frac{RT_0}{p_0} = \frac{0.09615\,(273.15)}{10} = 2.62625 \text{ m}^3/\text{kg}$$

Specific volume of saturated Freon 22 vapour at 273.15 K

$$v_2 = 0.0471 \text{ m}^3/\text{kg, from Eq. (1.15)}$$

Substituting values in Eq. (1.31), we get

$$(s - s^{id})_{0°\text{C}} = s_2 - s_0 = -0.4016 \text{ kJ/kg}$$

Now, $$s_2 = s_1 + \frac{(h_{fg})_{0°\text{C}}}{273.15} = 1 + \frac{205}{273.15} = 1.7518 \text{ kJ/kg.K}$$

$$\Rightarrow \qquad s_0 = s_2 + 0.4016 = 2.1534 \text{ kJ/kg.K}$$

***Note* ✍:** *It will be interesting to see how the value of s_0 will change if $p_0' = 1$ kPa is taken. It will be seen that $s_0' > s_0$. But, it will not effect the value of s_4 since $s_3' > s_3$. See Fig. 1.15.*

Example 1.13 Entropy of Freon 22 Vapour

Find the entropy of superheated Freon 22 vapour at 1650 kPa and 95°C (368.15 K).

Solution

Ideal gas volume at 368.15 K

$$v_4^{id} = \frac{RT}{p_0} = \frac{0.09615\,(368.15)}{10} = 3.5398 \text{ m}^3\text{/kg}$$

Specific volume of superheated vapour at 1650 kPa and 368.15 K

$$v_4 = 0.0187 \text{ m}^3\text{/kg, from Eq. (1.15)}$$

Using Eq. (1.29)

$$s_3 - s_0 = \int_{T_0}^{T} C_{p_o} \frac{\mathrm{d}T}{T} = \int_{T_0}^{T} C_{v_o} \frac{\mathrm{d}T}{T} + R \ln \frac{T}{T_0}$$

$$= R \ln \frac{T}{T_0} + \left[C_1 \ln \frac{T}{T_0} + C_2\,(T - T_0) + \frac{C_3}{2} \right.$$

$$\left. (T^2 - T_0^2) - 2C_4 \left(\frac{1}{T} - \frac{1}{T_0} \right) \right] 10^{-3}$$

$$= 0.827 \text{ kJ/kg.K}$$

$$s_4 - s_3 = \left| \left(s - s^{id} \right)_{368.15\,\text{K}} \right|_{v^{id} = 3.5398}^{v = 0.0187} = -\,1.1488 \text{ kJ/kg.K}$$

Hence,

$$s_4 = s_0 + (s_3 - s_0) + (s_4 - s_3)$$

$$= 2.1534 + 0.827 - 1.1488 = 1.8316 \text{ kJ/kg.K}$$

1.16 MODES OF HEAT TRANSFER

The difference in temperature provides the necessary potential for heat transfer. There are three *modes of heat transfer*. They are *conduction, convection* and *radiation*.

Essentially heat is transferred within a *stationary* medium by conduction, viz., from particle to particle, whether it be solid, liquid or gas. In convection, there must be a bulk flow of the fluid. Heat is carried away from the wall surface by the flowing fluid. Convection, however, takes place in two ways, viz., *forced convection* and *natural* or *free convection*. In forced convection, the flow of the fluid is produced by an external source such as a pump or a fan. Examples are the shell and tube condenser of a refrigeration plant in which the flow of water is maintained by a pump, and the air-cooled condenser of an air conditioner in which the flow of air is maintained by a fan. In natural or free convection, the flow of the fluid is produced by the difference in density due to temperature difference. The higher temperature fluid, being lighter, rises up and the lower temperature fluid, being heavier, settles down. Thus a natural convection current is set up in the fluid. One example is the air-cooled condenser of a domestic refrigerator. In radiation, heat is transferred in the form of electromagnetic waves. For radiative heat transfer, therefore, the presence of a medium is not necessary.

1.17 LAWS OF HEAT TRANSFER

The *heat flux* q is the heat transfer rate $\dot{Q}$ per unit area A normal to the direction of flow of heat. The various laws relate the heat transfer rate or heat flux to temperature difference ΔT. The unit of ΔT is degrees kelvin (K) or degrees celsius (°C). Both have the same numerical value.

1.17.1 Fourier Law of Heat Conduction

The heat flux by conduction is proportional to the *temperature gradient* within a body. Thus at any point P in a body, if the temperature gradient is $\frac{\partial T}{\partial x}$ as shown in Fig. 1.16, then the heat flux is given by

$$q = \frac{\dot{Q}}{A} \propto \frac{\partial T}{\partial x}$$

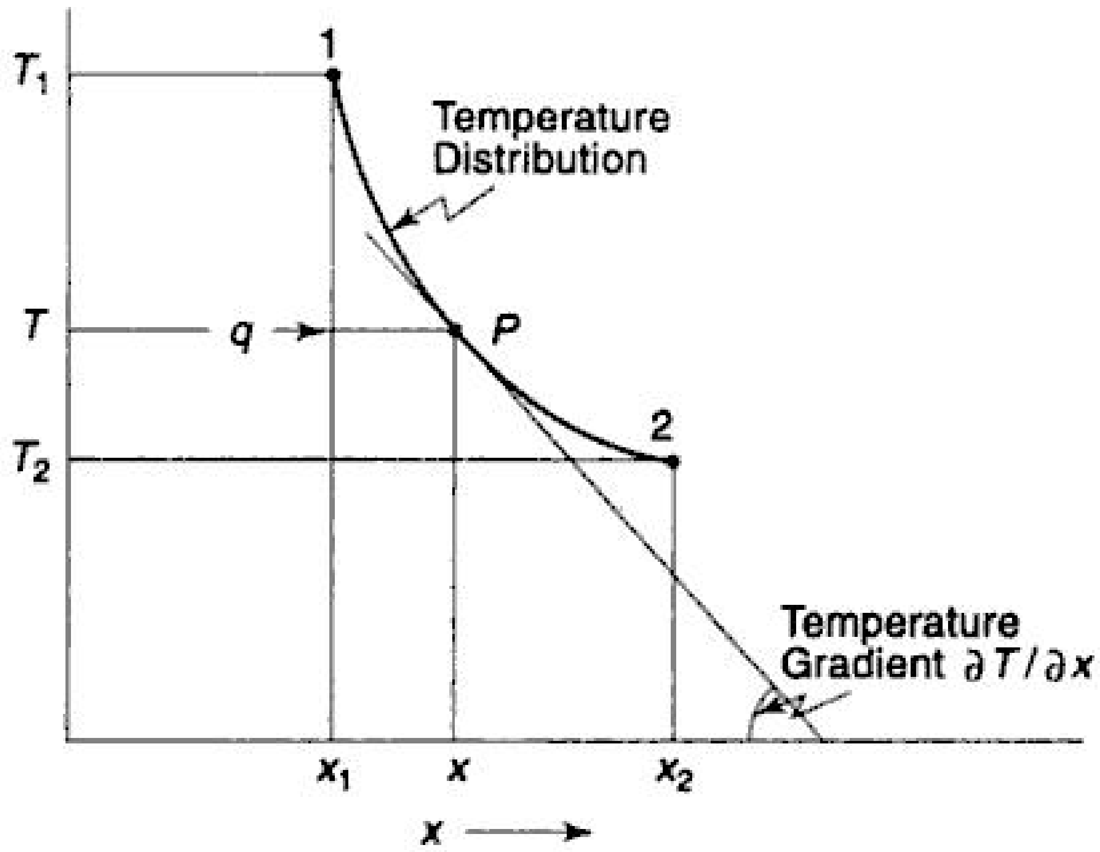


Fig. 1.16 Temperature Distribution in a Conduction Medium

or
$$\dot{Q} = -kA\frac{\partial T}{\partial x} \tag{1.32}$$

where k is the constant of proportionality, called *thermal conductivity* which is a property of the material of the body. The minus sign is inserted to make $\dot{Q}$ positive since $\frac{\partial T}{\partial x}$ is negative. Equation (1.32) is called the *Fourier law of heat conduction*.

The SI unit of thermal conductivity can be derived as below

$$[k] = \frac{[\dot{Q}]\,[\Delta x]}{[A]\,[\Delta T]}$$

$$= \frac{(\text{J/s})\ \text{m}}{\text{m}^2\ \text{K}} = \text{Wm}^{-1}\ \text{K}^{-1}$$

Physically, thermal conductivity represents the amount of heat that will flow per unit time, per unit area normal to the direction of flow of heat, through a unit thickness of the material and when the temperature difference across the material is unity. Experimental measurements of q and $\Delta T/\Delta x$ can be made to determine the thermal conductivity of materials. It is seen that the order of decreasing thermal conductivity is as follows:

Metals; non-metals; liquids; insulating materials; gases.

Metals

Common metals used in heat exchangers are copper, aluminium and iron. Their thermal conductivities in $\text{Wm}^{-1}\ \text{K}^{-1}$ units are as follows:

Copper	387
Aluminium	203
Iron	73

It is seen that copper has the highest thermal conductivity. It is, therefore, used in condensers, evaporators, etc., in Freon refrigeration systems. However, ammonia attacks copper. Hence, iron is used in ammonia heat exchangers. As aluminium is cheap, and widely available, attempts are being made to use it in Freon systems.

Thermal conductivities of other important substances are:

Liquids

Water	0.556
Ammonia	0.54
Freon 12	0.073

Gases

Air	0.024
Water vapour	0.0206

Low temperature insulating materials

Expanded polysterene (Thermocole)	0.037
Polyurethane foam (PUF)	0.0173

Their low values of thermal conductivities are essentially due to the low conductivity of gas/air trapped inside, as these are manufactured by blowing agents like air/R134a into the melted polyesterene and polyurethane respectively.

1.17.2 Newton's Law of Cooling for Convection

When a fluid flows over a wall which is at a different temperature than the fluid, heat will flow from the wall to the fluid or from the fluid to the wall depending on the direction of the temperature gradient. Although this mode of heat transfer is named convection, the physical mechanism of heat transfer at the wall is a conduction process.

Consider a fluid flowing along a wall maintained a temperature T_w as shown in Fig. 1.17. The free stream temperature of the fluid is T_∞. Then a temperature field varying from T_w to T_∞ will establish itself in the fluid near the wall. Let δ_T be the distance from the wall of the point in the fluid at which the temperature of the fluid just becomes equal to the free stream temperature T_∞. The distance δ_T is termed the *thermal boundary layer*. The distance δ from the wall in which the fluid velocity becomes equal to the free stream velocity is called the *hydrodynamic boundary layer*.

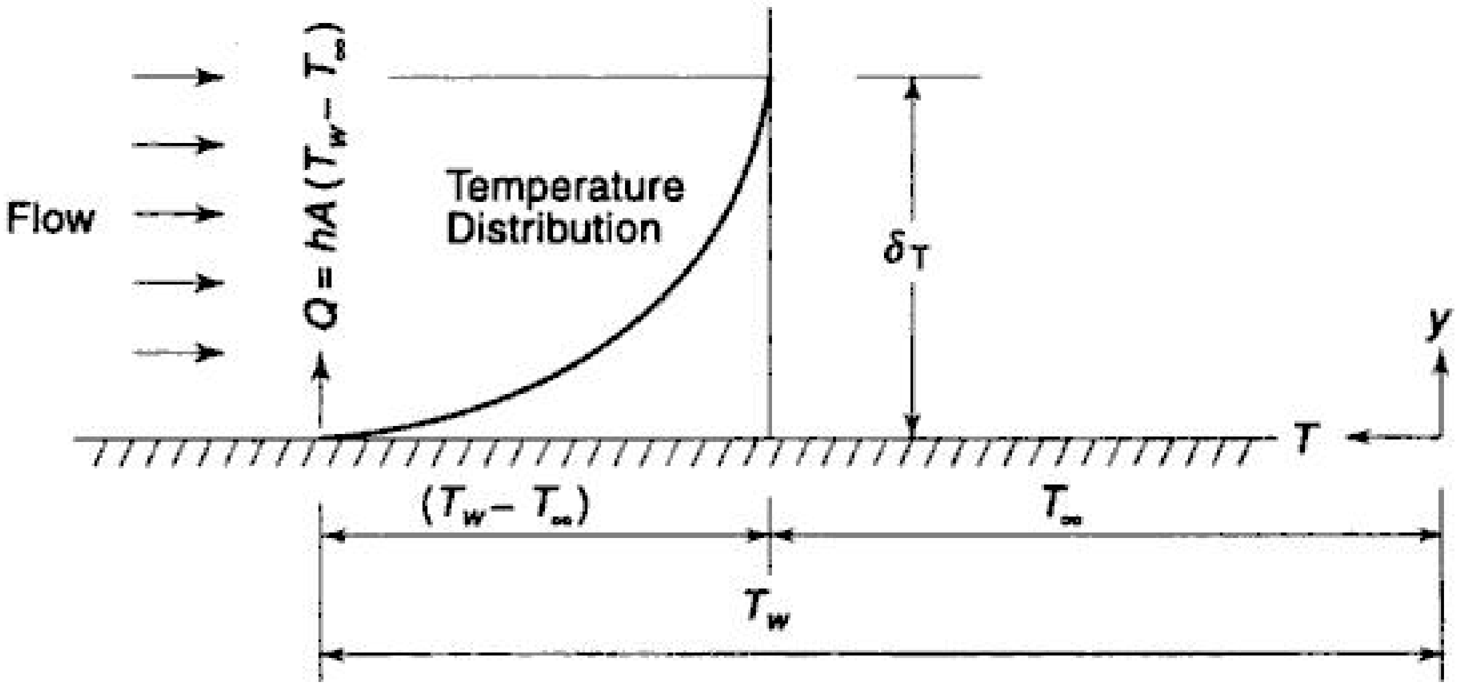


Fig. 1.17 Temperature Distribution in Thermal Boundary Layer in Convection

The concept of the *heat transfer coefficient* or *film coefficient* or *surface conductance*, denoted by the symbols h or f, was introduced by Newton. He recommended the following equation to evaluate the heat transfer rate by convection.

$$\dot{Q} = hA\,(T_w - T_\infty) = hA\,\Delta T \tag{1.33}$$

where A is the wall surface area and $\Delta T = (T_w - T\infty)$.

It must be noted that this h or f is not a property of the fluid. However, it depends on the *thermophysical* or *heat transport* properties of the fluid such as thermal conductivity k, dynamic viscosity μ, density ρ and specific heat C_p. In addition, it also depends on the hydrodynamic or flow parameters such as velocity of flow and characteristic dimensions.

The SI unit of the heat transfer coefficient can now be derived as follows:

$$[h] = \frac{[\dot{Q}]}{[A]\,[\Delta T]}$$

$$= \frac{\text{J/s}}{\text{m}^2\,\text{K}} = \text{Wm}^{-2}\,\text{K}^{-1}$$

Typical values of h are given in Table 1.2.

Table 1.2 Typical Values of Convection Heat Transfer Coefficients

Mode and Medium	h, W m^{-1} K^{-1}
Free convection, air	5 – 25
Forced convection, air	10 – 100
Forced convection, water	5,000 – 10,000
Boiling refrigerant	500 – 2,000
Condensing refrigerant	1,500 – 2,500

1.17.3 Thermal Radiation

The thermal radiation emitted by a body per unit area of its surface is called *emissive power* which is proportional to the fourth power of its absolute temperature.

For an *ideal radiator* or *black body*, the emissive power E_b is given by the *Stefan-Boltzman law* that follows in Eq. (1.34)

$$E_b = \sigma T^4 \tag{1.34}$$

where σ is the constant of proportionality called the *Stefan-Boltzman constant* having a numerical value of 5.669×10^{-8} W m^{-2} K^{-4}.

The emissive power of an actual radiator is expressed by the relation given in Eq. (1.35)

$$E = \varepsilon E_b = \varepsilon \sigma T^4 \tag{1.35}$$

where $\varepsilon = \dfrac{E}{E_b}$ is the emissivity of the actual surface.

Thermal radiations emitted by two bodies of areas A_1 and A_2, at temperatures T_1 and T_2 will then be given by

$$\dot{Q}_1 = \varepsilon_1 A_1 \sigma T_1^4 \text{ and } \dot{Q}_2 = \varepsilon_2 A_2 \sigma T_2^4$$

However, the heat exchange by thermal radiation between two bodies will also depend on the extent to which the two bodies "see" each other geometrically. The expression for such a heat exchange may be expressed as given in Eq. (1.36)

$$\dot{Q} = \sigma A_1 F_{12} (T_1^4 - T_2^4) = \sigma A_2 F_{21} (T_1^4 - T_2^4) \tag{1.36}$$

where $A_1 F_{12} = A_2 F_{21}$ is the *reciprocity relation*, and F_{12} and F_{21} are *geometric factors* which depend on the emissivities ε_1 and ε_2, and the geometry and orientation of the bodies.

In the case of a small body of area A_1 surrounded by a large body of area A_2, completely seeing each other, we have

$$F_{12} = \varepsilon_1 \tag{1.36}$$

so that

$$\dot{Q} = \varepsilon_1 A_1 \sigma (T_1^4 - T_2^4) \tag{1.37}$$

Often, the heat exchange by radiation $\dot{Q}_R$ between two surfaces is expressed in terms of a *radiation coefficient* h_R defined by

$$\dot{Q}_R = h_R A_1 (T_1 - T_2) = A_1 F_{12} \sigma (T_1^4 - T_2^4)$$

so that

$$h_R = \frac{F_{12} \sigma (T_1^4 - T_2^4)}{(T_1 - T_2)}$$

1.18 ELECTRICAL ANALOGY[1]

It is found convenient to handle complicated heat transfer problems involving composite materials and multi-modes by unifying the concept of heat transfer with that of the flow of electric current. Comparing the flow of electrical energy and the flow of heat, it is found that the following similarities hold.

Quantity	*Electrical Energy*	*Heat*
Driving potential	Voltage, V	Temperature difference, ΔT
Flow	Current, I	Heat transfer rate, $\dot{Q}$

The governing law for the transfer of electrical energy is Ohm's law

$$I = \frac{V}{R_e}$$

where R_e is the electrical resistance. One can similarly express the heat transfer rate by Eq. (1.38)

$$\dot{Q} = \frac{\Delta T}{R} = C \Delta T \tag{1.38}$$

where R is the *thermal resistance* and $C = 1/R$ is the *thermal conductance*. The units of the two quantities are $\mathrm{W}^{-1}\,\mathrm{K}$ and WK^{-1} respectively.

Comparring Eq. (1.38) with Eqs (1.32), (1.33) and (1.36) we obtain expressions for conductive, convective and radiative resistances respectively as follows:

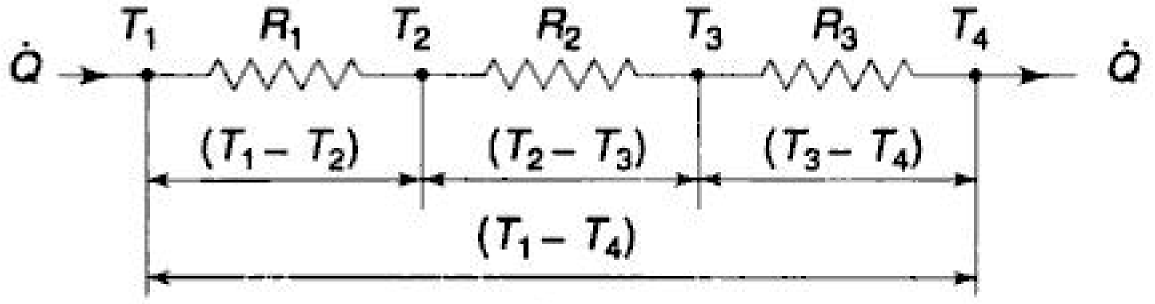
$$R_{\text{COND}} = \frac{\Delta x}{kA}, \text{ for plane wall of thickness } \Delta x \tag{1.39}$$

$$R_{\text{CONV}} = \frac{1}{hA} \tag{1.40}$$

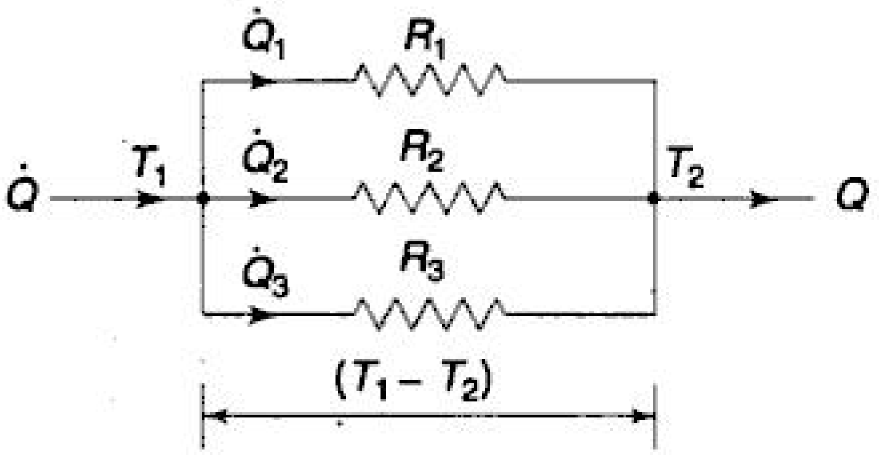
$$R_{\text{RAD}} = \frac{\Delta T}{\sigma A_1 F_{12} (T_1^4 - T_2^4)} \tag{1.41}$$

1.18.1 Resistances in Series

When resistance are in series as shown in Fig. 1.18 (a), it implies that the heat transfer rate through all the resistances is the same. We can then write: as given in Eq. (1.42)



(a) Resistances in Series



(b) Resistances in Parallel

Fig. 1.18 Thermal Resistances in Series and in Parallel

$$\dot{Q} = \frac{T_1 - T_2}{R_1} = \frac{T_2 - T_3}{R_2} = \frac{T_3 - T_4}{R_3} = \frac{\Delta T}{R} \tag{1.42}$$

where $\Delta T = (T_1 - T_4)$ is the overall temperature difference, and R is the overall thermal resistance. From Eq. (1.42), the individual temperature drops are:

$$T_1 - T_2 = \dot{Q} R_1$$

$$T_2 - T_3 = \dot{Q} R_2$$

$$T_3 - T_4 = \dot{Q} R_3$$

Adding, $\quad T_1 - T_4 = \dot{Q}\,(R_1 + R_2 + R_3)$

$$\Rightarrow \qquad \dot{Q} = \frac{T_1 - T_4}{R_1 + R_2 + R_3} = \frac{\Delta T}{R}$$

so that the overall thermal resistance R is given by Eq. (1.43)

$$R = R_1 + R_2 + R_3 + \cdots + \tag{1.43}$$

1.18.2 Resistances in Parallel

When resistances are in parallel, as shown in Fig. 1.18(b), the net heat transfer rate is equal to the sum of the heat transfer rates through all sections. At the same time, the temperature drop ΔT across each resistance is the same. Hence,

$$\dot{Q}_1 = \frac{\Delta T}{R_1}$$

$$\dot{Q}_2 = \frac{\Delta T}{R_2}$$

$$\dot{Q}_3 = \frac{\Delta T}{R_3}$$

Adding

$$\dot{Q} = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3$$

$$= \Delta T \left[\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} \right] = \frac{\Delta T}{R}$$

whence we get for the overall thermal resistance Eq. (1.44)

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots + \cdots \tag{1.44}$$

1.19 STEADY-STATE CONDUCTION

In general, the temperature distribution throughout a body may vary with location and time. Under *steady-state* conditions, however, the temperature does not change with time. We shall now examine some steady-state heat conduction problems in one dimension.

1.19.1 Heat Flow Through a Slab or Plane Wall

Let there be a slab of thickness Δx, the two faces of which are maintained at temperature T_1 and T_2 as shown in Fig. 1.19. Consider a section at a distance x from one end. Let the temperature gradient at this section be dT/dx. Then heat entering the wall, per unit time, at this section is

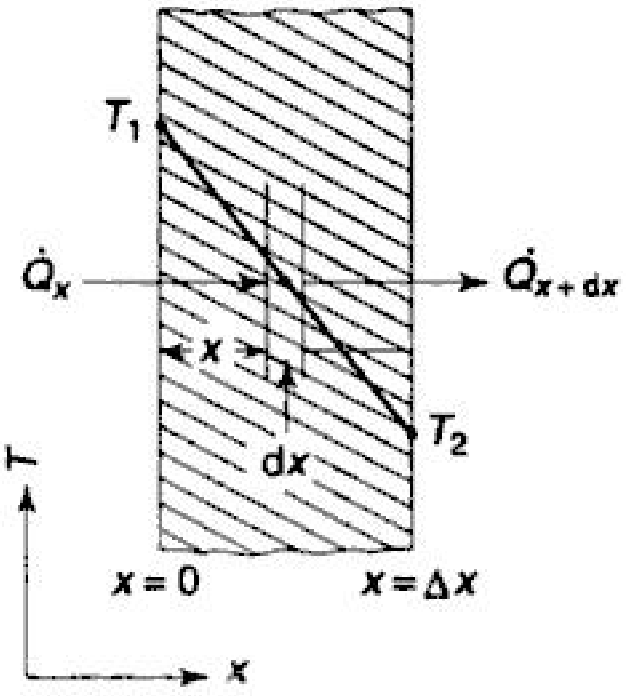


Fig. 1.19 Heat Conduction Through a Plane Wall

$$\dot{Q}_x = -kA\frac{\mathrm{d}T}{\mathrm{d}x} \tag{1.45}$$

Similarly, heat leaving the wall at $(x + \mathrm{d}x)$ is

$$\dot{Q}_{x+\mathrm{d}x} = \dot{Q}_x + \frac{\mathrm{d}\dot{Q}_x}{\mathrm{d}x}\,\mathrm{d}x \tag{1.46}$$

Under steady-state conditions

$$\dot{Q}_x = \dot{Q}_{x+\mathrm{d}x}$$

so that from Eqs (1.45) and (1.46) we have Eq. (1.47)

$$\frac{\mathrm{d}^2T}{\mathrm{d}x^2} = 0 \tag{1.47}$$

Using the boundary conditions

(i) $T = T_1$ at $x = 0$, (ii) $T = T_2$ at $x = \Delta x$

and solving the differential Eq. (1.47), we obtain for the temperature distribution in the wall

$$\frac{T - T_1}{T_2 - T_1} = \frac{x}{\Delta x} \tag{1.48}$$

which is linear with respect to x. The temperature gradient is

$$\frac{\mathrm{d}T}{\mathrm{d}x} = \frac{T_2 - T_1}{\Delta x} \tag{1.49}$$

which is constant. The heat transfer rate is given by the Fourier law in Eq. (1.50)

$$\dot{Q} = -kA\frac{\mathrm{d}T}{\mathrm{d}x} = \frac{kA\,(T_1 - T_2)}{\Delta x} = \frac{T_1 - T_2}{\Delta x/kA} \tag{1.50}$$

It is seen that the thermal resistance of the wall is

$$R = \frac{\Delta x}{kA}$$

1.19.2 Heat Flow Through a Cylinder

Consider a cylindrical shell of thickness dr at a distance r from the axis of a cylindrical tube or pipe of length L and outer and inner radii of r_0 and r_1 respectively, as shown in Fig. 1.20. The area of the inside surface of the shell will be $2\pi rL$. Then the heat flows in the radial direction at r and r + dr are:

$$\dot{Q}_r = -k\,(2\,\pi rL)\frac{\mathrm{d}T}{\mathrm{d}r}, \qquad \dot{Q}_{r+\mathrm{d}r} = \dot{Q}_r + \frac{\mathrm{d}\dot{Q}_r}{\mathrm{d}r}\cdot\mathrm{d}r$$

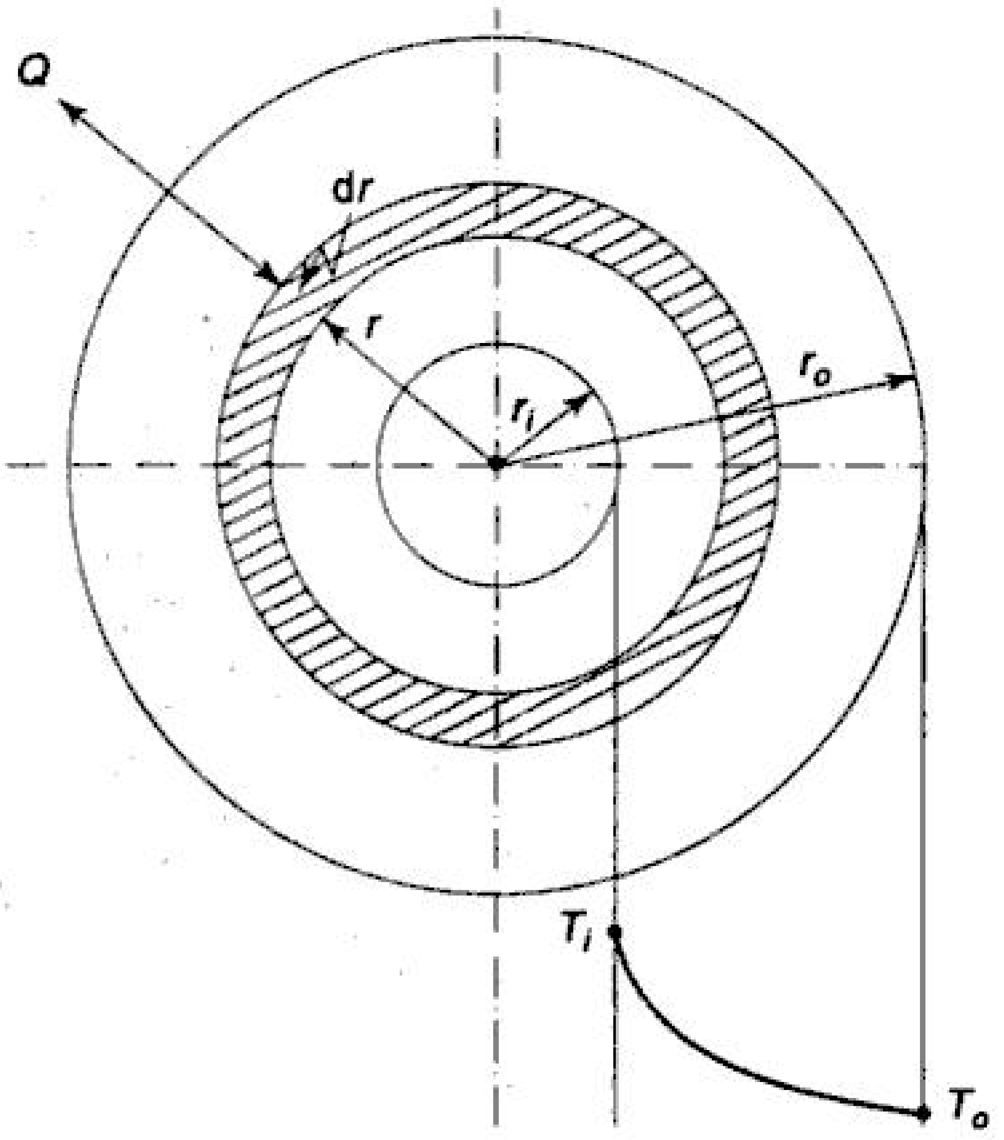


Fig. 1.20 Heat Conduction Through a Hollow Cylinder

Under steady-state conditions, equating the two we get

$$\frac{\mathrm{d}\dot{Q}_r}{\mathrm{d}r} = 0$$

$$\Rightarrow \qquad \frac{\mathrm{d}}{\mathrm{d}r}\left(r\,\frac{\mathrm{dT}}{\mathrm{d}r}\right) = 0 \qquad (1.51)$$

Using the boundary conditions

(i) $T = T_i$ at $r = r_i$

(ii) $T = T_o$ at $r = r_0$

and solving Eq. (1.51) we get for the temperature distribution

$$\frac{T - T_o}{T_i - T_o} = \frac{\ln\left(\dfrac{r}{r_o}\right)}{\ln\left(\dfrac{r_i}{r_o}\right)} \qquad (1.51a)$$

and for heat flow we have Eq. (1.52)

$$\dot{Q} = -\,kA\,\frac{\mathrm{d}T}{\mathrm{d}r}$$

$$= -\,k(2\pi rL)\,\frac{\mathrm{d}T}{\mathrm{d}r}$$

$$= \frac{T_i - T_o}{\left(\frac{1}{2\pi kL}\right)\ln\left(\frac{r_o}{r_i}\right)} \tag{1.52}$$

The thermal resistance of the cylindrical shell is then as follows in Eq. (1.53)

$$R = \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi kL} \tag{1.53}$$

1.19.3 Heat Flow Through a Composite Wall with Convection Boundaries

Consider a wall comprising of more than one material and convection at the two surfaces as shown in Fig. 1.21. It is seen that all the thermal resistances are in series. An equivalent electrical analogue of the wall is also shown in the figure. The overall thermal resistance is given by Eq. (1.54)

$$R = \frac{1}{h_o A} + \frac{\Delta x_A}{k_A A} + \frac{\Delta x_B}{k_B A} + \frac{\Delta x_C}{k_C A} + \frac{1}{h_i A} \tag{1.54}$$

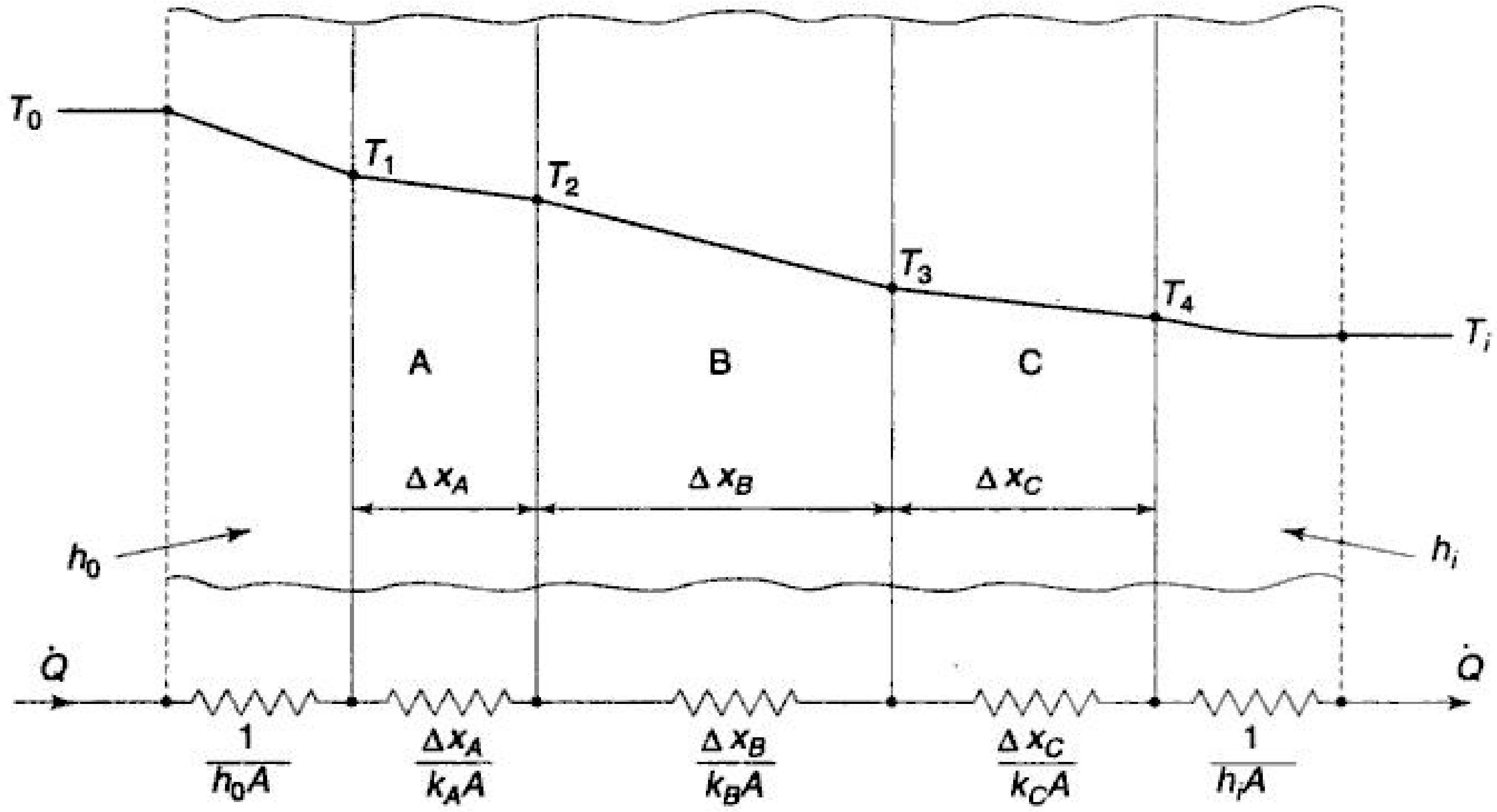


Fig. 1.21 Composite Plane Wall with Resistances in Series

where subscripts A, B and C refer to the three materials of the wall and h_i and h_o are the convective heat transfer coefficients between the inside and outside wall surfaces and surrounding air.

Similarly, if some of the resistances are in parallel while some are in series as shown in Fig. 1.22, then the overall thermal resistance is given by

$$R = \frac{1}{h_o A} + R_A + \left[\frac{1}{\dfrac{1}{R_B} + \dfrac{1}{R_C} + \dfrac{1}{R_D}} \right] + R_E + \frac{1}{h_i A} \tag{1.55}$$

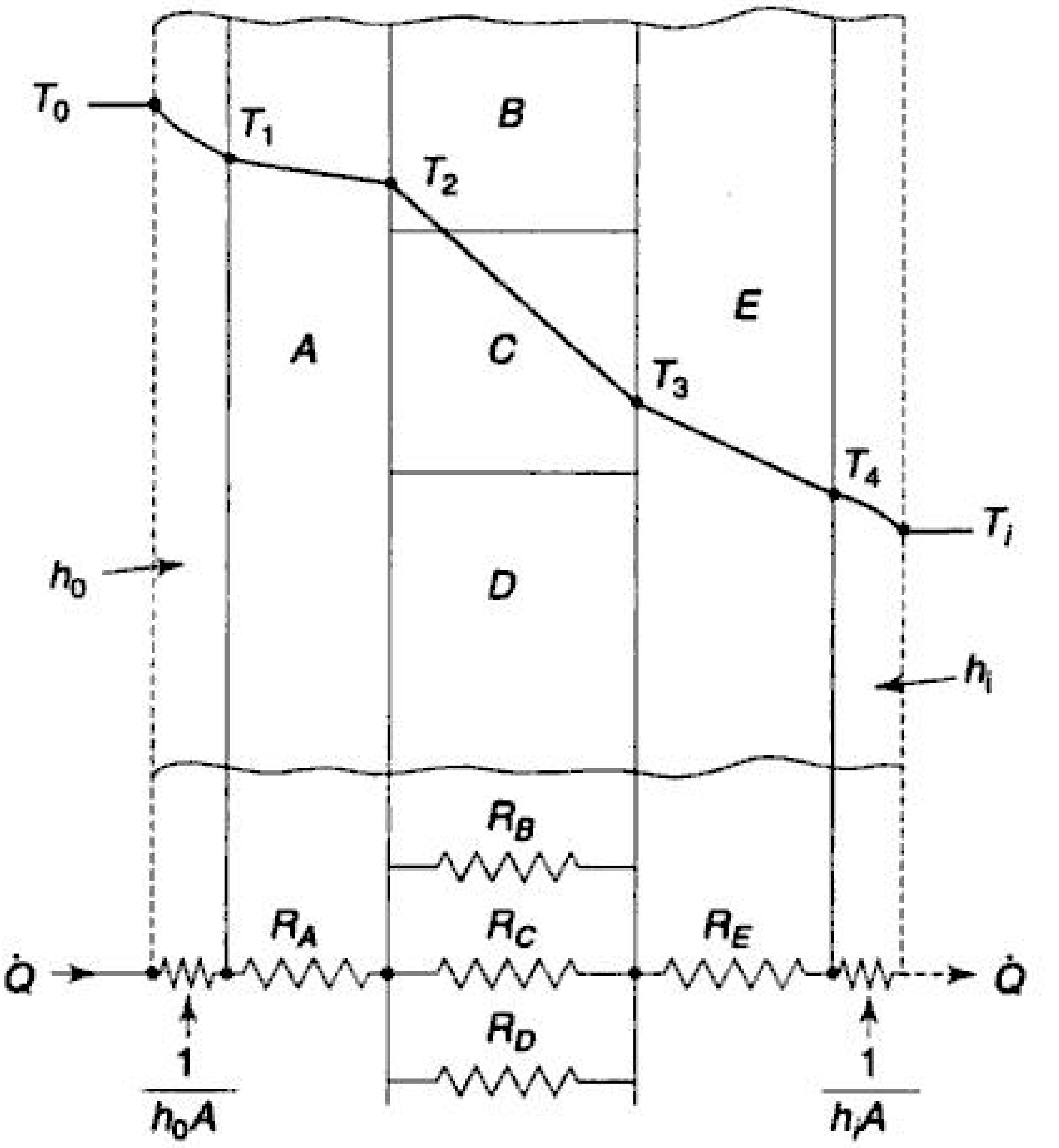


Fig. 1.22 Composite Plane Wall with Resistances in Series as well as in Parallel

In both cases, if T_o and T_i are the outside and inside air temperatures respectively, then the heat flow rate is given by

$$\dot{Q} = \frac{T_o - T_i}{R}$$

Example 1.14 An exterior wall of a house consists of 10.2 cm brick and 3.8 cm gypsum plaster. What thickness of loosely-packed rockwool insulation should be added to reduce the heat transfer through the wall by 80 per cent?

The thermal conductivities of brick, gypsum plaster and rockwool are 0.7, 0.48 and 0.065 W.m^{-1} K^{-1} respectively.

Solution

Since ΔT is the same in both cases, we have $(\dot{Q}\ R)' = 0.2\ (\dot{Q}\ R) = \Delta T$

$$\Rightarrow \qquad \frac{\dot{Q}' \text{ with insulation}}{\dot{Q} \text{ without insulation}} = 0.2 = \frac{R \text{ without insulation}}{R' \text{ with insulation}}$$

The resistances of brick and plaster are (per unit area of wall)

$$R_{\text{brick}} = \frac{0.102}{0.7} = 0.145 \text{ m}^2.\text{K.W}^{-1}$$

$$R_{\text{plaster}} = \frac{0.038}{0.48} = 0.079 \text{ m}^2.\text{K.W}^{-1}$$

Then the resistance without insulation is

$$R = R_{\text{brick}} + R_{\text{plaster}}$$

$$= 0.145 + 0.079 = 0.224 \text{ m}^2.\text{K.W}^{-1}$$

and the resistance with insulation is

$$R' = \frac{R}{0.2} = \frac{0.224}{0.2} = 1.122 \text{ m}^2.\text{K.W}^{-1}$$

Hence, the resistance of rockwool is

$$R_{\text{rockwool}} = R' - R$$

$$= 1.122 - 0.224 = 0.898 \text{ m}^2.\text{K.W}^{-1}$$

The required thickness of rockwool insulation is, therefore,

$$\Delta x_{\text{rockwool}} = (k\,AR)_{\text{rockwool}}$$

$$= (0.065)\ 1\ (0.898) = 0.0585 \text{ m } (\approx 6 \text{ cm})$$

1.19.4 Heat Flow Through a Composite Cylinder

Consider a composite tube of three materials with fluids flowing inside as well as outside the tube as shown in Fig. 1.23. Summing up all the resistances which are in series, we obtain Eq. (1.56)

$$R = \frac{1}{h_i A_i} + \frac{\ln \frac{r_2}{r_1}}{2\pi k_A L} + \frac{\ln \frac{r_3}{r_2}}{2\pi k_B L} + \frac{\ln \frac{r_4}{r_1}}{2\pi k_C L} + \frac{1}{h_o A_o} \tag{1.56}$$

For the case of a single tube as in heat exchangers, this becomes

$$R = \frac{1}{h_i A_i} + \frac{\ln \frac{r_o}{r_i}}{2\pi k L} + \frac{1}{h_o A_o} \tag{1.57}$$

where A_0 and A_i are the outside and inside tube surface areas respectively. Then the heat flow rate can be determined using the expression in Eq. (1.57).

Example 1.15 A 3 cm OD pipe is to be covered with two layers of insulation, each having a thickness of 2.5 cm. The average thermal conductivity of one insulation is five times that of the other. Determine the percentage decrease in heat transfer if the better insulating material is next to the pipe than if it is the outer layer.

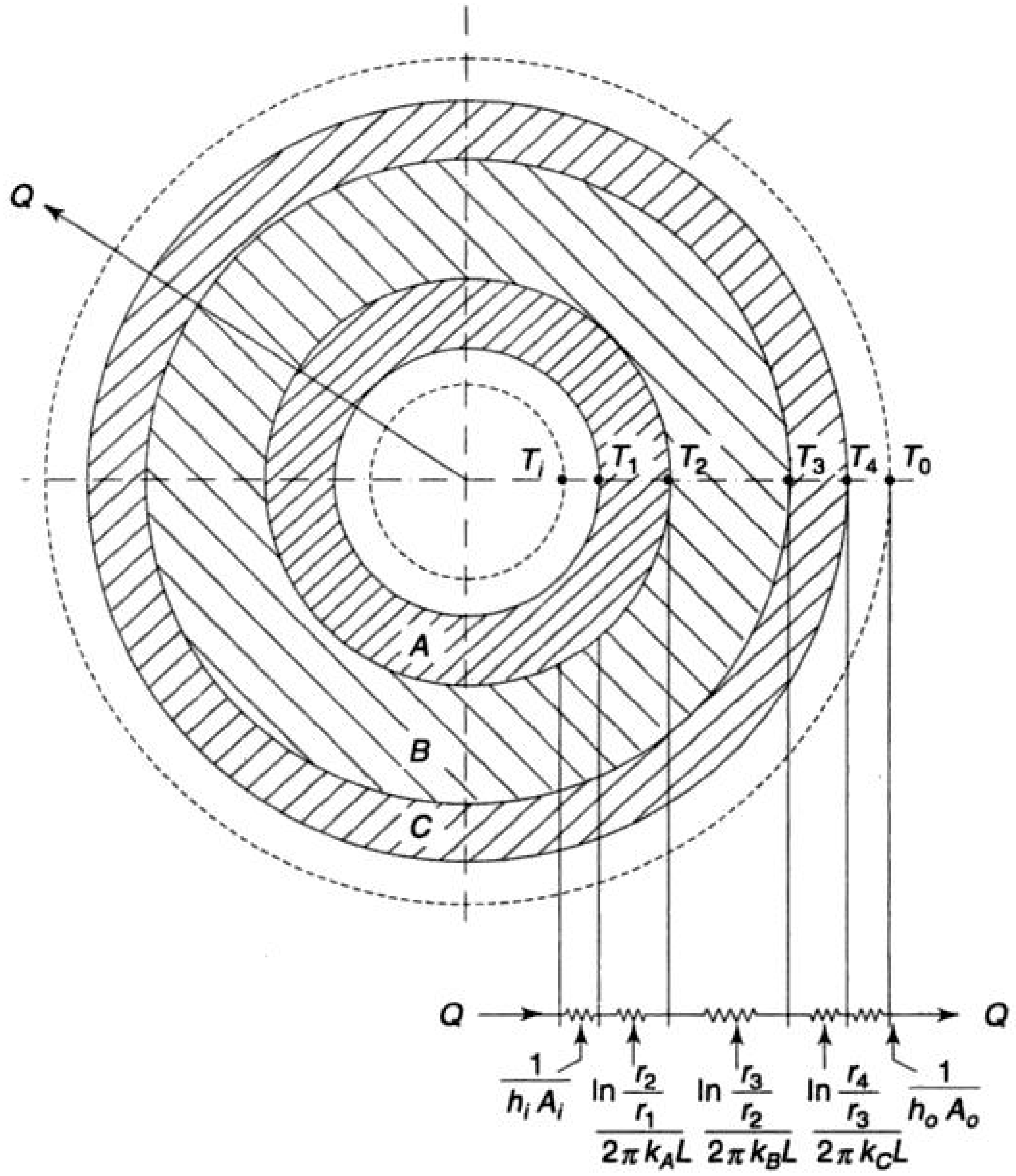


Fig. 1.23 Heat Flow Through a Composite Cylinder with Convection Boundaries

Assume that the outside and inside surface temperatures of the composite insulation are fixed.

Solution

Let the thermal conductivity of the poorer insulation be $k_1 = 5k$, and that of the better insulation be $k_2 = k$.

Case I: Better insulating material next to pipe

$$\dot{Q}_1 = \frac{\Delta T}{\frac{1}{2\pi k_1 L}\ln\frac{r_2}{r_1} + \frac{1}{2\pi k_2 L}\ln\frac{r_3}{r_2}}$$

$$= \frac{2\pi L\Delta T}{\frac{1}{k}\ln\frac{4}{2.5} + \frac{1}{5k}\ln\frac{6.5}{4}} = \frac{2\pi k L\Delta T}{1.0774}$$

Case II: Poorer insulating material next to pipe

$$\dot{Q}_{\text{II}} = \frac{2\pi L\Delta T}{\frac{1}{5k}\ln\frac{4}{2.5} + \frac{1}{k}\ln\frac{6.5}{4}} = \frac{2\pi kL\Delta T}{0.683}$$

Percentage reduction in heat flow

$$\frac{\dot{Q}_{\text{II}} - \dot{Q}_{\text{I}}}{\dot{Q}_{\text{II}}} \times 100 = \frac{\frac{1}{0.683} - \frac{1}{1.0774}}{\frac{1}{0.683}} \times 100 = 36.6\%$$

1.19.5 Overall Heat Transfer Coefficient

In order to calculate the rate of heat flow through a combination of resistances, the concept of *overall heat transfer coefficient* is introduced. Denoting it by the symbol U we express it by the relation

$$\dot{Q} = UA\,\Delta T = \frac{\Delta T}{R}$$

so that

$$\frac{1}{UA} = R$$

For a plane composite wall with all resistances in series as in Fig. 1.21, we find by comparison with Eq. (1.54) that

$$\frac{1}{UA} = \frac{1}{h_i A} + \frac{\Delta x_A}{k_A A} + \frac{\Delta x_B}{k_B A} + \cdots + \frac{1}{h_o A}$$

$$\Rightarrow \qquad \frac{1}{U} = \frac{1}{h_i} + \frac{\Delta x_A}{k_A} + \frac{\Delta x_B}{k_B} + \cdots + \frac{1}{h_o} \tag{1.58}$$

For a composite cylinder, the overall heat transfer coefficient can be based on either the outside or the inside tube surface area. Thus, we have

$$\dot{Q} = U_o A_o \Delta T = U_i A_i \Delta T = \frac{\Delta T}{R}$$

Then, by comparison with Eq. (1.56), we obtain

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{\ln\frac{r_2}{r_1}}{2\pi k_A L} + \frac{\ln\frac{r_3}{r_2}}{2\pi k_B L} + \cdots + \frac{1}{h_o A_o} \tag{1.59}$$

Equation (1.59) may be used to determine U_o or U_i. We see that $U_o A_o = U_i A_i$.

Example 1.16 (a) Find the overall heat transfer coefficient of a flat built-up roof having the construction shown in Fig. 1.24.
(b) Find the value of U if rigid roof deck insulation of resistance $R = 0.76$ K.W^{-1} is added to this construction.

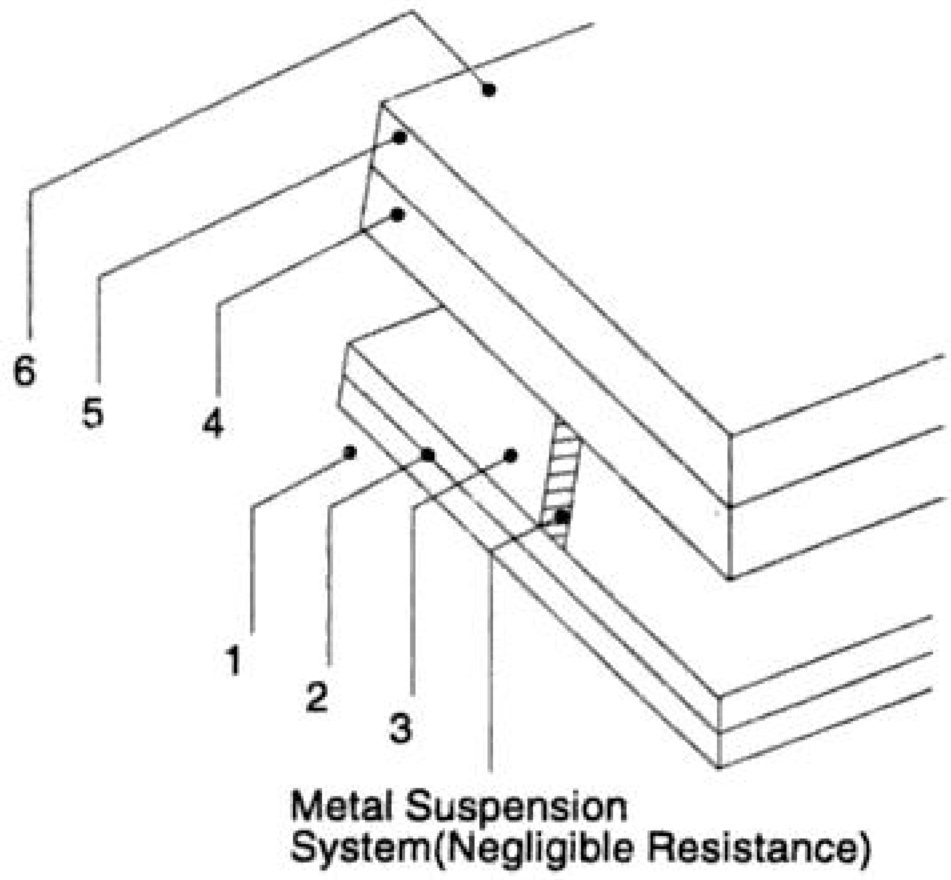


Fig. 1.24 Built-up Roof Construction for Example 1.16

Solution

(a) The values of resistances per unit area as found from ASHRAE Handbook[3] are as follows:

Resistance No.	*Construction material*	*Unit resistance, $\Delta x/k$ or $1/h$ m^2.K.W^{-1}*
1.	Inside surface (still air)	0.107
2.	Metal lath and 0.75 inch plaster	0.083
3.	Air space (greater than 9 cm width)	0.164
4.	Concrete slab, 5 cm	0.391
5.	Built-up roofing, 9.5 cm	0.058
6.	Outside surface (20 kmph wind velocity)	0.03
	Total thermal resistance, R_a	0.833

Overall heat transfer coefficient for construction '*a*'

$$U_a = \frac{1}{R_a} = \frac{1}{0.833} = 1.2 \text{ W.m}^{-2}\text{. K}^{-1}$$

(b) Total thermal resistance for construction '*b*'

$$R_b = R_a + 0.76 = 0.833 + 0.76 = 1.593 \text{ m}^2\text{. K.W}^{-1}$$

Overall heat transfer coefficient for construction '*b*'

$$U_b = \frac{1}{R_b} = \frac{1}{1.593} = 0.68 \text{ W.m}^{-2}\text{. K}^{-1}$$

1.20 HEAT TRANSFER FROM EXTENDED SURFACE

A *heat exchanger* is an apparatus which affects the transfer of heat from one fluid to another. The overall heat transfer coefficient of a heat exchanger surface is determine principally by the greatest single resistance. As an illustration, neglecting the thermal resistance of the metal wall of a heat exchanger, the overall heat transfer coefficient between the two fluids is given by

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o}$$

Taking $A_i = A_o = A$, and values of h_i and h_o as 1000 and 10 W. m^{-2}. K^{-1} respectively, we see that

$$\frac{1}{U} = \frac{1}{1000} + \frac{1}{10} = 0.101$$

$$\Rightarrow \qquad U = 9.9 \text{ W.m}^{-2}.\text{K}^{-1}$$

so that the value of U is less than the value of the lower of the two heat transfer coefficients, viz., $h_o = 10$, in this case. The lower coefficient is, therefore, the *controlling coefficient*. The influence of the higher coefficient is only marginal. The size of the heat exchanger will thus be uneconomically large. There are two methods by which the heat exchange can be improved. One is to augment the lower heat transfer coefficient. The other is to employ an extended surface on the side of the lower coefficient. Both methods decrease the thermal resistance, one by increasing h, and the other by increasing A.

Consider an extended surface in the form of a thin rod protruding from a surface into a surrounding fluid as shown in Fig. 1.25. The *root* or *base* of the rod is at temperature T_1 while the fluid temperature is T_∞. The length, cross-sectional area and perimeter of the rod are l, A and P respectively. The temperature distribution along the rod is also shown in Fig. 1.25.

Consider an element dx at a distance x from the base of rod. The heat flows by conduction within the rod, and by convection from its surface to the surrounding fluid. The energy balance over the element dx under steady-state gives

$$\dot{Q}_x - \dot{Q}_{x+\mathrm{d}x} = \dot{Q}_{\mathrm{conv}}$$

$$\left(-kA\frac{\mathrm{d}T}{\mathrm{d}x}\right)_x - \left(-kA\frac{\mathrm{d}T}{\mathrm{d}x}\right)_{x+\mathrm{d}x} = h(P\mathrm{d}x)\,(T - T_\infty) \tag{1.60}$$

where T is the temperature of the rod at x, k is the thermal conductivity of the rod, and h is the heat transfer coefficient from the surface to the fluid. Equation (1.60) can be simplified to

$$\frac{\mathrm{d}^2\theta}{\mathrm{d}x^2} = m^2\theta = 0 \tag{1.61}$$

by putting θ = excess temperature = $(T - T_\infty)$,

and $$m = \sqrt{\frac{hP}{kA}}$$

Using the boundary conditions

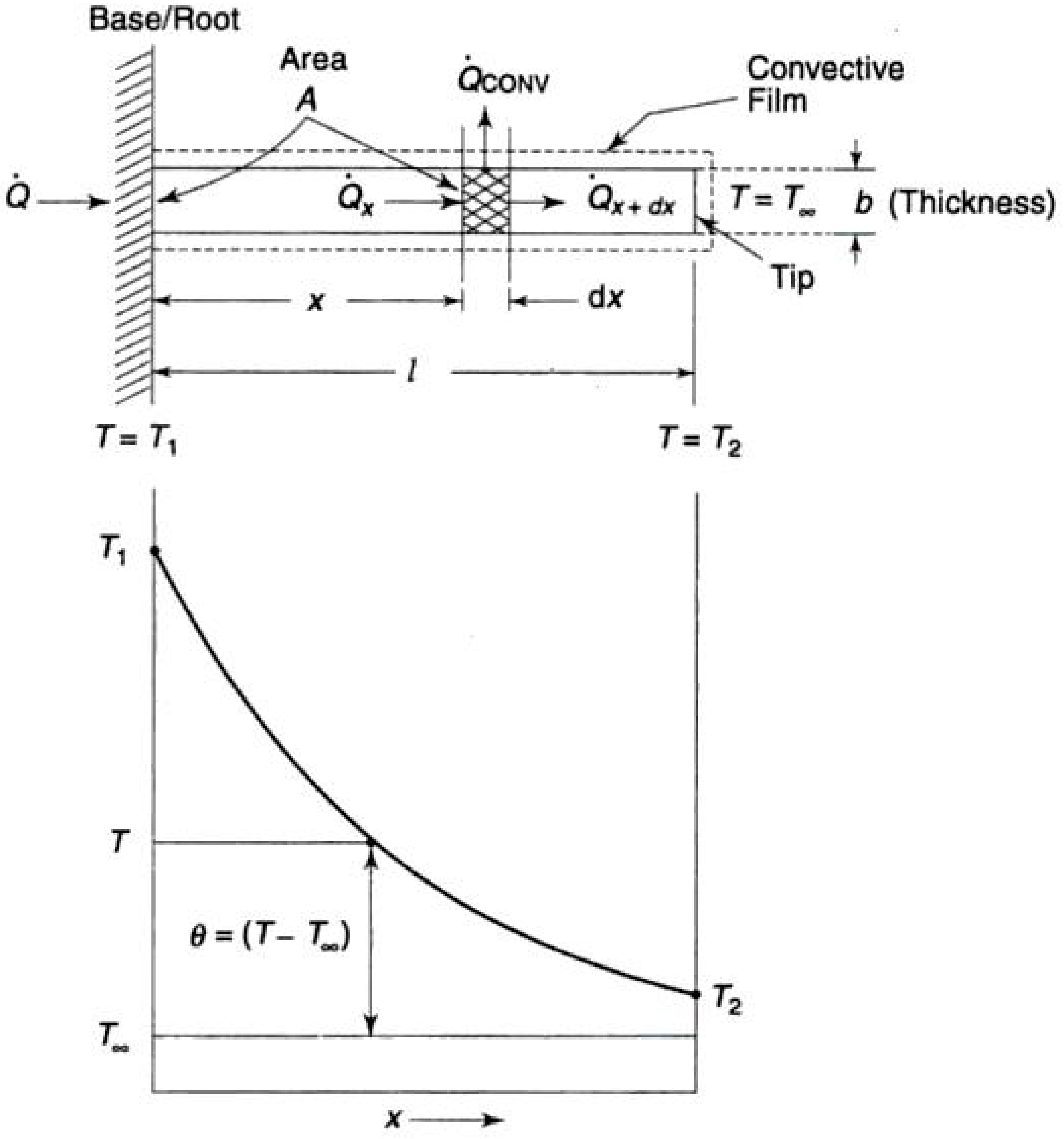


Fig. 1.25 Heat Flow and Temperature Distribution along the Length of a Rod Protruding from a Surface

(i) $\theta = T_1 - T_\infty = \theta_1$ at $x = 0$, and

(ii) $\dfrac{d\theta}{dx} = 0$ at $x = l$ (no heat conduction at *tip*)

we obtain the solution of Eq. (1.61) for temperature distribution as

$$\frac{\theta}{\theta_1} = \frac{\cosh m(l - x)}{\cos ml} \tag{1.62}$$

Then the heat flow from the base of the rod (at $x = 0$) is given by Eq. (1.63)

$$\begin{aligned}\dot{Q}_1 &= -kA \left|\frac{d\theta}{dx}\right|_{x=0} \\ &= mkA\theta_1 \tanh ml \\ &= \sqrt{hPkA}\ \theta_1 \tanh ml \end{aligned} \tag{1.63}$$

The term $\sqrt{hPkA}\ \theta_1 = mkA\theta_1$ is constant. The value of tanh *ml* increases with *ml* and tends to unity. It becomes 0.964 at $ml = 2$ and 0.9951 at $ml = 3$ as seen from Table 1.3. It is, therefore, apparent that increasing the length of an extended surface beyond $ml = 2$ or 3 does not improve the heat flow.

For a *rectangular fin* of height l, width L and thickness b, in place of a rod, as shown in Fig. 1.25 the cross-sectional area and perimeter are

$$A = bL,\ P = 2L + 2b = 2(L + b) \approx 2L$$

so that the parameter m is

$$m = \sqrt{\frac{hP}{kA}} = \sqrt{\frac{2h(L+b)}{kbL}} \cong \sqrt{\frac{2h}{kb}}$$

The entire fin is not at temperature T_1. It drops from T_1 at the root to T_2 at the tip. Hence, the whole fin surface is not equally effective.

The fin efficiency η_f is now defined as the ratio of the actual heat transferred by a fin to that which would be transferred if the entire fin surface were assumed to be at the base temperature. Thus

$$\eta_f = \frac{\dot{Q}_1 \text{ with fin}}{\dot{Q} \text{ with fin surface at base temperature}}$$

For a rectangular fin, therefore,

$$\eta_f = \frac{mkA\theta_1 \tanh ml}{h(Pl)\,\theta_1} = \frac{\tanh ml}{ml}$$

Table 1.3 gives the values of tanh ml and η_f as a functions of ml. It is seen that tanh ml, and hence $\dot{Q}_1$ increase rapidly at first as l increases. The increase, then, slows down finally reaching an asymptotic value at $ml \approx 3$. The fin efficiency is quite high upto $ml \approx 0.5$, but decreases rapidly as ml increases.

Table 1.3 Numerical values of tanh ml and η_f

ml	0	0.5	1.0	1.5	2	3	4	5	6
tanh ml	1	0.4621	0.7616	0.9052	0.964	0.995	0.9993	0.9999	1
η_f		0.924	0.7616	0.603	0.482	0.332	0.2498	0.20	0.167

1.20.1 Efficiency of Circular Fins and Finned Tube Arrays

A largely empirical method, developed by Schmidt[15] for finding the efficiency of a circular fin the configuration of which is shown in Fig. 1.26, is summarized as follow in Eqs (1.64) and (1.65)

$$\eta_f = \frac{\tanh(mr\phi)}{(mr\phi)} \tag{1.64}$$

where

$$\phi = \left(\frac{R}{r} - 1\right)\left[1 + 0.35 \ln\left(\frac{R}{r}\right)\right] \tag{1.65}$$

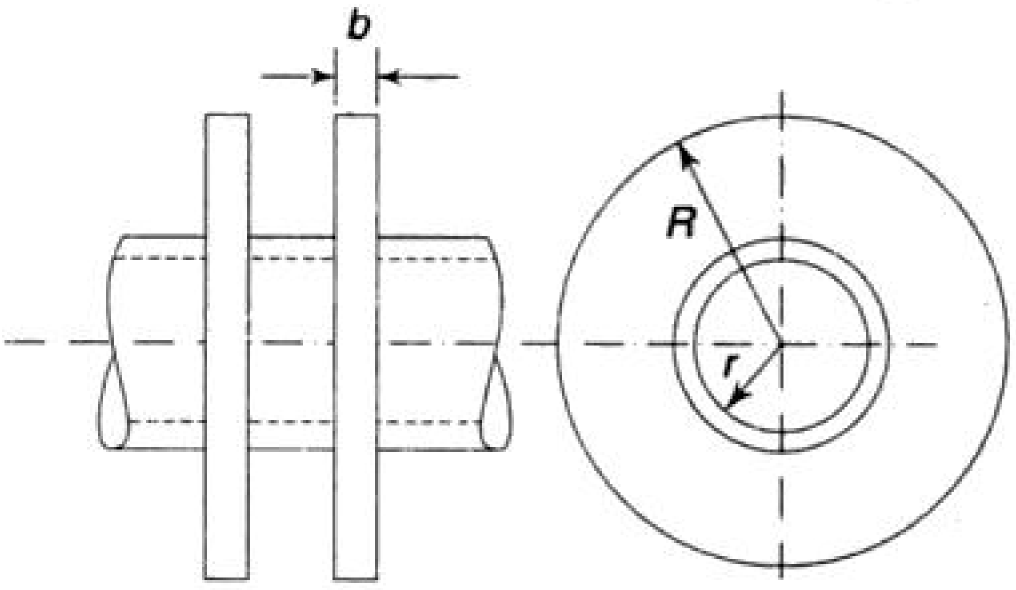


Fig. 1.26 Circular Fins

However, in air conditioning, continuous plate fins are used in finned tube cooling coils. The two array configurations, viz., the rectangular tube array and the angular tube array (hexangular fin) are shown in Figs 1.27 and 1.28 respectively.

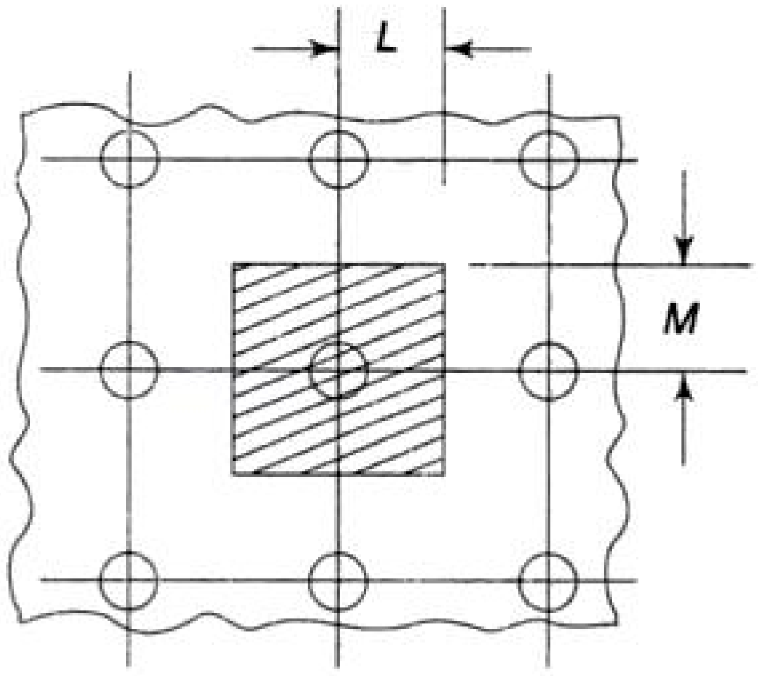


Fig. 1.27 Rectangular Tube Array Fin

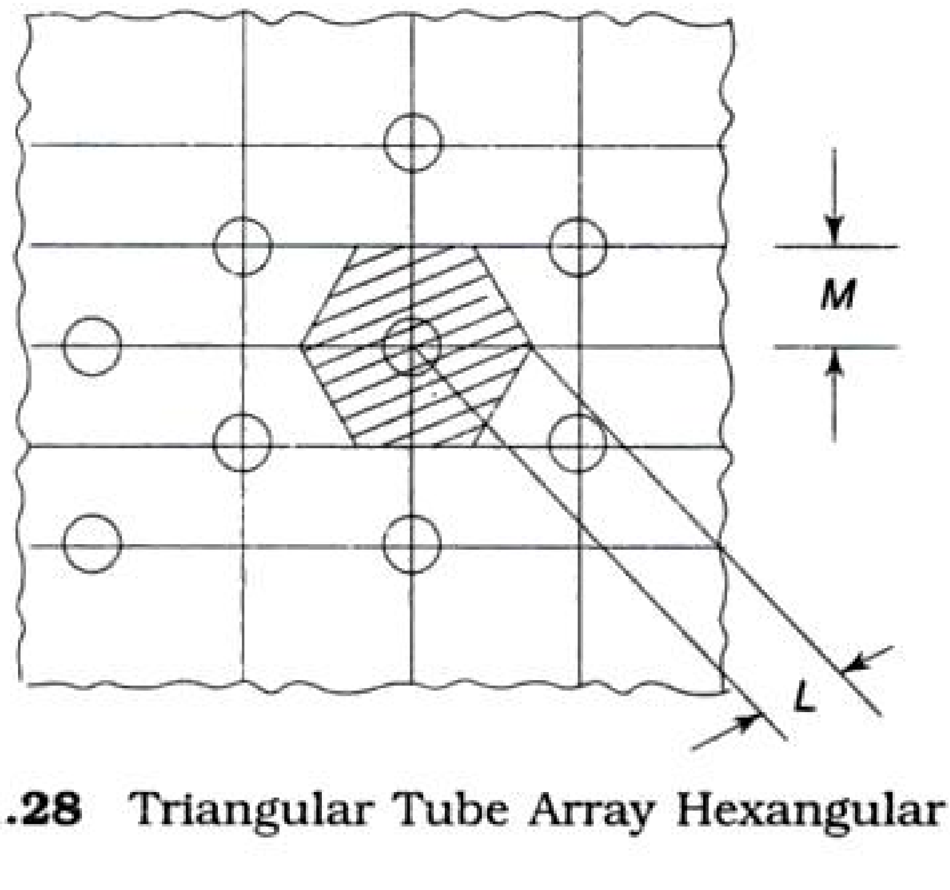


Fig. 1.28 Triangular Tube Array Hexangular Fin

Schmidt, again, provides an empirical method for calculating the efficiency of these fins. The method is based on selecting a circular fin with an equivalent radius R_e that has the same fin efficiency as the rectangular fin as follows in Eqs (1.66) and (1.67).

For the Rectangular Tube Array Fin

$$\frac{R_e}{r} = 1.28\ \psi(\beta - 0.2)^{1/2} \tag{1.66}$$

For the Triangular Tube Array Hexangular Fin

$$\frac{R_e}{r} = 1.27\ \psi(\beta - 0.3)^{1/2} \tag{1.67}$$

In both the above expressions, from Figs (1.27) and (1.28)

$$\psi = \frac{M}{r} \text{ and } \beta = \frac{L}{M}$$

in which L is always selected to be greater than or equal to M.

1.21 UNSTEADY-STATE CONDUCTION

In unsteady-state conduction, the temperature of the body changes with time. This change in temperature is represented by $\partial T/\partial \tau$ where τ stands for time. Thus, in the case of a plane wall element dx (Fig. 1.19), we have

$$\dot{Q}_x - \dot{Q}_{x+\mathrm{d}x} = C_p\,(A\,\mathrm{d}x)\,\frac{\partial T}{\partial \tau} \tag{1.68}$$

where ρ is the density, and C_p is the specific heat of the wall.

The right hand side in Eq. (1.68) represents the increase in stored energy of the element. Substituting for $\dot{Q}_x$ and $\dot{Q}_{x+\mathrm{d}x}$ from Eqs (1.45) and (1.46) we obtain the unsteady-state heat conduction equation in one dimension as

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial \tau} \tag{1.69}$$

where $\alpha = k/\rho C_p$ is called the *thermal diffusivity* of the wall material. It will be seen that the units of thermal diffusivity are $\mathrm{m}^2\,\mathrm{s}^{-1}$.

Various analytical methods have been used to solve Eq. (1.69). These include, among others, *the methods of separation of variables* and *the method of transformation of coordinates.* In all cases, the solutions are of the following dimensionless form

$$\frac{\theta}{\theta_i} = f\left(\frac{x}{L}, \mathrm{Bi}, \mathrm{Fo}\right)$$

where

θ = excess temperature at any time τ at $x = t - t_{\text{reference}}$

θ_i = initial excess temperature at x

x/L = dimensionless distance

Bi = *Biot number* = $\dfrac{hL}{k}$

Fo = *Fourier number* = $\dfrac{\alpha\tau}{L^2}$

Here h represents the heat transfer coefficient at the surface and L is the distance between the centre line of the material and the surface. Many charts, such as those of Gurnie-Lurie and Heisler[5], present the solution of Eq. (1.69) in graphical form. In addition to analytical methods, there are also numerical methods employed to solve this equations. One such method using *finite difference approach* has been employed in Chapter 18.

1.22 FORCED CONVECTION CORRELATIONS

Correlations for the heat transfer coefficient h in forced convection are expressed in the following form

$$\mathrm{Nu} = f(\mathrm{Re}, \mathrm{Pr})$$

where Nu, Re and Pr are dimensionless numbers expressed as follows:

$$\mathrm{Nu} = \textit{Nusselt number} = \frac{hL}{k}$$

$$\mathrm{Re} = \textit{Reynolds number} = \frac{Lu\rho}{\mu} \text{ or } \frac{Du\rho}{\mu}$$

$$\mathrm{Pr} = \textit{Prandtl number} = \frac{C_p \mu}{k}$$

Here L is any characteristic length in the flow geometry. In the case of pipes, it is equal to the diameter D. Also u is the mean flow velocity and k, ρ, C_p and μ are the thermal conductivity, density, specific heat and dynamic viscosity of the fluid respectively.

The Reynolds number represents the ratio of inertia forces to viscous forces. It is a flow criterion. Generally, for a pipe, the flow is streamlined or *laminar* when $\mathrm{Re} < 2100$, and *turbulent* when $\mathrm{Re} > 2300$. For intermediate values of Re, the flow is in the transition region.

For turbulent flow in smooth tubes, the following correlation is commonly used.

Dittus-Boelter equation

$$\mathrm{Nu} = 0.023\,(\mathrm{Re})^{0.8}\,(\mathrm{Pr})^{n} \tag{1.70}$$

where

$$n = 0.4 \text{ for heating}$$
$$= 0.3 \text{ for cooling}$$

For turbulent flow of fluids perpendicular to tubes such as water in the shell the following correlation (see Kern[7]) is used

$$\mathrm{Nu} = 0.36\,(\mathrm{Re})^{0.55}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{1.71}$$

For determining Nu and Re, the outside diameter of the tube is taken as the characteristic length. The velocity is measured in the free stream. The properties of the fluid are taken at the bulk mean temperature, whereas μ_w is at the wall temperature.

1.23 FREE CONVECTION CORRELATIONS

In free or natural convection, the flow velocity is developed as a result of the temperature difference causing a buoyance force to act on the fluid. The flow criterion in the case of free convection is, therefore, the *Grashof number* which is defined by

$$\mathrm{Gr} = \frac{g\beta\rho^2\,\Delta T L^3}{\mu^2}$$

in which β is the *coefficient of thermal expansion* of the fluid defined by

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_p$$

For an ideal gas

$$v = \frac{RT}{p}, \left(\frac{\partial v}{\partial T}\right)_p = \frac{R}{p}, \text{ so that } \beta = \frac{1}{T}$$

Also, $\Delta T = (T_w - T_\infty)$ is the temperature difference between the wall and the surrounding fluid.

Free convection correlations are found to have the following form

$$\mathrm{Nu} = f(\mathrm{Gr}, \mathrm{Pr})$$

In many cases, the relationship simplifies to

$$\mathrm{Nu} = C\,(\mathrm{Gr}\cdot\mathrm{Pr})^m = C\,\mathrm{Ra}^m$$

The product (Gr · Pr) is named Rayleigh number (Ra). It is a criterion of transition from laminar to turbulent flow in free convection.

For laminar flow, $10^4 < \mathrm{Gr}\cdot\mathrm{Pr} < 10^9$

For turbulent flow, $\mathrm{Gr}\cdot\mathrm{Pr} > 10^9$

For the case of air, Eq. (1.72) is simplified as written below in Eqs (1.73a), (1.73b), (1.74a) and (1.74b).

Vertical plate or cylinder to air

$$h = 1.42\left(\frac{\Delta T}{L}\right)^{1/4} \text{ laminar} \tag{1.73a}$$

$$h = 0.95\,(\Delta T)^{1/3} \text{ turbulent} \tag{1.73b}$$

Horizontal cylinder to air

$$h = 1.32\left(\frac{\Delta T}{D}\right)^{1/4} \text{ laminar} \tag{1.74a}$$

$$h = 1.24 \left(\frac{\Delta T}{D} \right)^{1/3} \text{ turbulent} \tag{1.74b}$$

1.24 DESIGN OF HEAT EXCHANGERS

The thermal design of a heat exchanger is primarily concerned with the determination of the heat transfer area required to transfer a specified amount of heat between two fluids with specified flow rates and inlet temperatures.

In a heat exchanger, usually, the temperatures of fluids on both sides of the heat transfer surface vary as a result of heat exchange except in the case of the fluid undergoing change of phase. This variation has to be taken into account in any design procedure.

There are three basic types of heat exchangers. They are:

(i) Parallel flow
(ii) Counter flow
(iii) Cross flow.

In parallel flow, both the fluids flow past the heat transfer surface in the same direction as shown in Fig. 1.29. In counter flow, the two fluids flow in opposite directions as shown in Fig. 1.30, In cross flow, the two fluids flow at right angles to each other.

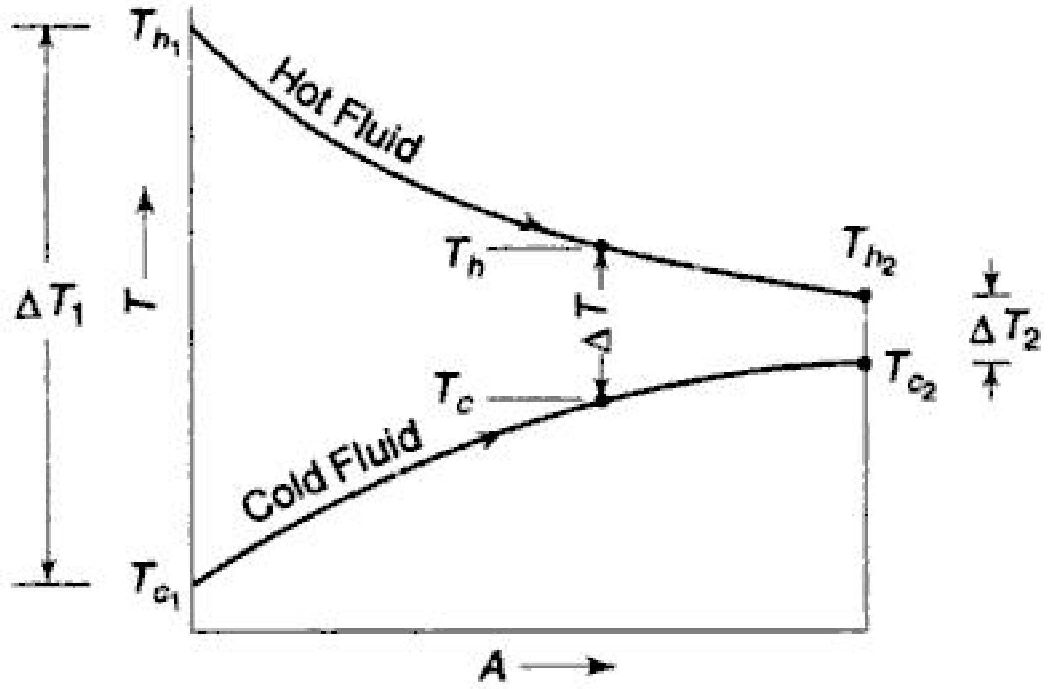


Fig. 1.29 Temperature Variation in Parallel Flow Heat Exchanger

1.24.1 Log Mean Temperature Difference (LMTD) Method

It is proposed to calculate the required heat transfer surface area by the equation

$$\dot{Q} = UA\,\Delta T_m$$

where ΔT_m is the *log mean temperature difference* given by

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln \dfrac{\Delta T_1}{\Delta T_2}} \tag{1.75}$$

where ΔT_1 and ΔT_2 are the temperature differentials between the two fluids at the two ends of the heat exchanger.

There are two great advantages of counter flow heat exchangers.

(1) A counterflow heat exchanger has a higher value of LMTD, for the same *end temperatures* as obtained from Eq. (1.75) and seen from Fig. 1.30.

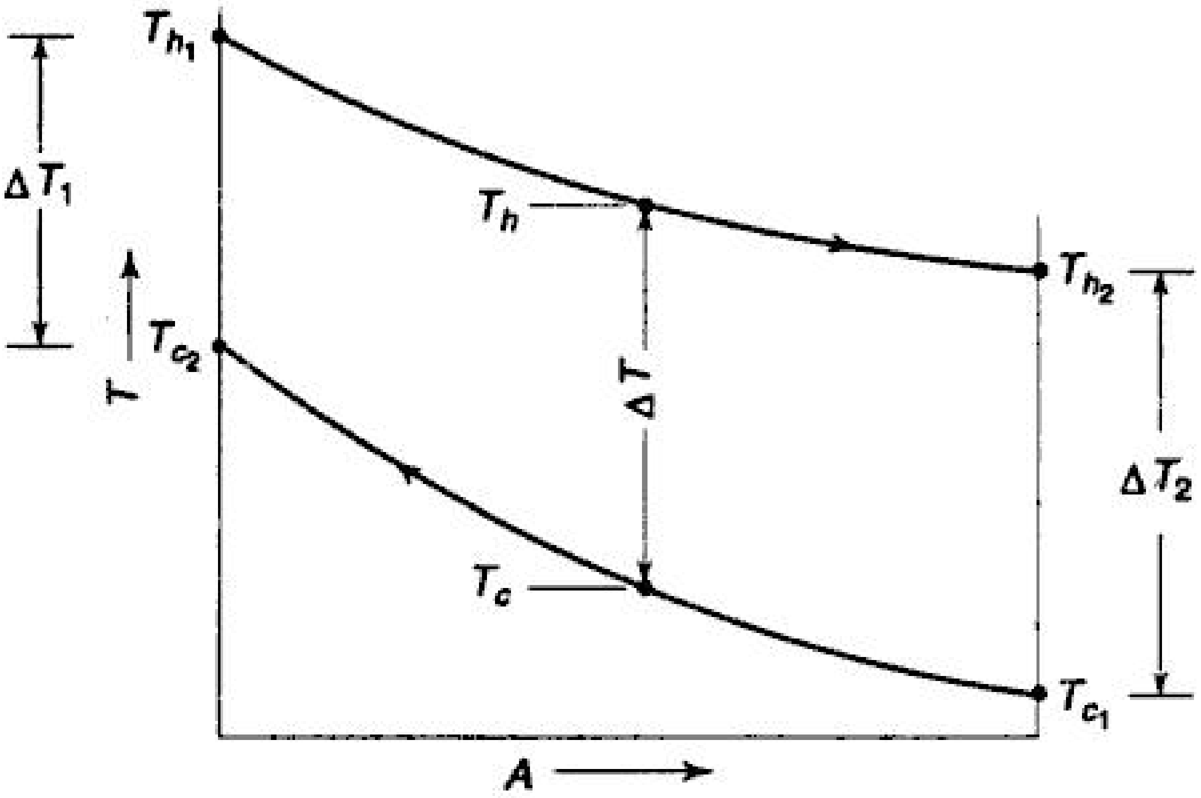


Fig. 1.30 Temperature Variation in Counter Flow Heat Exchanger

Hence, it requires smaller heat transfer surface area A than parallel flow heat exchanger for a specified heat transfer rate.

(2) It can also be seen from Fig. 1.30 that in the case of counterflow heat exchanger, the hot fluid can be cooled to the inlet temperature of the cold fluid (from t_{h_1} to t_{c_1}), and similarly, cold fluid can be heated to the inlet temperature of the hot fluid (from t_{c_1} to t_{h_1}) at least theoretically in the limit, if large heat transfer surface area is provided. This is not possible in parallel flow heat exchanger.

Note ✍: *In the case of phase change of a fluid, the temperature on one side is constant as in the case of condensers and evaporators. In these cases, both have the same LMTD, and there is no difference between counterflow and parallel flow.*

1.25 MASS TRANSFER

In *mass transfer*, we deal with the movement of species in a multicomponent system. The driving potentials for the transfer of mass can be obtained in various ways. But here we shall confine our discussion to *molecular diffusion* and *convective mass transfer*.

1.25.1 Molecular Diffusion: Fick's Law

Mass transfer by molecular diffusion of a species through another stationary medium is analogous to heat transfer by conduction. *Fick's Law* relates the *diffusion rate* or *mass flow* $\dot{m}_A$ of a species A to its driving potential which is the *concentration gradient* $\partial C_A/\partial x$ as follows in Eq. (1.76)

$$\frac{\dot{m}_A}{A} = -D\,\frac{\partial C_A}{\partial x} \tag{1.76}$$

where A is the normal area and D is called the *diffusion coefficient* or *mass transfer diffusivity*. The unit of diffusion coefficient is found to be $\mathrm{m^2\,s^{-1}}$. It has the dimensions of kinematic viscosity $\nu = \mu/\rho$ or thermal diffusivity $\alpha = k/\rho C$.

1.25.2 Convective Mass Transfer

In molecular diffusion, bulk velocities are insignificant. Most physical applications of mass transfer involve the bulk motion of fluids. This gives rise to convective mass transfer which is similar to convective heat transfer. Accordingly, we define the *mass transfer coefficient* h_M by the relation

$$\frac{\dot{m}_A}{A} = h_M\,(C_w - C_\infty) = \frac{k_\omega}{\rho}\,(\omega_w - \omega_\infty) \tag{1.77}$$

where C_w and C_∞ denote the concentrations of the species A at the wall and free stream. We see that h_M has the units of $\mathrm{m\,s^{-1}}$. The expression $\rho\ h_M$ is replaced by *diffusion coefficient* k_ω, and C by specific humidity ω in the case of diffusion of water vapour into air. Its use is illustrated in Chapters 14 and 20. It is seen that $k_\omega = \rho h_M$ if $\Delta C = \Delta\omega$, which is a satisfactory assumption (see Chap. 14).

1.26 ANALOGY BETWEEN MOMENTUM, HEAT AND MASS TRANSFER

D has the same significance in mass transfer as ν in momentum transfer and α in heat transfer. Dimensionsless numbers can be formed from the ratio of any two of these properties.

Prandtl number (as defined earlier for convective heat transfer)

$$\mathbf{Pr} = \frac{\nu}{\alpha} = \frac{C_p\,\mu}{k}$$

Lewis number

$$\mathbf{Le} = \frac{\alpha}{D} = \frac{k}{\rho C_p D}$$

Schmidt number

$$\mathbf{Sc} = \frac{\nu}{D} = \frac{\mu}{\rho D} = \frac{\text{Dynamic viscosity}}{\text{Mass diffusivity}}$$

where the product ρD is termed as *mass diffusivity*. It has the units of $\mathrm{kgm^{-1}\,s^{-1}}$.

1.26.1 Analogy between Momentum and Heat Transfer

Then if $\nu = \alpha$ or Pr = 1, the velocity and temperature distributions in the flow will be the same. In that case, the heat transfer coefficient h can be determined from the knowledge of the friction factor f by the similarity relation called *Reynolds analogy*, viz.,

$$\mathrm{St} = \frac{f}{2} \tag{1.78}$$

where St denotes the *Stanton number* defined by

$$\mathrm{St} = \frac{\mathrm{Nu}}{\mathrm{Re}\cdot\mathrm{Pr}} = \frac{h}{\rho u C_p} = \frac{h}{G C_p}$$

Here $G = \rho u$ is the *mass velocity*.
To account for some variation in Pr, Eq. (1.78) is modified to

$$j_H = \mathrm{St}\cdot\mathrm{Pr}^{2/3} = \frac{f}{2}\left(= \frac{h}{G C_p}\,\mathrm{Pr}^{2/3}\right) \tag{1.79}$$

which is called the *Colburn analogy*, and j_H is the *Colburn j-factor* for heat transfer.

1.26.2 Analogy between Momentum and Mass Transfer

Similarly, the velocity and concentration profiles will have the same shape if $\nu = D$, or Sc = 1. Thus, the Schmidt number plays the same role in mass transfer as does the Prandtl number in heat transfer. An equivalent of the Nusselt number in mass transfer is the *Sherwood number* defined by

$$\mathrm{Sh} = \frac{h_M L}{D} = \frac{k_\omega L}{\rho D}$$

where L is any characteristic length. The convective mass transfer correlations are, therefore, expressed in the form

$$\mathrm{Sh} = f(\mathrm{Re}, \mathrm{Sc})$$

similar to forced convection heat transfer correlations which are usually expressed in the form Nu $= f(\mathrm{Re}\cdot\mathrm{Pr})$.

The Stanton number can then be replaced by the number

$$\frac{\mathrm{Sh}}{\mathrm{Re}\cdot\mathrm{Sc}} = \frac{h_M}{u} = \frac{\rho h_M}{G} \tag{1.80}$$

in mass transfer correlations, as given in Eq. (1.80). And similar to Eq. (1.78), we can write for mass transfer factor j_M as

$$j_M = \left(\frac{\mathrm{Sh}}{\mathrm{Re}\cdot\mathrm{Sc}}\right)\mathrm{Sc}^{2/3} = \left(\frac{\rho h_M}{G}\right)\mathrm{Sc}^{2/3} = \frac{f}{2} \tag{1.81}$$

The numerical value of the Schmidt number for the diffusion of water vapour into air at 25°C and 1 atm pressure is 0.6.

1.26.3 Analogy between Heat and Mass Transfer

Combining Eqs (1.78) and (1.80) to eliminate $f/2$, we obtain a similarity relation between heat transfer and mass transfer coefficients as follows in Eq. (1.82)

$$\frac{h}{h_M} = \rho C_p \left(\frac{\text{Sc}}{\text{Pr}}\right)^{2/3}$$

$$= \rho C_p \left(\frac{\alpha}{D}\right)^{2/3}$$

$$= \rho C_p \,\text{Le}^{2/3} \tag{1.82}$$

where Le = α/D is the *Lewis number.*

If Le = 1, the temperature and concentration profiles are the same and then $j_M = j_H$. Incidentally for air and water vapour mixtures at atmospheric pressure, the value of the Lewis number is approximately equal to unity. This simplifies the design of air-conditioning equipment. Writing h as f_g for heat transfer coefficient of air, and h_M as k_ω/ρ for diffusion of water vapour in air, we have for Le = 1

$$k_\omega = \frac{f_g}{C_p}$$

as used in Chapters 14 and 16.

Empirical relations for Sh and Nu for air over water droplets can be expressed as follows:

$$\text{Sh} = \frac{h_M L}{D} = \frac{k_\omega L}{\rho D} = 2 + 0.6\ \text{Re}^{0.5}\ \text{Sc}^{0.33} \tag{1.83a}$$

$$\text{Nu} = \frac{hL}{k} = 2 + 0.6\ \text{Re}^{0.5}\ \text{Pr}^{0.33} \tag{1.83b}$$

The average value of diffusivity D of water vapour in air is $2.495 \times 10^{-5}\ \text{m}^2\ \text{s}^{-1}$.

Example 1.17 **Mass Transfer Coefficient of Water Vapour in Air.**

Air at 35°C and 1 atm flows at a mean velocity of 30 m/s over a flat plate 0.5 m long. Calculate the mass transfer coefficient of water vapour from the plate into air. Assume the concentration of vapour in air as very small. The diffusion coefficient of water vapour into air is $0.256 \times 10^{-4}\ \text{m}^2\ \text{s}^{-1}$. The Colburn j-factor for heat transfer coefficient is given by $j_H = 0.0296\ \mathbf{Re}^{-0.2}$.

Solution Properties of air at 35°C, 1 atm

$$\rho = 1.146\ \text{kg. m}^{-3}$$

$$C_p = 1.006\ \text{kJ. kg}^{-1}\ \text{K}^{-1}$$

$$\mu = 2 \times 10^{-5}\ \text{kg. m}^{-1}\ \text{s}^{-1}$$

$$\text{Pr} = 0.706$$

$$\text{Sc} = \frac{\mu}{\rho D} = \frac{2 \times 10^{-5}}{(1.146)\,(0.256 \times 10^{-4})} = 0.682$$

$$\text{Re} = \frac{Lu\rho}{\mu} = \frac{(0.5)\,(30)\,(1.146)}{2 \times 10^{-5}} = 859{,}600$$

$$j_M = j_H = 0.0296\ \text{Re}^{-0.2} \tag{1.84}$$

$$= 0.0296\ (859{,}600)^{-0.2} = 1.925 \times 10^{-3}$$

$$h_M = j_M\, u/(\text{Sc})^{2/3}$$

$$= \frac{(0.001925)\,(30)}{(0.682)^{2/3}} = 0.075\ \text{m. s}^{-1}$$

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Chapter 2

Refrigerating Machine and Reversed Carnot Cycle

2.1 REFRIGERATING MACHINES

There are essentially two categories of thermal plants. These are:

(i) Thermal power plants or *work producing plants.*
(ii) Refrigeration/heat pump plants or *work consuming plants.*

The work producing plants or *heat engines* lead to the conversion of heat to work. The work consuming plants, viz., *refrigerators/heat pumps*, are not those which are in any way related to the conversion of work into heat. No ingenuity at all is required for the conversion of work into heat. In fact, all work (mechanical/electrical energy) that is consumed in machinery is ultimately dissipated as heat to the environment. The objective of work consuming plants, actually, is to lead to the flow of heat from a low temperature body to a high temperature body. The work is consumed to achieve this.

Examples of common work consuming plants, viz., refrigerators are the following:

Cold storages. Central air conditioning plants. Domestic refrigerators. Room air conditioners. Ice Plants. Food freezing and freeze-drying plants. Air liquefaction plants. etc.

Heat pumps are heating plants. But they operate in the same way as refrigerators.

Refrigeration equipment, in general, is relatively smaller in size as compared to work producing plants. A very large super cold storage or a central air conditioning plant for a multistoreyed building may consume power in the range of 2000 to 5000 kW. A window-type room air conditioner may consume only 2.5 kW of power, and a domestic refrigerator just 100 to 250 W only.

2.2 A REFRIGERATING MACHINE—THE SECOND LAW INTERPRETATION

A *refrigerating machine* is a device which will either cool or maintain a body at a temperature below that of the surroundings. Hence, heat must be made to flow from a body at low temperature to the surroundings at high temperature.

However, this is not possible on its own. We see in nature that heat *spontaneously* flows from a high temperature body to a low temperature body.

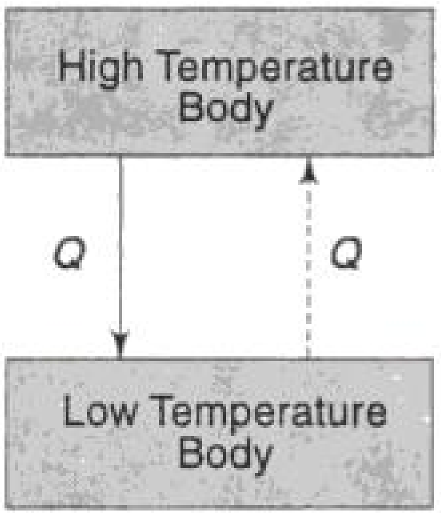

Fig. 2.1 A Thermodynamic Cycle Involving Heat Transfer Alone: Not Possible

The reverse process to complete the thermodynamic cycle, in which heat Q will flow back from the low temperature body to the high temperature body, is not possible. Thus, we see that a thermodynamic cycle involving heat transfer alone as shown in Fig. 2.1 cannot be devised. The logical conclusion is that there must be a process in which some work is done.

The second law of thermodynamics, like the first law, is based on the observations of actually existing processes and devices in nature. Figure 2.2 shows the schematic diagram of an actual refrigeration system which works on the well-known *vapour compression cycle*. Most refrigeration devices/plants, including air conditioners and refrigerators such as the ones illustrated in Chap. 1, work on this cycle only.

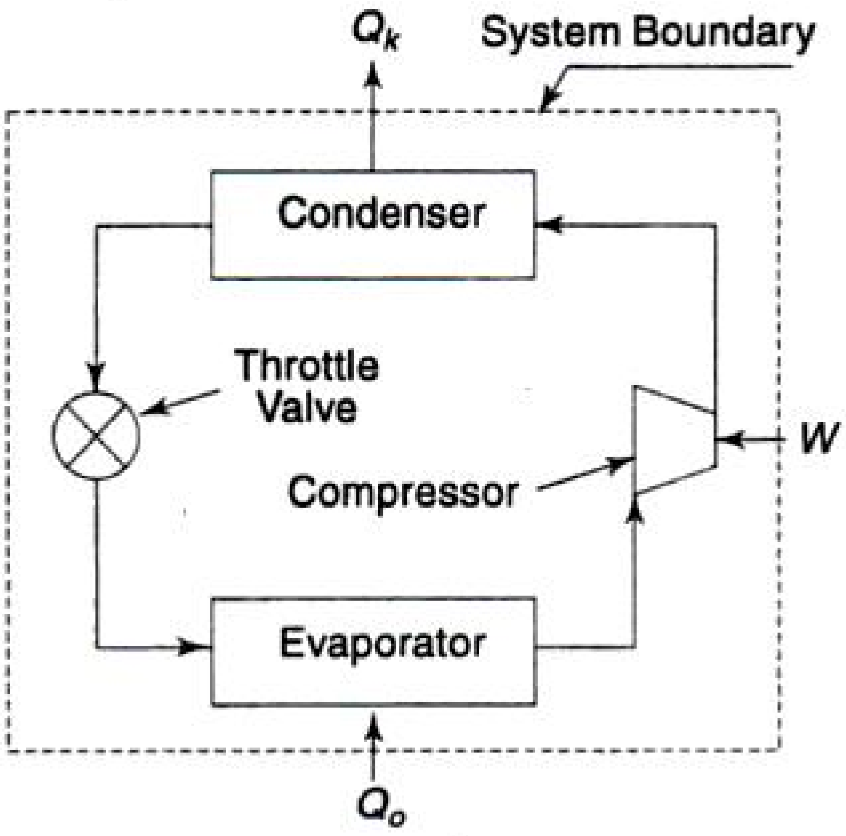

Fig. 2.2 Illustration of an Actual Refrigerator/Heat Pump: The Simple Vapour Compression System

The heat and work interactions of the processes of the cycle are as follows:

(i) Heat Q_0 is absorbed in the evaporator by the evaporation of a liquid refrigerant at a low pressure p_0, and corresponding low saturation temperature T_0.

(ii) The evaporated refrigerant vapour is compressed to a high pressure p_k in the compressor consuming work W. The pressure after compression is such that the corresponding saturation temperature T_k is higher than the temperature of the surroundings.

(iii) Heat Q_k is then rejected in the condenser to the surroundings at high temperature T_k.

The application of the first law, $\oint \delta Q = \oint \delta W$, to the cycle gives:

$$-Q_k + Q_o = -W$$
$$Q_k - Q_o = W$$

This, also, represents an *energy balance* of the system in Fig. 2.2 obtained by drawing a boundary around it.

There are two statements of the second law of thermodynamics, the *Kelvin-Planck statement*, and the *Clausius statement*. The Kelvin-Planck statement pertain to heat engines such as E represented in Fig. 2.3.

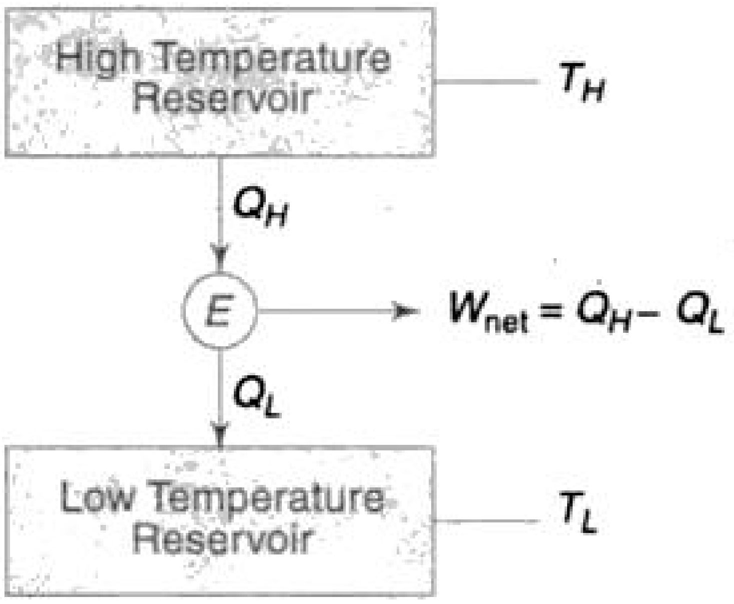

Fig. 2.3 Schematic Representation of a Heat Engine

The Clausius statement pertains to refrigerators/heat pumps. The above observation from illustration of actually existing refrigerators/heat pumps leads to the Clausius statement which is as follows:

"*It is impossible to construct a device which will operate in a cycle and produce no effect other than the transfer of heat from a low temperature body to a high temperature body*".

The statement implies that a refrigerator *R* of the type shown in Fig. 2.4 which will absorb heat Q_o from a low temperature body and transfer it to a high temperature body is impossible.

The only alternative is that there must be some work input W_{in}. Accordingly, we obtain a schematic representation of a refrigerating machine/heat pump as shown in Fig. 2.5, and from the first law $W_{in} = Q_k - Q_o$. Accordingly, heat transferred Q_k is more than heat absorbed Q_o by the amount of work input W_{in}.

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after evaporation at low pressure will have to be brought back to the initial state of high pressure liquid again. That will mean forming a complete thermodynamic cycle.

The Clausius statement eliminates the possibility of obtaining refrigeration without doing work. The statement necessitates a further clarification regarding heat-operated refrigerating machines such as the vapour absorption type or ejector type, using heat directly to produce refrigeration. Such systems may be considered as a combination of a heat engine and a refrigerating machine. The heat engine part of the system utilizes heat from a body at a higher temperature than the surroundings and delivers the required mechanical work, ***within the system***, which is directly used by the refrigerating machine part. Thus the usual process of the conversion of thermal energy, first into work (or electrical energy) and then its utilization in a refrigerating machine, is replaced by a combined process.

2.3 HEAT ENGINE, HEAT PUMP AND REFRIGERATING MACHINE

It may be concluded from the preceding discussion that a reversible heat engine may be converted into a refrigerating machine by running it in the reversed direction. Schematically, therefore, a refrigerating machine is a reversed heat engine which can be seen by comparing Figs 2.5 and 2.3.

As for the *heat pump*, there is no difference in the cycle of operation between a refrigerating machine and a heat pump. The same machine can be utilized either

(i) to absorb heat from a cold body (a cooled space) at temperature T_o and reject it to the surroundings at temperature $T_k \geq T_a$, or

(ii) to absorb heat from the surroundings at temperature $T_o \leq T_a$ and reject it to a hot body (a heated space) at temperature T_k,

where T_a is the temperature of the surroundings.

Figure 2.7 illustrates the manner of application of heat engine *E*, heat pump *H* and refrigerating machine *R*. It implies that the same machine can be used either for cooling or for heating. When used for cooling, it is called a refrigerating machine and when used for heating it is called a heat pump.

The main difference between the two is in their operating temperatures. A refrigerating machine operates between the ambient temperature $T_a \approx T_k$ and a low temperature T_o. A heat pump operates between the ambient temperature $T_a \approx T_o$ and a high temperature T_k.

Another essential difference is in their useful function. In a refrigerating machine, the heat exchanger that absorbs heat is connected to the conditioned space. In a heat pump, instead, the heat exchanger that rejects heat is connected to the conditioned space. The other heat exchanger in each case is connected to the surroundings. Thus if a refrigerating machine, that is used for cooling in summer, is to be used as a heat pump for heating in winter, it will be necessary, either

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Thus HP/TR is inversely proportional to COP for cooling. In the above derivation, imperial horsepower has been used. If the horsepower is metric, then

$$\text{HP} = \frac{\dot{W}}{0.736}$$

$$\text{HP/TR} = \frac{4.78}{\mathcal{E}_0}$$

2.3.3 Heat Pump vs. Electric Resistance Heater

It may be seen from the simple transposition of Eqs (2.1) and (2.2) that

$$\frac{Q_k}{W} = 1 + \frac{Q_o}{W}$$

$$\mathcal{E}_h = 1 + \mathcal{E}_c \tag{2.4}$$

The above relationship expresses a very interesting feature of a heat pump. According to Eq. (2.4), COP for heating is always greater than unity. It is so since Q_k is always greater then Q_o by the amount W.

Thus for the purpose of heating, it is far more economical to use a heat pump rather than an electric-resistance heater. For example, if W is the energy consumption of an electric resistance heater, the heat released to the space will be W only. But if this electrical energy W is utilized in a heat pump, the heat pumped to the space will be

$$Q_k = \mathcal{E}_h W = (1 + \mathcal{E}_c) W$$

Therefore, whatever is the value of $\mathcal{E}_c$ (even zero), Q_k will always be greater than or equal to W. The value of $\mathcal{E}_h$ for air-conditioning applications is of the order of 3. Then the heat pumped will be 3 W in a heat pump unit while the power consumption is only W. The heat pump, therefore, is a definite advancement over the simple electric-resistance heater. Only the cost of the heat pump (which is a refrigerating machine also) is prohibitive. But when an air-conditioning plant is already installed for cooling in summer, it would always be prudent to use it for heating as well in winter, operating as a heat pump.

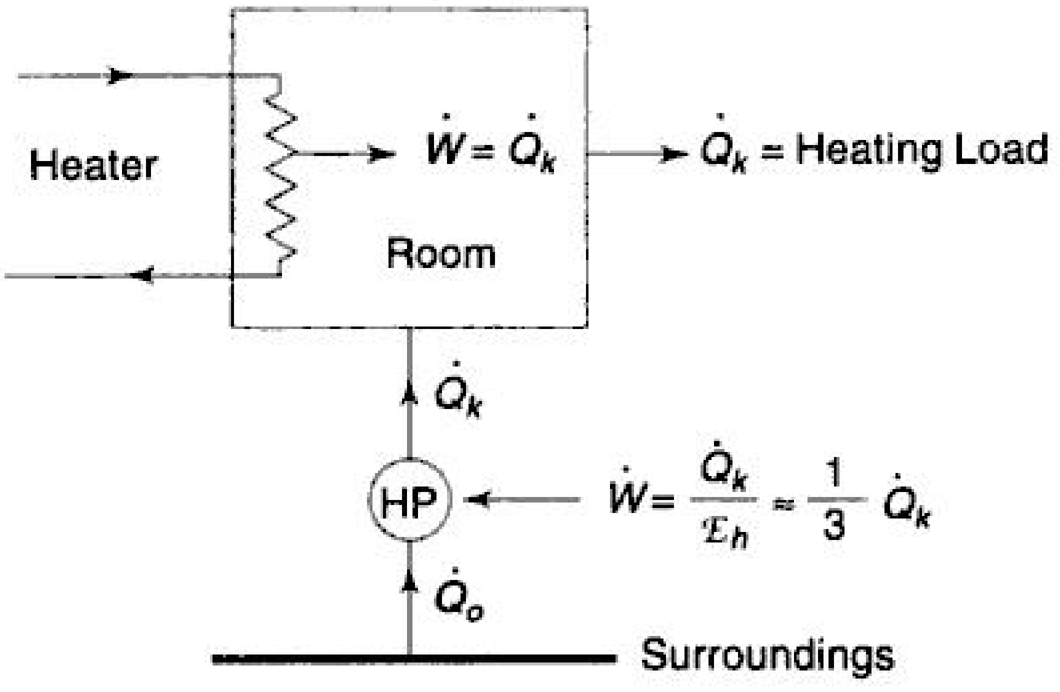

Fig. 2.8 Heating of a Room by Electric Heater and Heat Pump

Suppose a room loses $\dot{Q}_k$ kW of heat during winter. Then, it requires an equal amount of heat addition to maintain it at a desired temperature. Figure 2.8 illustrates the two methods for heating the room:

(i) Using electric resistance heater. The power consumption is

$$\dot{W} = \dot{Q}_k$$

(ii) Using heat pump. The power consumption is

$$\dot{W} = \frac{\dot{Q}_k}{\mathcal{E}_h} \cong \frac{1}{3}\dot{Q}_k$$

if $\mathcal{E}_h = 3$. Thus, power consumption of the heat pump is very much lower.

2.4 BEST REFRIGERATION CYCLE: THE CARNOT PRINCIPLE

It is possible to show that the cooling energy ratio of a refrigeration cycle working between two temperatures will be maximum when the cycle is a reversible one. For the purpose, consider a reversible refrigerating machine R and another irreversible refrigerating machine I, both working between two heat reservoirs at temperatures T_k and T_o, and absorbing the same quantity of heat Q_o from the cold reservoir at T_o as shown in Fig. 2.9 (a).

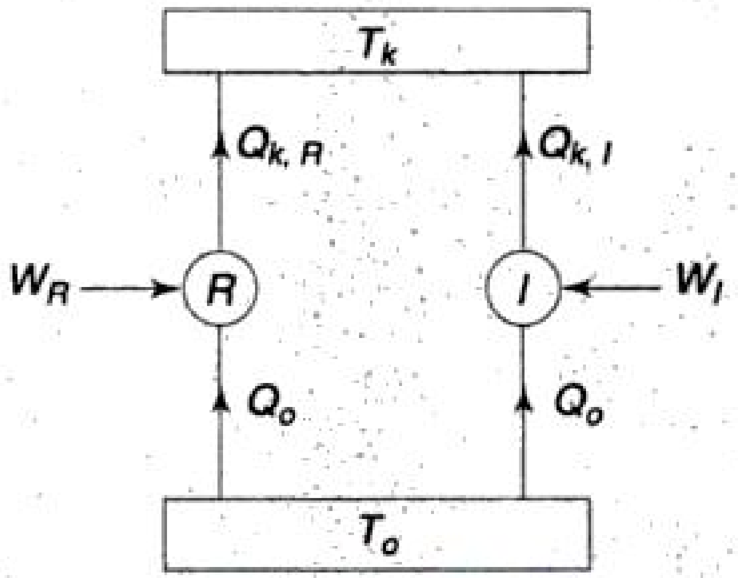

Fig. 2.9 (a) Reversible and Irreversible Refrigerating Machines

Now, to prove the contrary, let us assume that the COP of the irreversible machine is higher than that of the reversible machine, viz., $\mathcal{E}_I > \mathcal{E}_R$. Hence

$$\frac{\dot{Q}_o}{W_I} > \frac{\dot{Q}_o}{W_R}$$

$$W_R > W_1$$

And since

$$Q_{k,R} = Q_o + W_R$$
$$Q_{k,I} = Q_o + W_I$$

we have

$$Q_{k,R} > Q_{k,I}$$

and

$$Q_{k,R} - Q_{k,I} = W_R - W_I = W_{\text{net}}$$

If now, the reversible refrigerating machine is made to work as a heat engine and the irreversible refrigerating machine continues to work as a refrigerating machine, as shown in Fig. 2.9(b), the resultant combined system will work as a perpetual motion machine of the second kind taking heat equal to $Q_{k,R} - Q_{k,I}$ from the hot reservoir and converting it completely into work, thus violating the Kelvin-Planck statement of the Second Law applicable to heat engines as shown in Fig. 2.10.

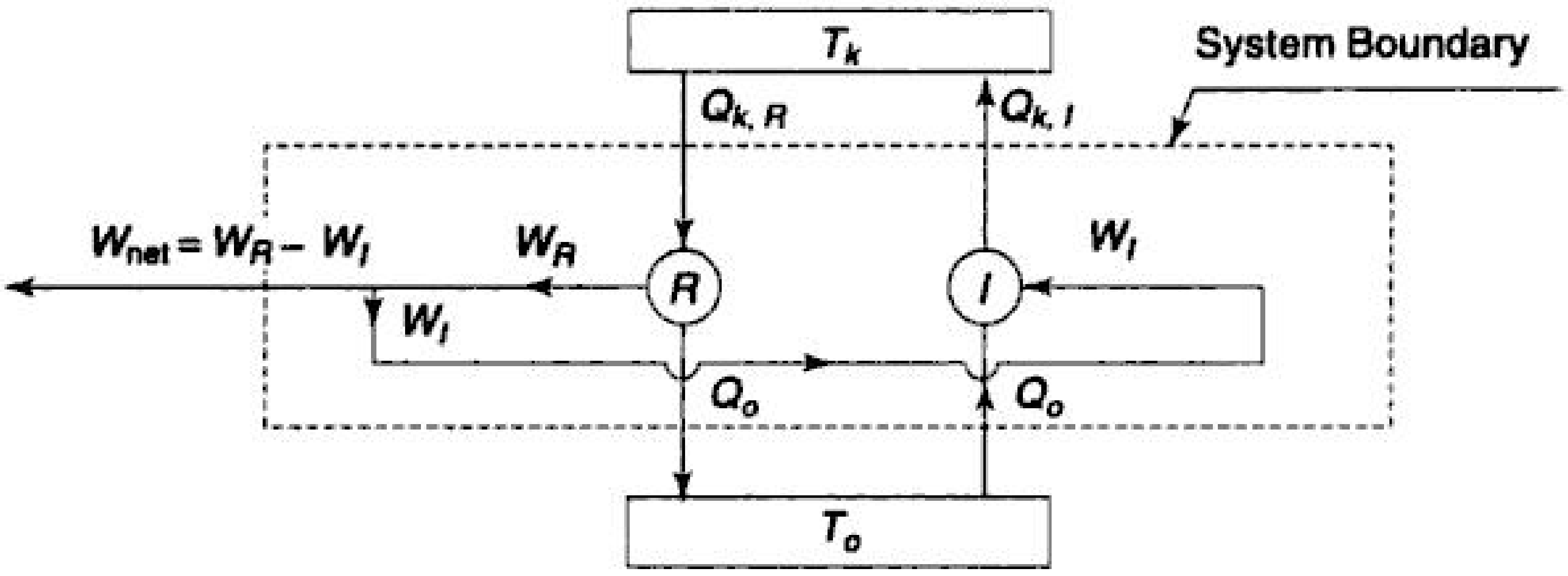

Fig. 2.9(b) Reversible Refrigerating Machine Working as a Heat Engine in Combination with an Irreversible Refrigerating Machine

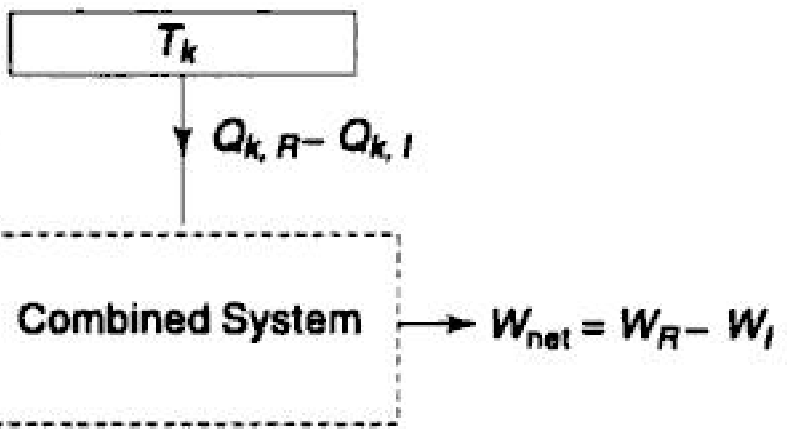

Fig. 2.10 Combined System Resulting in a Perpetual Motion Machine of the Second Kind thus Violating the Second Law

It is, therefore, concluded that a refrigeration cycle operating reversibly between two heat reservoirs has the highest coefficient of performance. Likewise, it can also be shown that all reversible refrigeration cycles have the same COP. These are two corollaries of the Second Law comprising the *Carnot Principle*.

2.4.1 Reversed Carnot Cycle

We now know that a reversible refrigeration cycle has the maximum COP. We know further that a reversible heat engine can be reversed in operation to work as a refrigerating machine.

Sadi Carnot, in 1824, proposed a reversible heat-engine cycle as a measure of maximum possible conversion of heat into work. A reversed Carnot cycle can, therefore, be employed as a reversible refrigeration cycle, which would be a measure of maximum possible COP of a refrigerating machine operating between two temperatures T_k of heat rejection and T_o of refrigeration.

A reversed Carnot cycle is shown in Fig. 2.11 on a *T–s* diagram. The cycle consists of two isothermals and two isentropics as follows:

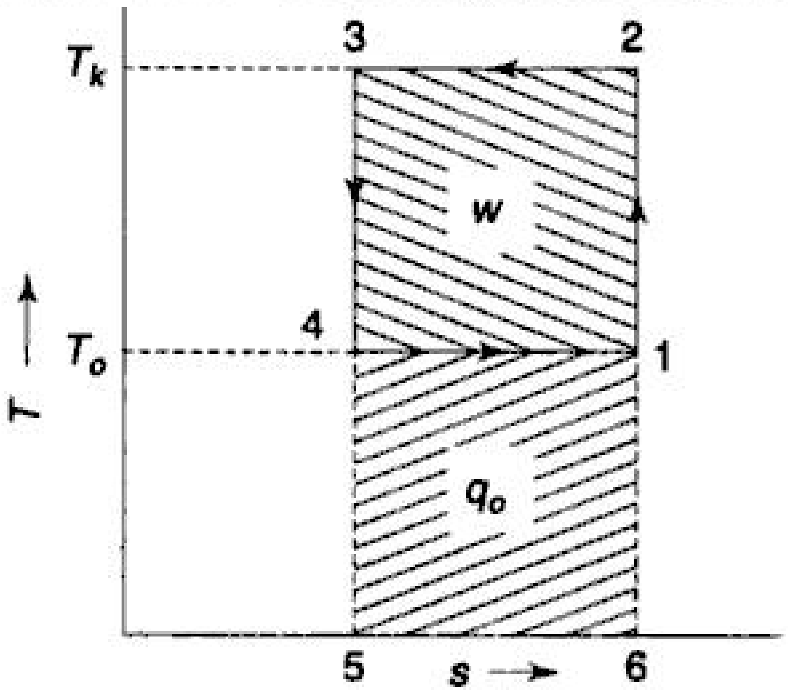

Fig. 2.11 Reversed Carnot Cycle

Process 1—2	Isentropic compression, $s_1 = s_2$
Process 2—3	Isothermal heat rejection to the hot reservoir at T_k = const.
Process 3—4	Isentropic expansion, $s_3 = s_4$
Process 4—1	Isothermal heat absorption from the cold reservoir at T_o = const.

The areas on the *T-s* diagram, representing $\int Tds$ give the heat transfers, and work done in the cycle as follows:

Heat absorbed from cold body, $Q_o = T_o \Delta S$ = area 1-4-5-6
Heat rejected to hot body, $Q_k = T_k \Delta S$ = area 2-3-5-6
Work done, $W = Q_k - Q_o = (T_k - T_o)\,\Delta s$ = area 1-2-3-4

Hence we obtain Carnot values of COP for cooling and heating as

$$\mathcal{E}_{c,\text{ Carnot}} = \frac{Q_o}{W} = \frac{T_o}{T_k - T_o} = \frac{1}{\dfrac{T_k}{T_o} - 1}$$

$$\mathcal{E}_{h,\text{ Carnot}} = \frac{Q_k}{W} = \frac{T_k}{T_k - T_o} = \frac{1}{1 - T_o/T_k}$$

Effect of Operating Temperatures We, thus, see that the Carnot COP depends on the operating temperatures T_k and T_o only. It does not depend on the working substance (refrigerant) used.

For cooling, T_o is the refrigeration temperature and T_k is the temperature of heat rejection to the surroundings. The lowest possible refrigeration temperature is $T_o = 0$ (absolute zero) at which $\mathcal{E}_c = 0$. The highest possible refrigeration temperature is $T_o = T_k$, i.e., when the refrigeration temperature is equal to the temperature of the surroundings (ambient) at which $\mathcal{E}_c = \infty$. Thus, Carnot COP for cooling varies between 0 and ∞.

For heating, T_o is the temperature of heat absorption from the surroundings and T_k is the heating temperature. Theoretically, the COP for heating varies between 1 and ∞.

It may, therefore, be noted that to obtain maximum possible COP in any application,

(i) the cold body temperature T_o should be as high as possible, and
(ii) the hot body temperature T_k should be as low as possible.

The lower the refrigeration temperature required, and higher the temperature of heat rejection to the surroundings, the larger is the power consumption of the refrigerating machine. Also, the lower is the refrigeration temperature required, the lower is the refrigerating capacity obtained.

Consider now, for example, a domestic refrigerator which produces refrigeration at –25°C (248 K). Let heat be rejected to the ambient at 60°C (333 K).

The maximum possible COP of this refrigerator would be

$$(\text{COP}_c)_{\max} = \frac{248}{333 - 248} = \frac{248}{85} = 2.9$$

Thus, the refrigerator would produce a maximum of 290 W of refrigeration per 100 W of power consumption. The most popular size, 165 L internal volume refrigerators of most manufacturers produce 89 W of refrigeration, and the power of the electric motors running these refrigerators is around 110 W. Considering that the refrigerators have a running time of 75% only, the average power consumption would come to 82.5 W. Thus, the actual COP of these machines would be only 1.08.

Compare this with the COP of a room air conditioner which produces refrigeration at a higher temperature of about 5°C (278 K). Assuming the temperature of heat rejection the same as 60°C (333 K), the maximum possible COP would be

$$(\text{COP}_c)_{\max} = \frac{278}{333 - 278} = \frac{278}{55} = 5$$

It would, thus, produce 5 kW of refrigeration per kW of power consumption. A 1.5 TR (5.3 kW) air conditioner will have a minimum power consumption of 1.05 kW. The actual air conditioner will, however, have power consumption of the order of 2.0 kW considering 75% running time. The actual COP is thus 2.6.

The above two examples show that the COP of a refrigeration system decreases, and power consumption increases as we go to lower and lower refrigeration temperatures.

2.4.2 Selection of Operating Temperatures

The selection of temperature T_0 depends on the particular application of refrigeration. Consider, for example, the simple summer air-conditioning system shown in Fig. 2.12(a). The room is maintained at temperature t_i equal to 25°C. To offset the heat entering the room, the air must be supplied at a temperature

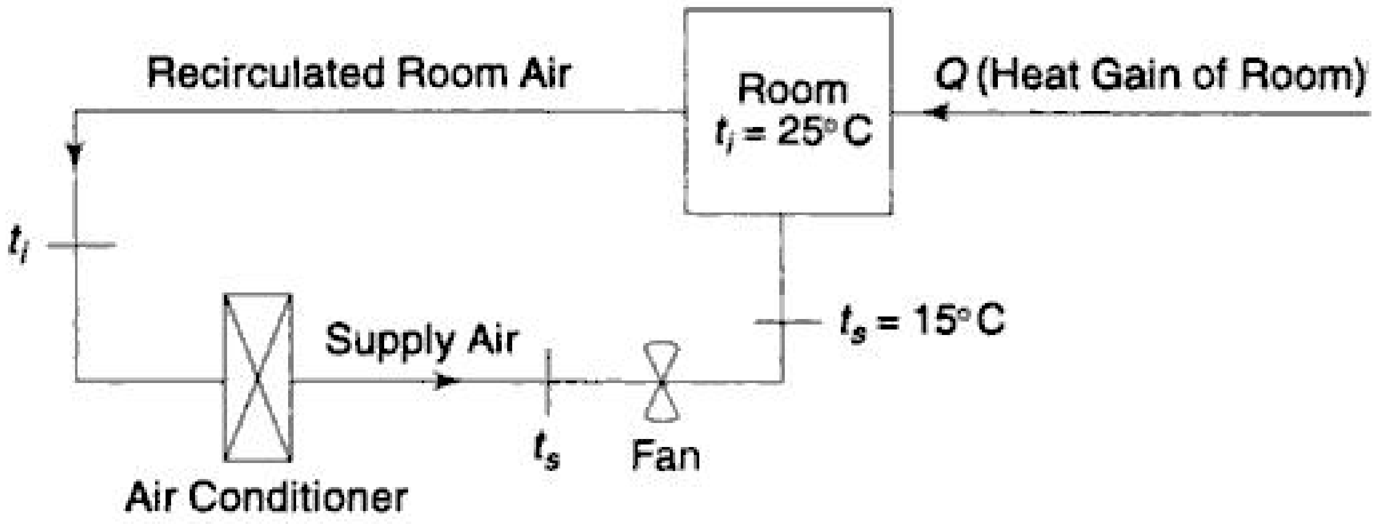

Fig. 2.12(a) Simple Summer Air-Conditioning System

lower than 25°C, at t_s equal to 15°C. The air conditioner will, therefore, cool the room air from 25°C to 15°C and then supply it back to the room. Accordingly, the refrigerant temperature t_o must be less than 15°C to absorb heat Q_o from the air maintaining a finite temperature difference Δt across the heat exchanger as shown in Fig. 2.12(b). If the temperature difference is zero, the *area requirement* of the heat exchanger will be infinite. Thus for air conditioning in summer, the temperature t_o is of the order of 0 to 10°C usually about 5°C. In a similar manner, the approximate refrigeration temperature requirements can be found out for various other applications as given in Table 2.1.

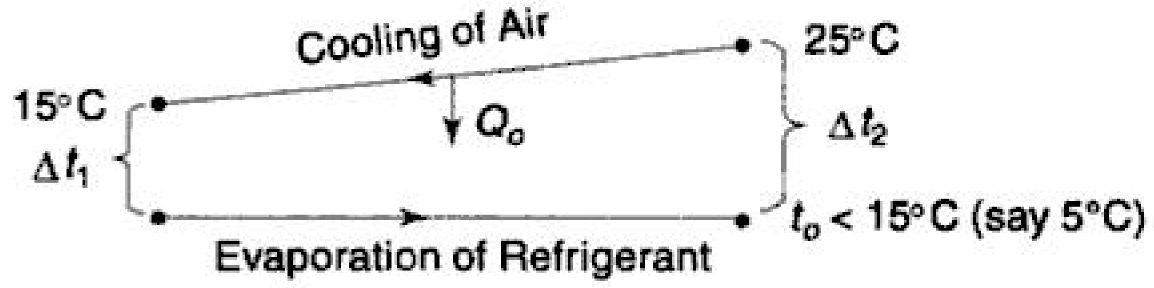

Fig. 2.12(b) Temperatures in a Cooling Coil

Table 2.1 Refrigeration Temperature Requirements of Common Applications

No.	*Application*	*Refrigeration Temperature, t_o, °C*
1.	Air conditioning in summer	0 to 10
2.	Cold storages	–10 to 2
3.	Domestic refrigerators	–25
4.	Frozen foods	–35
5.	Freeze drying and IQF (Instant Quick Freezing)	–35 to –45

The selection of temperature T_k depends on the surrounding medium used for heat rejection. There are three possible media in the surroundings to which heat Q_k may be rejected, viz., air, water and ground. The units that use air as a cooling medium are called the *air-cooled units* and those using water are called the *water-cooled units*.

Consider an air-cooled unit. Let the surrounding air temperature (in summer) be 45°C. Also, let the rise in temperature of air, after absorbing heat Q_k, be 10°C, as shown in Fig. 2.13(a). Hence the temperature of heat rejection t_k has to be greater than 55°C, say 65°C, so that the temperature differentials Δt_1 and Δt_2, across the heat exchanger (HE) are 20°C and 10°C respectively, and the arithmetic mean temperature difference is (20 + 10)/2 = 15°C.

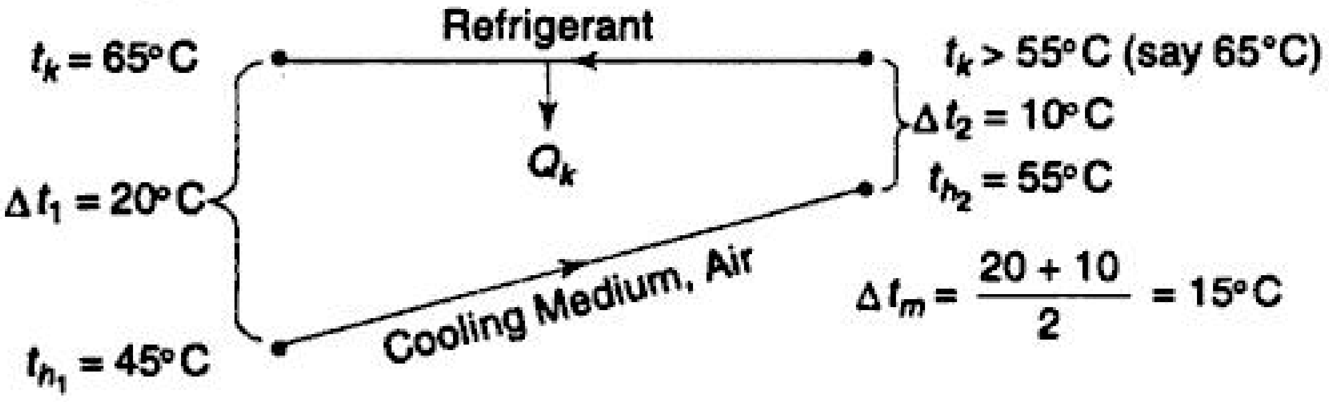

Fig. 2.13(a) Temperatures During Heat Rejection in an Air-Cooled Refrigerating Machine

Air is widely used as a cooling medium in small refrigerating machines such as refrigerators, water coolers, window-type air conditioners and small-package units.

Water as a cooling medium is preferable to air as it affords a lower value of T_k because of the following reasons:

(i) It is available at a temperature lower than that of air. Its temperature approaches the wet bulb temperature of the surrounding air. This is the limiting temperature to which heated water can be cooled in a cooling tower or a spray pond.

(ii) The specific heat of water is about four times that of air. Thus, for the same heat rejection Q_k and the same mass flow, the temperature rise of water is one-fourth that of air and correspondingly T_k is lower.

(iii) Water has a higher heat transfer coefficient than air mainly because of its high thermal conductivity, say, 5000 as against 100 in forced convection and 10 in free convection for air in $\mathrm{W m^{-2} K^{-1}}$. Thus, for the same heat rejected Q_k and the same area of the heat exchanger, the temperature difference Δt required across the heat exchanger is less which also results in a lower value of T_k.

***Note* ✍:** *The lower value of heat transfer coefficient of air is made up by providing extended surface on air-side in actual equipment.*

Consider, for example, a water-cooled unit, in place of the air-cooled unit of the preceding example. Let the wet bulb temperature of the surrounding air be 28°C (even though the dry bulb temperature may be 45°C). Then the temperature of water from the cooling tower may be taken to be about 30°C. With the same mass flow for air and water, the temperature rise of water as a result of its higher specific heat will be 10/4 = 2.5°C only, though taken as 5°C in design as shown in Fig. 2.13 (b). The temperature of water leaving the heat exchanger is then 35°C.

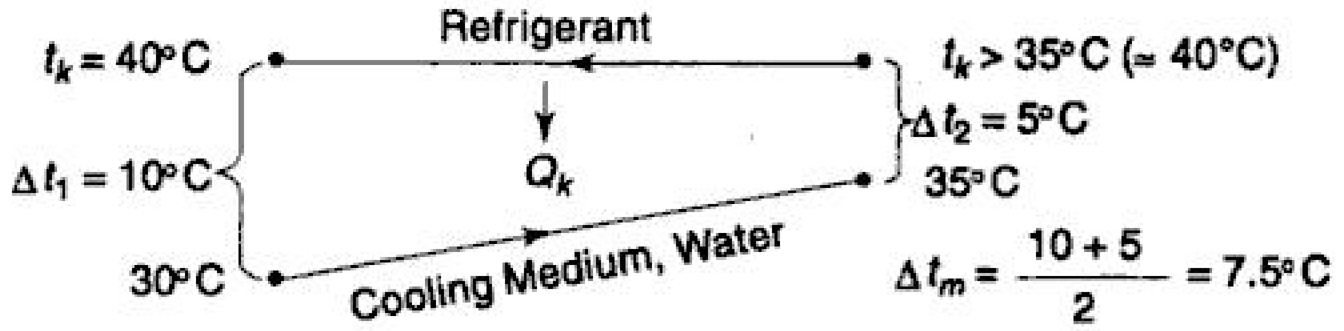

Fig. 2.13(b) Temperatures During Heat Rejection in a Water-Cooled Refrigerating Machine

Let the mean temperature difference required with water be 7.5°C instead of 15°C for air as a consequence of the higher heat transfer coefficient. Then the temperature of heat rejection required need only be 40°C for water as compared to 65°C for air. Normal design temperature T_k in Delhi is 43°C .

Thus, the use of water as a cooling medium results in a lower T_k, higher COP and lower power consumption in a refrigeration plant. At the same time, we have much more compact and smaller condenser even with a smaller value of Δt_m (7.5°C with water as against 15°C with air in the above illustration). Large refrigeration plants, including central air-conditioning plants, therefore, are invariably water-cooled as the saving in the cost of power consumption exceeds the added cost of a water-cooling plant (cooling tower, pump, piping, etc.). Examples 2.1 and 2.2 amply illustrate the advantages of water over air, as a cooling medium. Although, the calculations are based on the reversed Carnot cycle for the working substance, yet finite temperature differentials have been assumed in the heat exchangers as external irreversibilities.

The ground is also sometimes used as a heat sink, specially in locations where extreme high and low temperatures are reached and the ground temperatures are lower in summer and higher in winter as compared to the surrounding temperatures.

Example 2.1

(a) The ambient air temperatures during summer and winter in a particular locality are 45°C and 15°C respectively. Find the values of Carnot COP for an air conditioner for cooling and heating, corresponding to refrigeration temperatures of 5°C for summer and heating temperature of 55°C for winter. Assume suitable temperature differences in the exchanger that exchanges heat with the surroundings.

(b) If water from the cooling tower at 30°C is used as a cooling medium with 3°C temperature differential for air-conditioning in summer, what will be the Carnot COP for cooling?

(c) Also, find the theoretical power consumption per ton of refrigeration in each case. Assume no increase in the temperature of the surrounding air or water.

Solution

(a) *Air as a cooling medium*

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$$t_{h_2} = 30 + 2.26 = 32.26°\text{C}$$

LMTD in the heat exchanger

$$\Delta t_m = \frac{(35 - 30) - (35 - 32.26)}{\ln \dfrac{35 - 30}{35 - 32.26}} = 3.76°\text{C}$$

$$A = \frac{\dot{Q}_k}{U\Delta t_m} = \frac{3.9}{2.5\,(3.76)} = 0.41 \text{ m}^2$$

Checking with the actual area of 2.975 m^2, it is seen that temperature t_k could be much lower.

Second approximation of $t_k = 33°\text{C}$ gives $A = 0.95 \text{ m}^2$.

The area required is still one-fourth of the available area of 2.975 m^2. Thus, even if t_k is taken equal to $t_{h_2} = 32.24$ so that $\Delta t_2 = 0$, then $\Delta t_1 = 32.24 - 30 = 2.24°\text{C}$, and $\Delta t_m = 1.12°\text{C}$, it gives an area requirement of only (1.63/1/12) (0.95) = 1.38 m^2 which is still less than 2.975 m^2.

Hence the area of the heat exchanger is more than adequate with water as a cooling medium and can be reduced to at least one-third.

(c) *Water as a cooling medium* ($A = 1 \text{ m}^2$)

By iteration, we find the solution, viz.,

$$t_k = 32.94°\text{C}$$

$$\text{COP} = \frac{273 + 5}{32.94 - 5} = 9.95$$

$$\dot{Q}_k = 3.5167\left(1 + \frac{1}{9.76}\right) = 3.87 \text{ kW}$$

$$\Delta t_h = \frac{3.87}{0.413\,(4.1868)} = 2.24°\text{C}$$

$$t_{h_2} = 32.24°\text{C}$$

$$\Delta t_m = \frac{(32.9 - 30) - (32.9 - 32.24)}{\ln \dfrac{32.9 - 30}{32.9 - 32.24}} = 1.5°\text{C}$$

$$A = \frac{3.87}{2.5\,(1.5)} = 1.0 \text{ m}^2$$

Power consumption, $\dot{W} = \dfrac{3.5167}{9.95} = 0.35 \text{ kW}$

Saving in power consumption $= \dfrac{0.63 - 0.35}{0.63} \times 100 = 44.4\%$

Saving in area of the heat exchanger

$$= \frac{2.975 - 1}{2.975} \times 100 = 66\%$$

Note ✍: Indiscriminate use of air-cooled window-type air conditioners is wasteful of energy and equipment. Instead of using large number of window units in a big building, it is much more desirable to install a central air conditioning plant which is always water-cooled. This will save power and reduce cost. It will also minimise thermal pollution of the environment by diminishing $\dot{W}$ and $\dot{Q}_k$.

2.5 VAPOUR AS A REFRIGERANT IN REVERSED CARNOT CYCLE

The reversed Carnot cycle can be made almost completely practical by operating in the liquid-vapour region of a pure substance as shown in Fig. 2.14.

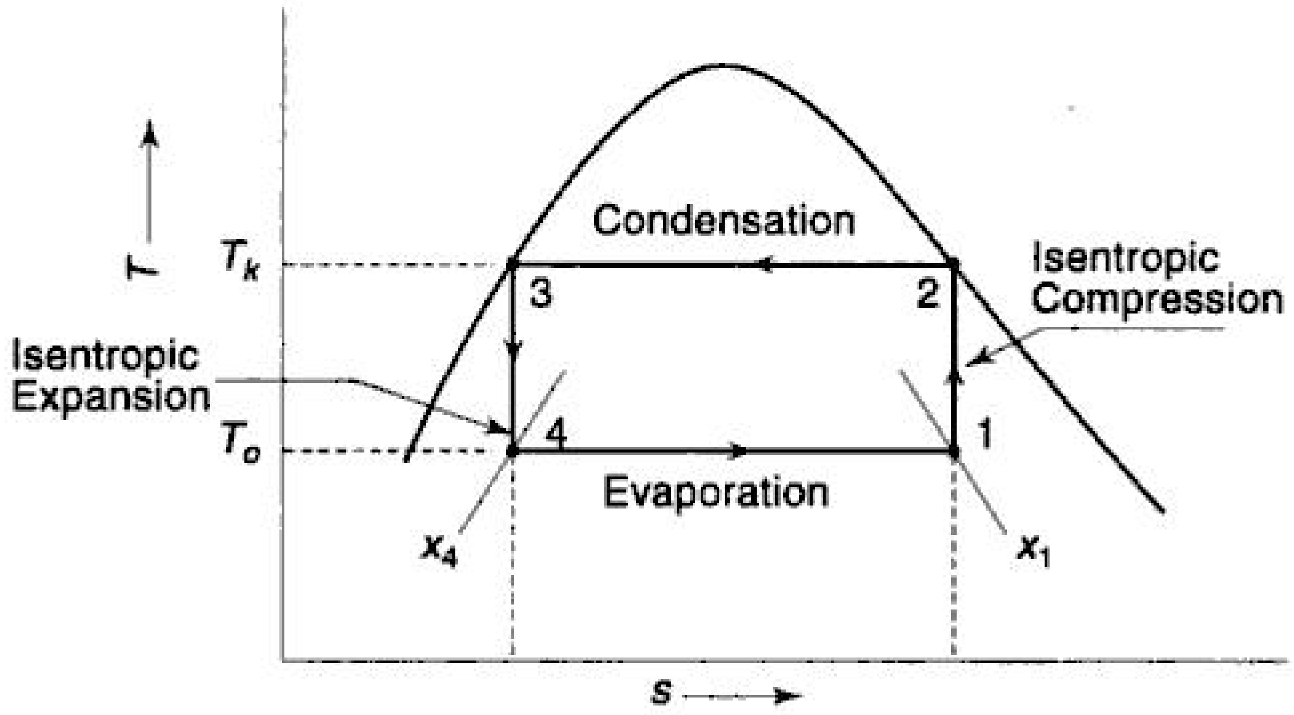

Fig. 2.14 Reversed Carnot Cycle with Vapour as a Refrigerant

The isothermal processes of heat rejection (2-3) and heat absorption (4-1) of the Carnot cycle are achieved by making use of the phenomena of condensation and evaporation of a pure substance at constant pressure and temperature. This alternate condensation and evaporation of a working substance is accompanied by alternate isentropic compression (1-2) and expansion (3-4) processes. It may be noted that the vapour during compression is wet although it is dry-saturated at the end of the process. Such a compression is called *wet compression*. It may also be seen that the isentropic expansion of the liquid from 3 to 4 results in flashing of the refrigerant with consequent temperature drop from T_k to T_o although such expansion of a liquid with partial vaporization is practically difficult to achieve in a fast-moving piston and cylinder mechanism.

The thermodynamic analysis per unit mass of the refrigerant, for the four flow processes of the cycle, using steady-state steady-flow energy equation is as follows:

Refrigerating effect, $q_o = q_{4-1} = w_{4-1}$ (for a perfect gas)

Note ✍: *Work of isentropic compression and expansion is the same. Hence Net work of the cycle.*

$$w = |w_{2-3}| - w_{4-1} = RT_k \ln \frac{v_2}{v_3} - RT_o \ln \frac{v_1}{v_4} \qquad (2.10)$$

Refrigerating effect, $q_o = q_{4-1} = RT_o \ln \dfrac{v_1}{v_4}$ (2.11)

Now for the isentropic processes 1-2 and 3-4

$$\frac{T_k}{T_o} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

and

$$\frac{T_k}{T_o} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = r^{\gamma-1}$$

where r is the compression ratio for the isentropic processes.

Hence $\dfrac{p_2}{p_1} = \dfrac{p_3}{p_4}$, and $r = \dfrac{v_1}{v_2} = \dfrac{v_4}{v_3}$

or $\dfrac{v_2}{v_3} = \dfrac{v_1}{v_4} = \dfrac{p_3}{p_2} = \dfrac{p_4}{p_1}$

since $p_2v_2 = p_3v_3$ and $p_4v_4 = p_1v_1$.

We have, for the COP for cooling

$$\mathcal{E}_c = \frac{q_o}{w} = \frac{RT_o \ln \dfrac{v_1}{v_4}}{RT_k \ln \dfrac{v_2}{v_3} - RT_o \ln \dfrac{v_1}{v_4}}$$

$$= \frac{T_o}{T_k - T_o}$$

$$= \frac{1}{\dfrac{T_k}{T_o} - 1} = \frac{1}{r^{\gamma-1} - 1} \qquad (2.12)$$

Thus COP is a function of compression ratio only.

2.7 LIMITATIONS OF REVERSED CARNOT CYCLE

It is found that serious practical difficulties are encountered in the application of Carnot cycle.

In the reversed Carnot cycle with vapour as refrigerant, the isothermal processes of condensation and evaporation are internally reversible processes, and they are easily achievable in practice although there may be some problem in having only partial evaporation. However, isentropic compression and expansion processes have some limitations which are discussed in Chap. 3. In brief, it is difficult to design an expander to handle a mixture of largely liquid and partly vapour for the process 3-4. Also, because of the internal irreversibilities in the compressor and the expander, the actual COP of the cycle is very low, though the ideal cycle COP is the maximum. A cycle which is closest to the reversed Carnot vapour cycle is the vapour compression cycle described in Chap. 3.

There are two drawbacks of reversed Carnot cycle with gas as a refrigerant:

(i) Firstly, it is not possible to devise, in practice, isothermal processes of heat absorption and rejection, 4-1 and 2-3 in Fig. 2.15 with gas as the working substance. These are impractical as these will be infinitely slow.

(ii) Secondly, the cycle on *p-v* diagram is very narrow since the volume is changing both during the reversible isothermal and reversible adiabatic processes. Drawn correctly to scale, the Carnot *p-v* diagram is much thinner than the diagram illustrated in Fig. 2.15. As a result, the stroke volume of the cylinder is very large. The cycle, therefore, suffers from poor actual COP as a result of irreversibilities of the compressor and expander.

A gas refrigeration cycle, which is closest to reversed Carnot cycle with gas as a refrigerant, is described in Chap. 11.

2.8 ACTUAL REFRIGERATION SYSTEMS

Although the Carnot cycle is theoretically the most efficient cycle between given temperatures T_k and T_o, it has limitations for practical use. It is, therefore, found useful only as a criterion of perfection of cycle. In an actual cycle, the COPs, $\mathcal{E}_c$ and $\mathcal{E}_h$, will be less than their Carnot values. For the purpose of comparison between the actual and Carnot values, we define the *second law efficiency* or *exergetic efficiency* for cooling and heating, $(\eta_{II})_c$ and $(\eta_{II})_h$ as below:

$$(\eta_{II})_c = \frac{\mathcal{E}_c}{\mathcal{E}_{c,\text{Carnot}}}$$

$$(\eta_{II})_h = \frac{\mathcal{E}_h}{\mathcal{E}_{h,\text{Carnot}}}$$

Note that $\mathcal{E}_c$ and $\mathcal{E}_h$ are the first law COPs.

The conventional refrigeration systems work on the *vapour compression cycle* which is closest to the Carnot vapour cycle and has a high COP. *Gas cycle refrigeration* is used in aircraft refrigeration. Among the less conventional ones are the heat-operated refrigerating machines working on the *vapour absorption cycle* and *steam ejector cycle*.

There are also the low temperature refrigeration cycles, e.g., Linde cycle, Claude cycle, etc., used for the liquefaction of gases. Also, we have Philips liquefier which employs a cycle approaching the reversible Stirling cycle.

A recent development is the thermoelectric refrigeration but its COP is so poor that it cannot be exploited commercially. Temperatures approaching absolute zero have been obtained by adiabatic demagnetization on a limited scale in laboratories.

PROBLEMS

2.1 (a) A refrigerator has working temperatures in the evaporator and condenser coils of –30 and 35°C respectively. What is the maximum possible COP of the refrigerator?

(b) If the actual refrigerator has a refrigerating efficiency of 0.75, calculate the refrigerating effect in kW and TR per kW of power input.

2.2 A reversed Carnot cycle has a COP for cooling of 4. Determine the temperature ratio T_k/T_o.

If the power consumption of the cycle is 7.5 kW, determine the refrigerating capacity of the machine in *TR*.

If the cycle is used as a heat pump with the same ratio of temperatures, determine its COP for heating and the quantity of heat pumped.

2.3 A Carnot refrigerator operates with Freon 12 ($CCl_2 F_2$) as a refrigerant condensing at 50°C and evaporating at –15°C.

Find its COP using the Carnot expression as well as the properties of Freon 12. Also determine the power consumption per ton of refrigeration.

2.4 The overall volume compression ratio of a reversed Carnot cycle working with air as a refrigerant is 10. The temperature limits of the cycle are 40°C and 0°C.

Determine:

(i) the pressure, volume and temperature at each point of the cycle,
(ii) the work done in the cycle,
(iii) the refrigerating effect, and
(iv) the COP of the cycle.

2.5 An air conditioning system is operating in an ambient of 45°C. The room temperature is maintained at 25°C. Determine the power consumption of the system per ton of refrigeration if it is,

(a) air-cooled as in a window-type air conditioner;
(b) water-cooled as in a central air conditioning plant.

The cooling water from cooling tower is available at 30°C. Assume suitable operating temperatures. Actual COP of the system is only 50% of the COP of the reversible cycle.

2.6 Determine the power consumption of a domestic refrigerator if its refrigerating capacity is $\frac{1}{8}$ TR. It is operating in an ambient of 40°C. Temperature in the freezer must be maintained at –15°C. COP of the system is half the Carnot COP. Assume suitable condensing and evaporating temperatures.

Chapter 3

Vapour Compression System

3.1 MODIFICATIONS IN REVERSED CARNOT CYCLE WITH VAPOUR AS A REFRIGERANT

The reversed Carnot cycle with vapour as a refrigerant can be used as a practical cycle with minor modifications. The isothermal processes of heat rejection and heat absorption, accompanying condensation and evaporation respectively, are nearly perfect processes and easily achievable in practice. The isentropic compression and expansion processes, however, have certain limitations which are discussed below and are, therefore, suitably modified.

3.1.1 Dry Versus Wet Compression

The compression process as shown in Fig. 3.1 involves the compression of wet-refrigerant vapour at 1′ to dry-saturated vapour at 2′. It is called *wet compression*. With a reciprocating compressor, wet compression is not found suitable due to the following reasons:

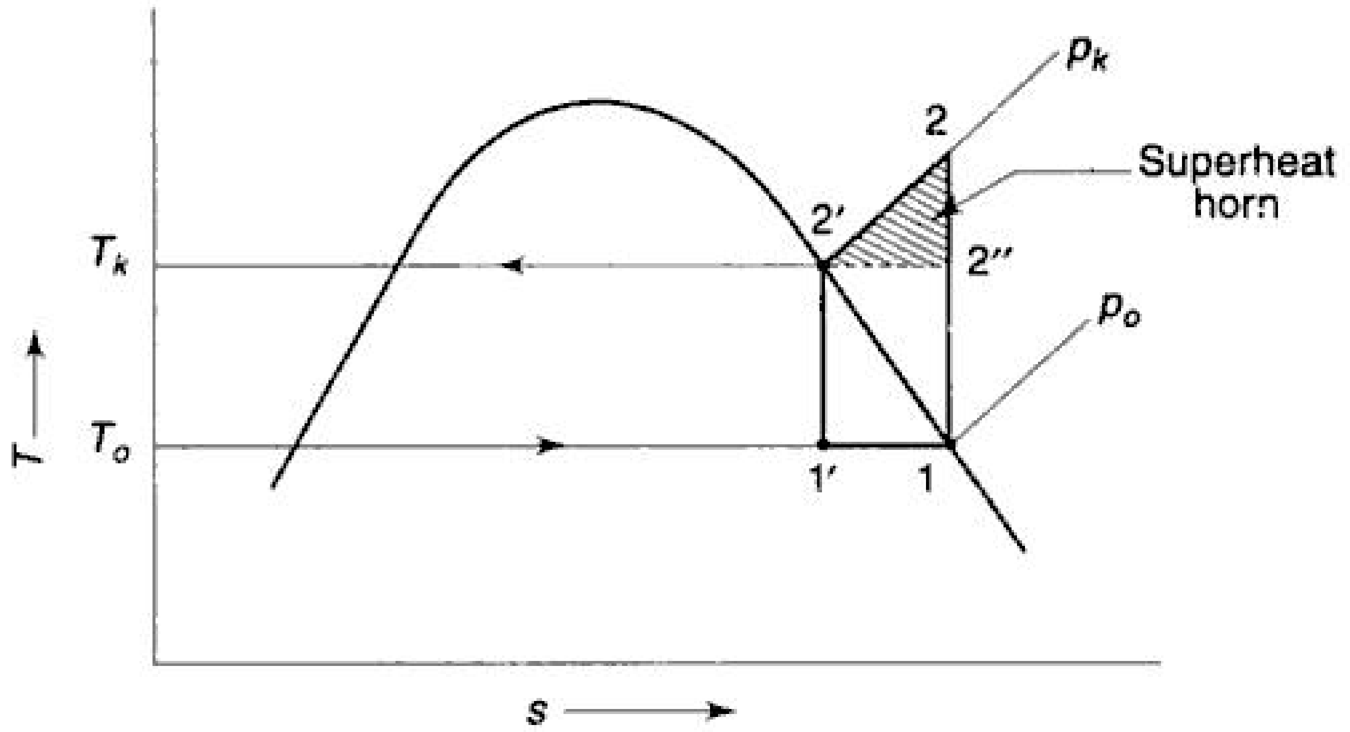

Fig. 3.1 Dry and Wet Compression Processes

(i) First, liquid refrigerant may be trapped in the head of the cylinder and may damage the compressor valves and the cylinder itself. Even though the state of vapour at the end of wet compression is theoretically dry-saturated, it is normal to expect some liquid droplets to remain suspended in the gas, as the time taken by the compression process is quite small compared to the time needed for evaporation of droplets. For example, in a modern high-speed compressor, say, running at 2800 rpm, the time available in one revolution is only 0.021 second.

(ii) Secondly, liquid-refrigerant droplets may wash away the lubricating oil from the walls of the compressor cylinder, thus increasing wear.

It is, therefore, desirable to have compression with vapour initially dry saturated at 1 as shown in Fig. 3.1, or even slightly superheated if a reciprocating compressor is used. Such compression is known as *dry compression*. The state of the vapour at the end of compression will, therefore, have to be at 2, at pressure p_k which is the saturation pressure of the refrigerant corresponding to the condensing temperature t_k, instead of being at 2″, which would be the state point if the Carnot cycle were to be executed. It results in the discharge temperature t_2 being higher than the condensing temperature t_k. Consequently, the refrigerant leaves the compressor superheated. The increased work of the cycle due to the substitution of wet compression by dry compression appears as the area 2–2′ – 2″, generally known as *superheat horn*.

It must, however, be stated here that wet compression in some cases is indeed desirable, and also practicable with the use of a continuous flow machine like a centrifugal or a screw compressor with no valves in place of a reciprocating compressor. The improvement in COP with wet compression is amply illustrated in the case of ammonia in Example 3.4 in which it is shown that the power consumption per ton refrigeration with wet compression is less by 10 per cent as compared with that of dry compression.

3.1.2 Throttling Versus Isentropic Expansion

Refrigerating machines are usually much smaller devices compared to power plants. Thus the net work required by refrigeration systems is quite small compared to the work done in power-generating plants.

Further, the positive work of the cycle, recovered during the isentropic expansion process, as shown by area 3-a-b-4′ in Fig. 3.2, is even smaller, as compared to the negative work of the cycle consumed during the isentropic compression process, shown by area 1-2-a-b. This is evident from the expression for work, viz., $-\int v\,dp$. Thus for the same pressure difference dp, the work depends on the volume v of the working substance. In the expander, the refrigerant is in the liquid state, whereas, in the compressor, it is in the gaseous state. The volume of the vapour is very large compared to the volume of the liquid ($v_g >> v_f$). Hence, the positive work of isentropic expansion is seldom large enough to justify the cost of an expander. On the other hand, the thermodynamic and friction losses of an expander, if employed, may even exceed the gain in work. Moreover, there are practical difficulties in smoothly expanding a liquid of a highly wet vapour in an expander.

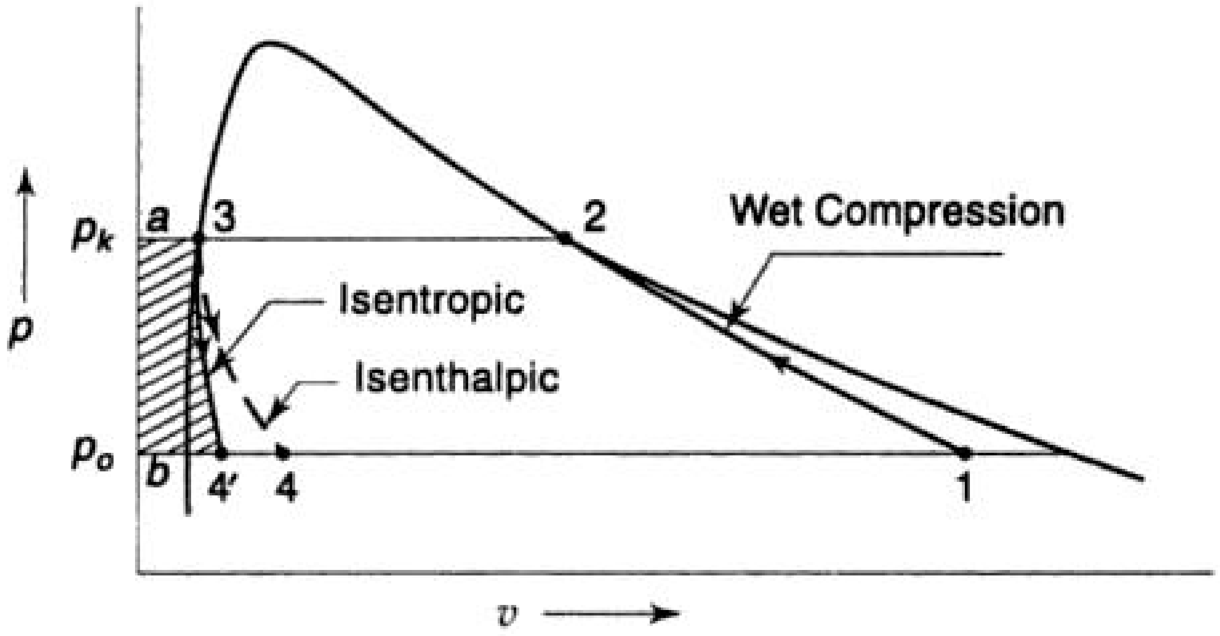

Fig. 3.2 Comparison of Throttling and Isentropic Expansion of Liquid on *p-v* Diagram

Accordingly, the isentropic expansion process of the Carnot cycle may be replaced by a simple throttling process or an isenthalpic process by the use of an expansion device such as a throttle valve or a capillary tube.

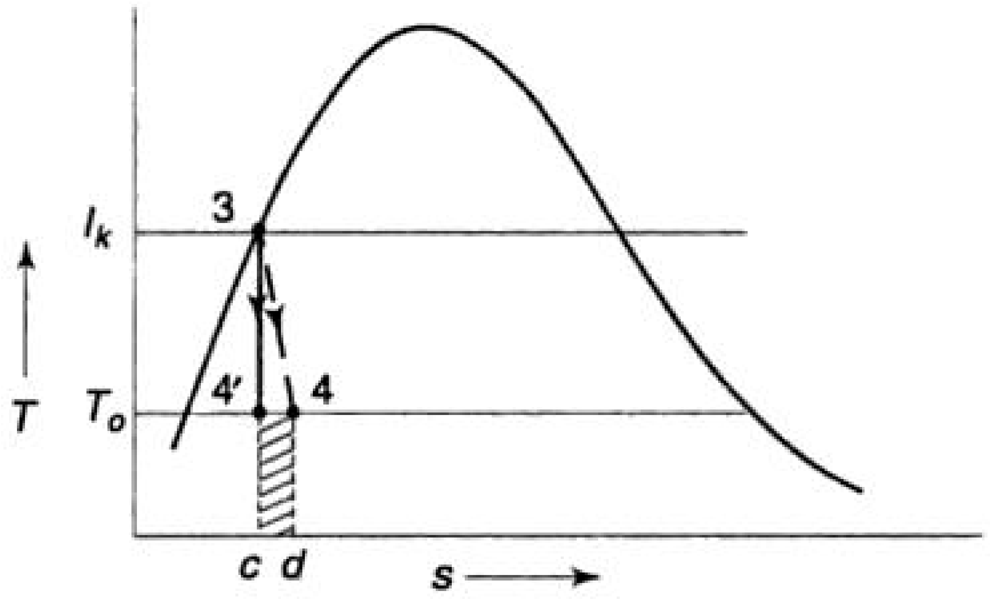

Fig. 3.3 Comparison of Throttling and Isentropic Expansion of Liquid on *T-s* Diagram

The process is an irreversible one and is accompanied by increase of entropy as shown by line 3-4 on the *T-s* diagram in Fig. 3.3. Thus, the substitution of the isentropic-expansion process 3-4′ by the isenthalpic/throttling process 3-4 would, theoretically, result in a loss of work represented by area 3-a-b-4′ on the *p-v* diagram and a decrease in the refrigerating effect represented by area 4′-c-d-4 on the *T-s* diagram. It can be shown that both these areas are equal.

3.2 VAPOUR COMPRESSION CYCLE

The cycle with the above two modifications is named as the *vapour compression cycle* and because of its high index of performance or efficiency, it is most widely used in commercial refrigeration systems. A complete vapour compression cycle is shown on the T-s diagram in Fig. 3.4 and on the p-v diagram in Fig. 3.5. Figure 3.4 also presents a comparison of the vapour compression cycle 1-2-3-4 with the reversed Carnot cycle 1-2″-3-4′, or 1-2″-3′-4, both operating between the same temperature limits of T_k and T_o.

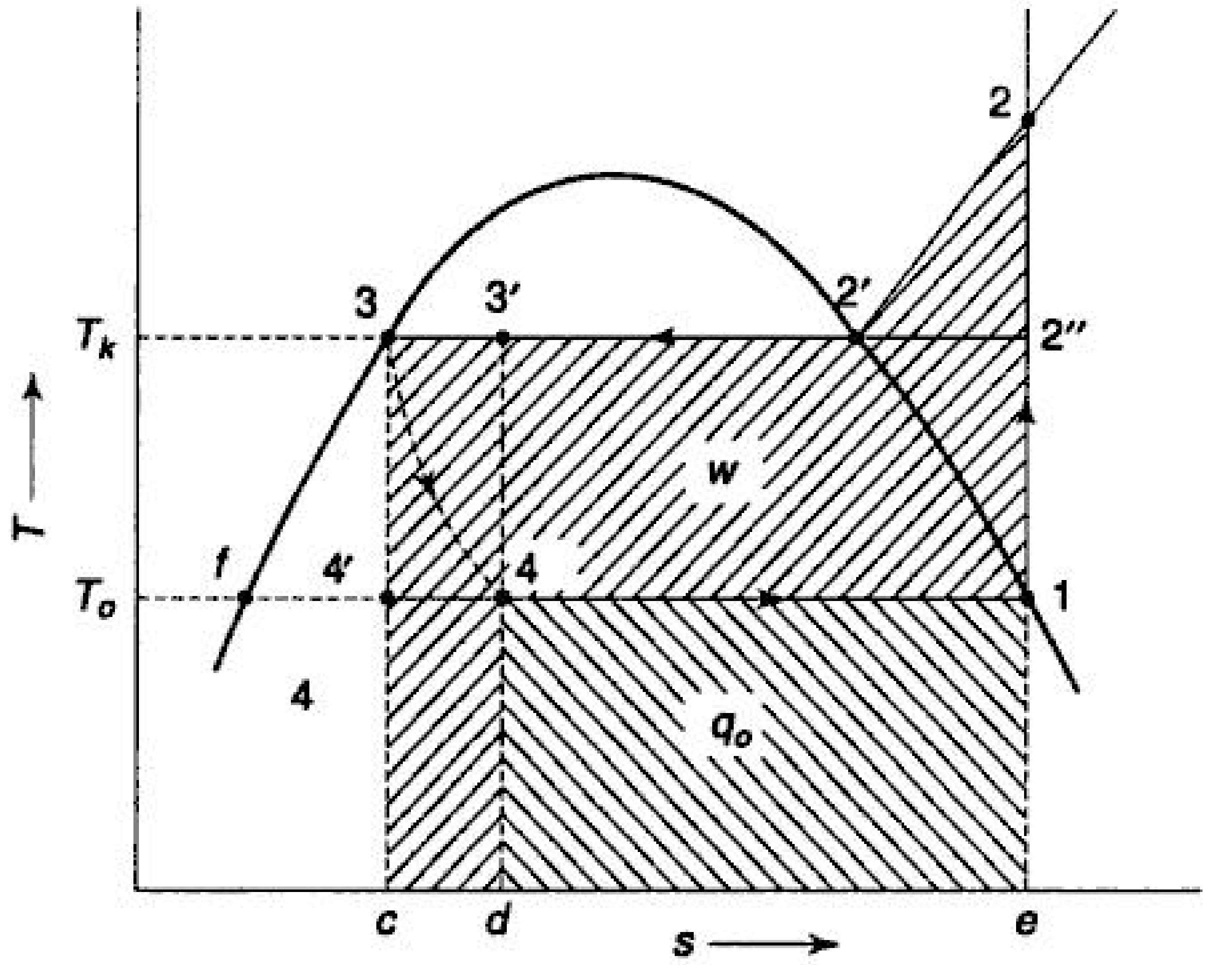

Fig. 3.4 Vapour Compression Cycle on T-s Diagram

In the vapour compression cycle:

Refrigerating effect, q_o = area 1-4-d-e

Heat rejected, q_k = area 2-2′-3-c-e

Work done, $w = q_k - q_o$

= area 1-2-2′-3-c-d-4-1

It may be seen that the vapour compression cycle presents three deviations from the reversed Carnot cycle, as indicated below:

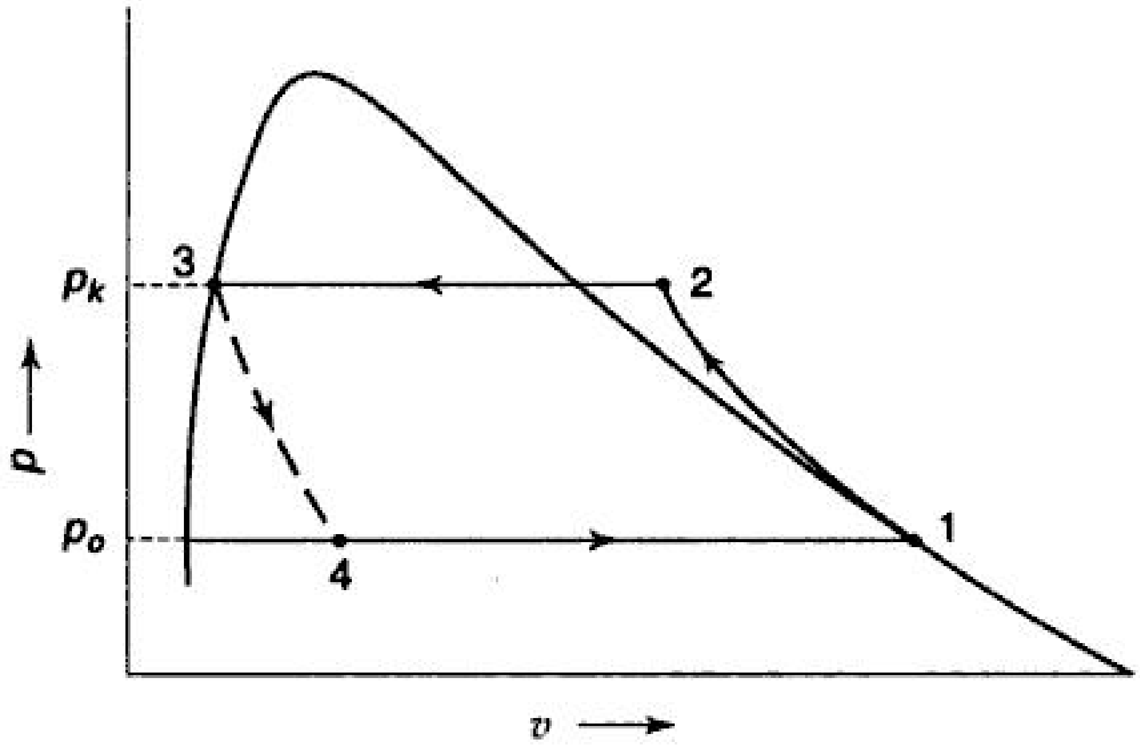

Fig. 3.5 Vapour Compression Cycle on p-v Diagram

cross-section of the expansion device causing *moisture choking* by freezing. The thermodynamic processes are as follows:

Process 1-2 Isentropic compression: $s_2 = s_1$. $Q = 0$

Work done, $$w = -\int v\mathrm{d}p = -\int \mathrm{d}h = -(h_2 - h_1) \tag{3.1}$$

Process 2-3 Desuperheating and condensation: p_k = const.

Heat rejected, $$q_k = h_2 - h_3 \tag{3.2}$$

Process 3-4 Isenthalpic expansion: $h_3 = h_4 = h_{f_4} + x_4 (h_1 - h_{f_4})$

$$\Rightarrow \qquad x = \frac{h_3 - h_{f_4}}{h_1 - h_{f_4}} \tag{3.3}$$

Process 4-1 Evaporation: p_o = const.

Refrigerating effect, $$q_o = h_1 - h_4 \tag{3.4}$$

3.3.1 Representation of Vapour Compression Cycle on Pressure-Enthalpy Diagram

It is worth noting that two of the processes are at constant pressure and one is at constant enthalpy. It is, therefore, found convenient to represent the vapour compression cycle on a pressure-enthalpy (*p-h*) diagram as shown in Fig. 3.7. Therefore, even though the fourth process is an isentropic one, the *p-h* diagram is still found convenient as the work done is given by the increase in enthalpy.

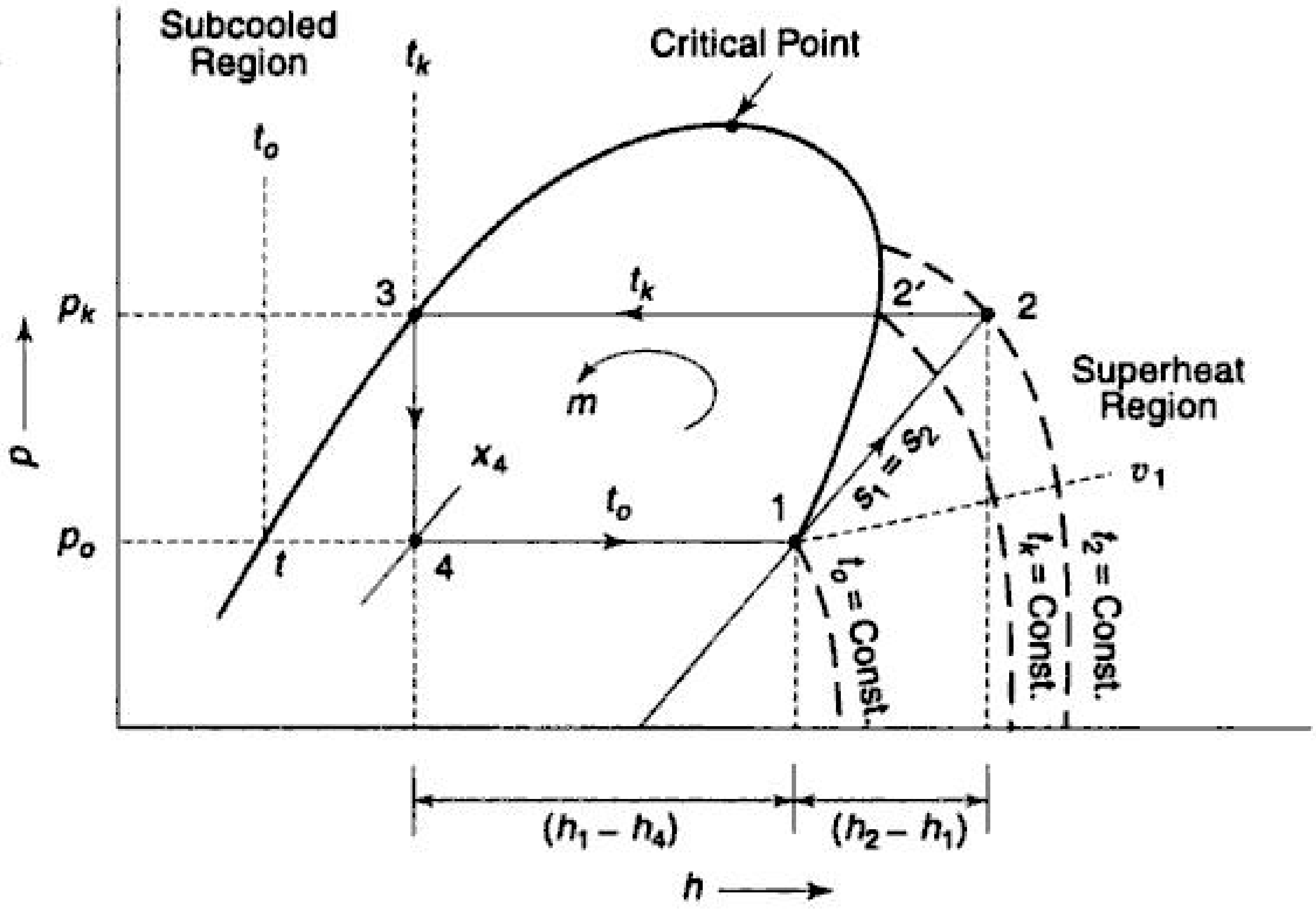

Fig. 3.7 Vapour Compression Cycle on *p-h* Diagram

The cycle described and shown in Figs 3.4, 3.5 and 3.7 is a *simple saturation cycle* implying that both the states, of liquid after condensation and vapour after evaporation, are saturated and lie on the saturated liquid and saturated vapour curves respectively. The condensation temperature t_k and evaporator temperature t_o, corresponding to the respective saturation pressures p_k and p_o, are also called *saturated discharge temperature* and *saturated suction temperature* respectively. However, the actual discharge temperature from the compressor is t_2.

Figure 3.7 also shows constant temperature lines in the subcooled and superheat regions along with constant volume lines. It may be noted that constant temperature lines in the subcooled liquid and low pressure vapour regions are vertical as the enthalpy of the liquid and the ideal gas are functions of temperature only and do not depend on pressure.

Further calculations of the cycle can be done as follows:

Heat rejected, $q_k = q_o + w = h_2 - h_3$

COP for cooling, $$\mathcal{E}_c = \frac{h_1 - h_4}{h_2 - h_1} \tag{3.5}$$

COP for heating, $$\mathcal{E}_h = \frac{h_2 - h_3}{h_2 - h_1} \tag{3.6}$$

Refrigerant circulation rate, $$\dot{m} = \frac{\text{refrigerating capacity}}{\text{refrigerating effect per unit mass}} = \frac{\dot{Q}_o}{q_o} \tag{3.7}$$

Specific volume of the vapour at suction = v_1

Theoretical piston displacement of the compressor or volume of the suction vapour,

$$\dot{V} = \dot{m} v_1 \tag{3.8}$$

Actual piston displacement of the compressor,

$$\dot{V}_p = \frac{\dot{m} v_1}{\eta_v}$$

where η_v is the volumetric efficiency.

Power consumption, $$\dot{W} = \dot{m} w = \dot{m}\,(h_2 - h_1) \tag{3.9}$$

Heat rejected in the condenser, $$\dot{Q}_k = \dot{m} q_k = \dot{m}\,(h_2 - h_3) \tag{3.10}$$

Expressing the power consumption per ton of refrigeration as *unit power consumption*, denoted by $\overset{*}{W}$, we have for mass flow rate and power consumption per ton refrigeration,

$$\overset{*}{m} = \frac{3.5167}{q_o} \text{ kg/(s)} \cdot \text{(TR)} \tag{3.11}$$

$$\overset{*}{W} = \overset{*}{m} w = 3.5167 \left(\frac{h_2 - h_1}{h_1 - h_4} \right) \text{ kW/TR} \tag{3.12}$$

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(f) the theoretical and actual COP.

Solution

(a) Refrigerating capacity

$$\dot{Q}_0 = \frac{10 \times 1000}{24} [4.1868 (30 - 0) + 335 + 1.94 (0.5)]$$
$$= 195{,}960 \text{ kJ/h}$$
$$= \frac{195{,}960}{3600} = 54.43 \text{ kW}$$

Refer to Fig. 3.8.

(b) From the table for ammonia

$h_1 = h_{g\,(-15°C)} = 1443.9$ kJ/kg and $s_1 = s_{g(-15°C)} = 5.8223$ kJ/(kg) (K)
$h_4 = h_3 = h_{f\,(35°C)} = 366.1$ kJ/kg
$q_0 = h_1 - h_4 = 1443.9 - 366.1 = 1077.8$ kJ/kg

Mass flow rate

$$\dot{m} = \frac{195{,}960}{1077.8} = 181.7 \text{ kg/h}$$

(c) From the table, entropies and enthalpies of vapour saturated at 35°C and superheated by 50 K and 100 K are, respectively,

$s_{g\,(35°C)} = 5.2086,\quad s_{(50\text{ K})} = 5.6466\quad s_{(100\text{K})} = 5.9806$

$h_{g\,(35°C)} = 1488.6\quad h_{(50\text{ K})} = 1633.6\quad h_{(100\text{K})} = 1703$

Now $s_2 = s_1 = 5.8223 > 5.6466$

Hence the vapour is superheated by more than 50 K.

Interpolating for discharge temperature

$$t_2 = 35 + 50 + \frac{5.8223 - 5.6466}{5.9806 - 5.6466} (50)$$
$$= 35 + 50 + 27.1 = 112.1°\text{C}$$

(d) Piston displacement of compressor

$$\frac{\pi D^2}{4} LN = \dot{V}_p = \frac{\dot{m} v_1}{\eta_v}$$

$$\Rightarrow \qquad \frac{\pi}{4} D^2 (1.2)\, D\, (1200)\, (60) = \frac{181.7\,(0.509)}{0.65}$$

whence $D = 0.128$ m

$L = 0.154$ m

(e) Enthalpy at discharge, by interpolation as for entropy

$$h_2 = 1633.6 + \frac{27.1}{50}(1761.6 - 1633.6) = 1633.6 + 69.4 = 1703 \text{ kJ/kg}$$

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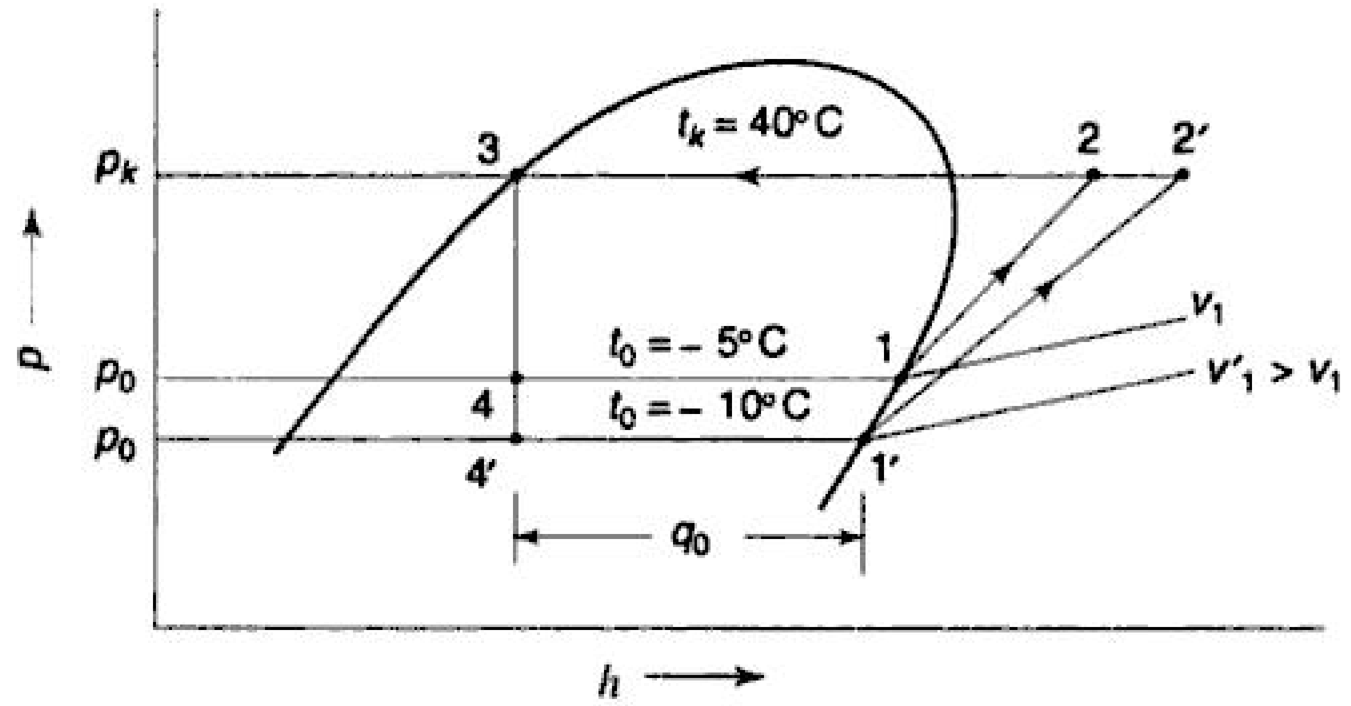

Fig. 3.12 Effect of Evaporator Pressure

$$\left(1 - \frac{V^*}{V^{*\prime}}\right) \times 100 = 16.8\%$$

and increases the power consumption per unit refrigeration by

$$\left(\frac{W^{*\prime}}{W^*} - 1\right) \times 100 = 9.7\%$$

It is observed that a decrease in evaporator temperature results in:

(i) Decrease in refrigerating effect from $(h_1 - h_4)$ to $(h'_1 - h'_4)$

(ii) Increase in the specific volume of suction vapour from v_1 to v'_1

(iii) Decrease in volumetric efficiency, due to increase in the pressure ratio, from η_v to η'_v.

(iv) Increase in compressor work from $(h_2 - h_1)$ to $(h'_2 - h'_1)$ due to increase in the pressure ratio as well as change from *steeper* isentropic 1-2 to *flatter* isentropic 1′ – 2′ as discussed in Sec. 3.5.3.

Since

$$\dot{Q}_0 = \dot{m} q_0 = \frac{\eta_v \dot{V}_p}{v_1} q_0$$

and $$\dot{W}^* = m^* w \tag{3.23}$$

expressions for the dependence of capacity and unit power consumption may now be written as follows:

$$\begin{aligned}\dot{Q}_0 &\propto q_0 = (h_1 - h_4)\\ &\propto \frac{1}{v_1}\\ &\propto \eta_v\end{aligned}$$

and $$\begin{aligned}W^* &\propto m^* \propto \frac{1}{q_0} = \frac{1}{h_1 - h_4}\\ &\propto w = h_2 - h_1\end{aligned}$$

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is a function of initial temperature T_1, pressure ratio p_2/p_1 and exponent γ. Thus, we see that it increases with the initial temperature T_1. That is why isentropic lines on the *p-h* diagram become flatter at higher temperatures as they move away more and more from the saturated vapour curve.

An increase in specific volume decreases the capacity. On the contrary, an increase in refrigerating effect will increase the capacity. The net effect of superheating is to theoretically reduce the capacity in ammonia systems and to increase it in Freon 12 systems. The ratio of the capacities with superheating Q'_0 and without superheating Q_0, can be written as

$$\frac{Q'_0}{Q_0} = \frac{h'_1 - h_4}{h_1 - h_4} \cdot \frac{v_1}{v'_1} \tag{3.28}$$

Similarly, for the work done per unit of refrigeration, it may be seen that there are two contradictory influences, viz., an increase in the refrigerating effect decreasing the mass flow requirement and hence work and an increase in the specific work itself due to an increase in the suction temperature. The resulting work per unit refrigeration may, therefore, increase or decrease depending on the refrigerant and operating temperatures.

The effect on work or power consumption per unit refrigeration is given by the ratio given in Eq. (3.29).

$$\frac{W^{*\prime}}{W^*} = \frac{h_1 - h_4}{h'_1 - h_4} \cdot \frac{h'_2 - h'_1}{h_2 - h_1} \tag{3.29}$$

The COP of the cycle with superheat is given by

$$\begin{aligned} \mathcal{E}'_c &= \frac{h'_1 - h_4}{h'_2 - h'_1} \\ &= \frac{(h_1 - h_4) + (h'_1 - h_1)}{(h_2 - h_1) + [(h'_2 - h'_1) - (h_2 - h_1)]} \end{aligned} \tag{3.30}$$

As both the numerator and the denominator increase, the numerical value of COP may increase or decrease or remain the same. It has been shown that in Freon 12 systems, superheating increases the COP whereas in Freon 22 and ammonia systems, it decreases it. In general, however, the effect of slight superheat on the volumetric efficiency of the reciprocating compressor and the COP is beneficial as it ensures complete vaporization of liquid refrigerant droplets in suspension in the suction vapour. In Freon 12 systems, a suction vapour temperature of 15 to 25°C is always beneficial, irrespective of the evaporator temperature.

It may be noted that superheating outside the evaporator or cold space results in a loss.

3.5.4 Effect of Liquid Subcooling

It is possible to reduce the temperature of the liquid refrigerant to within a few degrees of the temperature of the water entering the condenser in some condenser designs by installing a *subcooler* between the condenser and the expansion valve. The effect of subcooling of the liquid from $t_3 = t_k$ to t'_3 is shown in Fig. 3.16. It

Hence $$h_2 = 201.5 + \frac{10.2}{20}(216.4 - 201.5) = 208.2 \text{ kJ/kg}$$

$$q_0 = 181 - 69.5 = 111.5 \text{ kJ/kg}$$

$$w = 208.2 - 181 = 27.2 \text{ kJ/kg}$$

$$\text{COP} = \frac{111.5}{27.2} = 4.09$$

$$\text{HP/TR} = \frac{4.761}{4.09} = 1.16$$

(b) *Liquid-vapour heat exchanger cycle*

Degree of superheat at suction

$$t'_1 - t_1 = 15 - (-15) = 30°\text{C}$$

By interpolation for superheated vapour

$$h'_1 = 193.2 + \frac{10}{20}(205.7 - 193.2) = 199.45 \text{ kJ/kg}$$

$$s'_1 = 0.751 + \frac{10}{20}(0.7942 - 0.751) - 0.7726 \text{ kJ/(kg)} \cdot \text{(K)} = s'_2$$

Now, $$h'_1 - h_1 = 199.45 - 181 = 18.45 = h_3 - h'_3$$

Hence, $$h'_3 = 69.5 - 18.45 = 51.05 \text{ kJ/kg}$$

By interpolation for the degree of superheat

$$\Delta t = 20 + \frac{0.7726 - 0.731}{0.7741 - 0.731}(20) = 20 + 19.3 = 39.3°\text{C}$$

Hence $$h'_2 = 216.4 + \frac{19.2}{20}(231 - 216.4) = 230.5 \text{ kJ/kg}$$

$$q'_0 = h_1 - h'_3 = 181 - 51.05 = 129.95 \text{ kJ/kg}$$

$$w = 230.4 - 199.45 = 30.95$$

$$\text{COP} = \frac{129.95}{30.95} = 4.199$$

Increase in $$\text{COP} = \frac{4.199 - 4.09}{4.09}(100) = 2.56\%$$

$$\text{HP/TR} = \frac{4.761}{4.199} = 1.134$$

Decrease in $$\text{HP/TR} = \frac{1.16 - 1.134}{1.2}(100) = 2.5\%$$

Note ✍: *Theoretically, the increase in COP is not very large for Freon 12. And for Freon 22 there is, in fact, a decrease in COP. However, superheat improves the performance by ensuring complete vaporization of liquid. In refrigerators and air conditioners, the capillary tube is joined to the suction line, thus forming a regenerative heat exchanger.*

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Point 1d: $t = -28°C$ (saturation at 1.324 bar) $p = 1.324$ bar

Point 1c: $t = -18°C$ $p = 1.324$ bar $h = 1451.3$ kJ/kg

Point 1b: $t = -8°C$ $p = 1.324$ bar $h = 1473.6$

Point 1: $t = -8°C$ $h = 1473.6$ kJ/kg ($= h_{1b}$)

Points 2, 2a and 2b: $t = 140°C$ $p = 13.5$ bar $h = 1777$ kJ/kg

Point 2c: $t = 130°C$ $p = 13.5$ bar $h = 1751$ kJ/kg

Point 3: $t = 35°C$ (saturation at 13.5 bar) $p = 13.5$ bar

Point 3a: $t = 30°C$ $p = 13.5$ bar $h = 341.8$ kJ/kg

(a) Refrigerating capacity (from brine)

$$\dot{Q}_0 = 102 \times 3.14 \times 7 \times 1.02 = 2287 \text{ kJ/min}$$

$$= \frac{2287}{211} = 10.84 \text{ TR}$$

(b) Refrigerating effect

$$q_0 = h_{1c} - h_4 = 1451.3 - 351.5 = 1099.8 \text{ kJ/kg}$$

Ammonia circulated

$$\dot{m} = \frac{2287}{1099.9} = 2.08 \text{ kg/min}$$

(c) Note: From 2 to 2b, it is an isenthalpic process. It has been assumed here that enthalpy is a function of temperature only and hence $t_{2b} = t_2$. Enthalpy increase during compression $= h_2 - h_1$

$$= 1777 - 1473.6 = 303.4 \text{ kJ/kg}$$

Total enthalpy increase of ammonia $= H_2 - H_1$

$$= \dot{m}\,(h_2 - h_1) = 2.08\,(303.4) = 631.1 \text{ kJ/min}$$

Heat to jacket water, $\dot{Q}_j = 5 \times 4.1868 \times 8.9 = 186.3$ kJ/min

Compressor work, $|\dot{W}| = (H_2 - H_1) + |\dot{Q}_j|$

$$= 631.1 + 186.3 = 817.4 \text{ kJ/min}$$

$$= 13.62 \text{ kW}$$

$$\text{Compressor IHP} = \frac{13.62 \times 10^3}{746} = 18.26$$

Compressor input = Power consumption of motor × motor efficiency

$$= 18.8 \times 0.92 = 17.3 \text{ kW}$$

$$\text{Compressor BHP} = \frac{17.3 \times 10^3}{746} = 23.19$$

Mechanical efficiency, $\eta_m = \dfrac{\text{IHP}}{\text{BHP}} = 0.787$

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(iv) Condenser heat rejected

$$\dot{Q}_k = \dot{m}\, q_k = \dot{m}\, (h_{3n} - h_6) = 0.817 \times 10^{-3}\ (234 - 77.65)$$

$$= 0.128 \text{ kW } (128 \text{ W})$$

(v) Suction vapour volume

$$V_s = \frac{60\, \dot{m}}{N}\, v_{10} = \frac{0.817 \times 10^{-3}\,(0.18)}{2800/60}$$

$$= 3.151 \times 10^{-6} \text{ m}^3\text{/rev} = 3.151 \text{ cc/rev}$$

Volumetric efficiency of compressor

$$\eta_v = \frac{V_s}{V_p} = \frac{3.51}{4.33} = 0.732\ (73.2\%)$$

Note ✍: *Results for alternatives to CFC 12 in refrigerators are discussed in Chap. 4.*

3.7.1 Pull-Down Characteristic and Ice-Making Time of Refrigerators[3]

According to the Indian Standards Institution (ISI) specification, for the testing of refrigerators, the environment temperature is maintained at 43°C. The *no-load test* is performed by adjusting the thermostat position corresponding to an average cabinet temperature of 7°C. The purpose of this test is to find the *pull-down period*, the no-load power consumption, and the percentage running time as per ISI code. The pull-down period is the time required to reach the specified temperatures inside the cabinent after switching on the unit. The temperature at the geometric centre of the evaporator shall not exceed –5°C at pull-down.

Another test performed on the refrigerators is the *ice-making time* test. For this test, after stable operating conditions are obtained following pull-down period, a quantity of 0.5 kg of water at 30 ± 1°C in two standard ice trays is kept in the freezer, and the time for each 'ON' and 'OFF' is noted. After specified time of 3 hours, the ice trays are examined for the formation of ice, and the *running time* of the unit during the 3 hours of operation is obtained, say, from the data acquisition system monitoring 'ON-OFF' cycle. This running time of the unit is the ice-making time. If ice is not formed during 3 hours of operation, the system operation is to be continued.

For the case of Freon 12 refrigerator of Example 3.8, the pull down period is 90 minutes, and the ice making time is 125 minutes.

Purpose of these tests is to compare the values for different refrigerators, and for different refrigerants. Increase in pull-down period, running time, and ice making time implies inadequate capacity.

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PROBLEMS

3.1 A 15 TR Freon 22 vapour compression system operates between a condenser temperature of 40°C and an evaporator temperature of 5°C.

(a) Determine the compressor discharge temperature:

(i) Using the *p-h* diagram for Freon 22.

(ii) Using saturation properties of Freon 22 and assuming the specific heat of its vapour as 0.8 kJ/kg. K.

(iii) Using superheat tables for Freon 22.

(b) Calculate the theoretical piston displacement and power consumption of the compressor per ton of refrigeration.

3.2 A simple saturation ammonia compression system has a high pressure of 1.35 MN/m^2 and a low pressure of 0.19 MN/m^2. Find per 400,000 kJ/h of refrigerating capacity, the power consumption of the compressor and COP of the cycle.

3.3 (a) A Freon 22 refrigerating machine operates between a condenser temperature of 40°C and an evaporator temperature of 5°C. Calculate the increase (per cent) in the theoretical piston displacement and the power consumption of the cycle:

(i) If the evaporator temperature is reduced to 0°C.

(ii) If the condenser temperature is increased to 45°C.

(b) Why is the performance of a vapour compression machine more sensitive to change in evaporator temperature than to an equal change in the condenser temperature?

3.4 In a vapour compression cycle saturated liquid Refrigerant 22 leaving the condenser at 40°C is required to expand to the evaporator temperature of 0°C in a cold storage plant.

(a) Determine the percentage saving in net work of the cycle per kg of the refrigerant if an isentropic expander could be used to expand the refrigerant in place of the throttling device.

(b) Also determine the percentage increase in refrigerating effect per kg of refrigerant as a result of use of the expander. Assume that compression is isentropic from saturated vapour state at 0°C to the condenser pressure.

3.5 An ammonia refrigeration system operates between saturated suction temperature of –20°C, and saturated discharge temperature of +40°C. Compare the COP of the cycle using wet compression with that of the cycle using dry compression.

Assume that the vapour leaving the compressor is saturated in the case of wet compression, and the vapour entering the compressor is saturated in the case of dry compression. The refrigerant leaves the condenser as saturated liquid.

3.6 A standard vapour compression cycle using Freon 22 operates on simple saturation cycle at the following conditions:

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Chapter 4

Refrigerants

4.1 A SURVEY OF REFRIGERANTS

The first refrigerant used was ether, employed by Perkins in his hand operated vapour-compression machine. In the earlier days, ethyl chloride (C_2H_5Cl) was used as a refrigerant which soon gave way to ammonia as early as in 1875. At about the same time, sulphur dioxide (SO_2) in 1874, methyl chloride (CH_3Cl) in 1878 and carbon dioxide (CO_2) in 1881, found application as refrigerants. During 1910-30 many new refrigerants, such as N_2O_3, CH_4, C_2H_6, C_2H_4, C_3H_8, were employed for medium and low-temperature refrigeration. Hydrocarbons were, however, found extremely inflammable. Dichloromethane (CH_2Cl_2), dichloroethylene ($C_2H_2Cl_2$) and monobromomethane (CH_3Br) were also used as refrigerants for centrifugal machines.

A great breakthrough occurred in the field of refrigeration with the development of Freons (a trade name) in 1930s by E.I. du Pont de Nemours and Co.[32] Freons are a series of fluorinated hydrocarbons, generally known as *fluorocarbons*, derived from methane, ethane, etc., as bases.

With fluorine, chlorine and sometimes bromine in their molecule, these form a series of refrigerants with a wide range of *normal boiling points* (boiling points or saturation temperatures at one atmosphere pressure) to satisfy the varied requirements of different refrigerating machines. The presence of fluorine in the molecule makes the compound non-toxic and imparts other desirable physical and physiological characteristics.

Plank[31] has given individual treatment to some 50 inorganic and organic refrigerants and many more have been listed. Among the most common inorganic refrigerants are:

Ammonia (NH_3)	Used with reciprocating and screw compressors, in cold storages, ice plants, food refrigeration, etc.

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mal ambient temperatures. Because of these reasons, these substances are not used in common refrigeration and air conditioning applications.

To the left of R 12, in the row with one H atom, we have another well-known refrigerant HCFC 22 or R 22, viz., $CHClF_2$. It has an N.B.P. of – 40.8°C which is about 10°C lower than that of R 12. It is, therefore, a comparatively higher pressure refrigerant. Consequently, it has a smaller specific volume of the suction vapour and gives about 40 per cent more capacity in a positive displacement compressor as compared to R 12. Previously, R 22 was employed for air conditioning in large capacity plants and package units above 5 TR only. But now, because of the capacity bonus, it is used even in one ton window-type air conditioners. However, its value of γ is higher than that of R 12. Therefore, it has higher compressor discharge and winding temperatures. It is, therefore, necessary to affect greater cooling of motor windings in R 22 hermetically sealed units with the help of incoming suction vapours, and also by adequate heat transfer to surrounding air, even by forced convection, from its body. In any case, R 22 units have more frequent burn-out problems than R 12. Warrantees are given only for 1-2 years unlike 5-10 years for R 12. The problem of burn-out would be more acute if R22 were to be used in domestic refrigerators. In these, because of lower $t_0 = -25°C$ and hence, lower suction pressure, the pressure ratio, and discharge and winding temperatures would be even higher. R 22, therefore, is not found suitable for use in domestic refrigerators.

Further, in domestic refrigerators, because of the very small capacities, the compressor sizes (bore and stroke) are already so small with R 12, e.g., 1.7 cm × 1.8 cm in a 165 L refrigerator, that no advantage would be gained by reducing their sizes further with the use of R 22. On the other hand, it would introduce more complications due to the need to maintain requisite small tolerances and accuracy of dimensions in the manufacture of such tiny sized cylinders. Because of its lower boiling point, R 22 also finds application in food freezing, freeze drying, etc., maintaining positive pressure in evaporators with temperatures as low as – 40°C.

Above R 22, in the 1-H column, we have R 21. It is, however, a little chemically unstable. Further above is R 20 which has a very high boiling point substance unsuitable as a refrigerant. Below R 22, we have R 23 with an N.B.P. of – 82.2°C. It is similar to R 13.

Other substances in Table 4.1, are either chemically unstable or highly inflammable due to the pressence of H atoms in the molecule. Methane is the most inflammable of all. R 40 or methyl chloride (CH_3Cl) with an N.B.P. of –23.74°C, used to be a very popular refrigerant a few decades ago. It is, however, poisonous and is odourless at the same time. Its leakage, therefore, gives no warning and may lead to death. Its use is, therefore, banned inspite of its favourable thermodynamic characteristics.

***Note* ✍:** *Here, we notice that substances with normal boiling points in the range – 50 to + 50°C only find application as refrigerants in commercial refrigeration and air conditioning.*

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Table 4.5 gives the suction volume requirements in m^3 per hour per ton of refrigeration of some refrigerants along with their normal boiling points. Calculations have been done for 40°C condensing and – 15°C evaporating temperatures except for water for which the evaporating temperature is 5°C, and carbon dioxide for which the condensing temperature is 25°C.

Table 4.5 Suction Vapour Volumes of Refrigerants for $t_k = 40°C$ and $t_0 = -15°C$

Refrigerant	t_s °C	q_0 kJ/kg	v m^3/kg	V^* $m^3/h/TR$
Water	100	2342.5(t_0 = 5°C)	147.2	825.6
R 113	45.9	111.8	1.649	186.9
R 11	23.7	148.3	0.772	65.9
R 114	3.6	88.6	0.263	37.6
R 600a (Isobutane)	– 11.9	238.0	0.3993	21.24
R 152a	– 24.7	226.5	0.207	11.572
R 134a	– 26.2	139.8	0.12	10.867
R 12	– 29.8	108.4	0.093	10.857
Ammonia	– 33.4	1053.4	0.509	6.124
R 22	– 40.8	108.4	0.0775	6.668
R 290 (Propane)	– 42.1	252.4	0.15425	7.737
Carbon Dioxide	– 78.52	158.7 (t_k = 25°C)	0.0166	1.33
R 13	– 81.5	49.1	0.012	3.1

It is seen from Table 4.5 that the required volume of the suction vapour increases with increasing normal boiling point. Inspite of the large latent heat of vaporization (and hence large refrigerating effect q_0) for ammonia, because of its large specific volume, its suction volume requirement is nearly the same as that of R 22. Both have nearly equal N.B.Ps. Thus, the normal boiling point is once again established as the single factor which determines the refrigerating capacity per unit volume of the suction vapour.

One can, therefore, conclude that high-boiling refrigerants, such as water, R 113 and R 11 are low-pressure refrigerants and require large suction volumes. They are suited for use with centrifugal compressors in large-capacity refrigerating machines. Conversely, low-boiling refrigerants, such as ammonia, R 22, propane, carbon dioxide, etc., which are high-pressure refrigerants require small suction volumes and are suited for use with reciprocating compressors in medium and smaller capacity refrigerating machines. In the middle, we have isobutane, R 152a, R134a and R12 mostly used with reciprocating compressors in very small capacity machines such as domestic refrigerators, car air conditioners, water coolers, etc. Note that R 22 has also been used with centrifugal compressors in very large capacity air conditioning applications when suction volume requirement becomes large enough to suit a centrifugal compressor.

R 114 with N.B.P. of 3.6°C lies in the middle range. It has medium pressures and suction volume requirements. It is found convenient for use with rotary compressors. In any case, it is a CFC. Hence it is to be phased out.

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not occur until –51°C. With 18 per cent oil, separation will occur even at 0.5°C. No matter how little oil goes into the evaporator, as the evaporation of refrigerant proceeds, the composition of oil in the liquid solution increases and it is bound to pass through the *critical composition* for separation which usually lies between 15 and 20 per cent of oil in the liquid phase.

Table 4.7 Solubility of Refrigerants and Mineral Oils

Completely Miscible	*Partially Miscible*			*Immiscible*
	High Miscibility	*Intermediate Miscibility*	*Low Miscibility*	
R 11	R 13 B1	R 22	R 13	NH_3
R 600a				
R 12	R 501	R 114	R 14	CO_2
R 290				
R 21			R 115	R 134a
R 113				
R 152a				
R 500			R 502	

The return of oil in these compressors, therefore, presents a problem. At low refrigeration temperatures, it is all the more acute. One solution of the problem is to install an efficient oil separator. Another is to use synthetic oils instead of mineral oils which are completely miscible with the refrigerant at temperatures as low as – 80°C. Among the synthetic oils, a polybutyl silicate and alkyl benzenes have better miscibility with R 22. Another solution is to use R 501 which is a mixture of 75 per cent R 22 and 25 per cent R 12. Refrigerant 12 in the mixture improves the miscibility. But, R 501 is being-phased out along with R 12. Hence, one has to find another additive to R 22 to improve its miscibility with oil. Also, in *direct expansion evaporators*, oil-refrigerant emulsion can be easily carried to the compressor by high velocity suction vapours. These evaporators are, therefore, preferred over flooded evaporators, and are particularly used for low temperature refrigeration and with refrigerants such as R 22, R 502, etc.

In *flooded evaporators*, the separated oil-rich layer, being lighter, floats on top of the boiling liquid, and because of the extremely low velocity of the suction vapour, it cannot be carried to the compressor. Thus, when flooded evaporators are used with R 22, R 502, etc., a connection must be provided for the overflow of oil from the evaporator to the compressor crank-case.

In the case of ammonia, however, the oil being heavier than the refrigerant, it collects at the bottom of the evaporator and can be drained out, if necessary. The same is the case with R 12 also.

4.7.5 Action with Materials of Construction

The choice of a particular refrigerant has also a bearing on the selection of the materials of construction. Ammonia, for example, attacks copper and copper-bearing

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For ammonia leaks, a burning sulphur taper is used which in the presence of ammonia forms white fumes of ammonium sulphite.

An *electronic leak detector* is most sensitive and is used in the manufacture and assembly of refrigeration equipment. The operation of the instrument depends on the flow of current due to the ionization of the leaking gas between two opposite-charged platinum electrodes.

4.8.7 Cost of Refrigerant

The consideration of the cost of a refrigerant has for many years commended the use of ammonia as a refrigerant in large industrial plants, such as cold storages and ice plants. The cost factor, however, is only relevant to the extent of the price of the initial charge of the refrigerant which is very small compared to the total cost of the plant and its installation. The cost of losses due to leakage is also important. In small-capacity units requiring only a small charge of the refrigerant, the cost of refrigerant is immaterial.

Ammonia is the cheapest refrigerant. R 12 is slightly cheaper than R 22. During the last few years, R 12 and R 22 have replaced ammonia in the dairy and frozen food industry. The fact that ammonia may attack some food products has been predominant cause for the change from ammonia to R 22 even in cold storages. But, because it is environment-friendly, the future may see the return of ammonia in a big way.

4.9 REFRIGERANT PIPING AND DESIGN

A well-designed refrigeration plant is one with a well-sized and well-laid refrigerant piping. Some salient features of this are discussed below. The material used for fluorocarbon refrigerant piping is either seamless copper tubing or iron, whereas for ammonia only iron pipes are used. Their sizes are given in terms of the outside diameter (OD) for copper tubing and normal iron pipe size (IPS) for iron pipes.

4.9.1 Location and Arrangement of Piping

The following factors are important:

(i) To minimize tubing and refrigerant requirement and pressure drops, refrigerant lines should be as short and direct as possible.

(ii) Piping is to be planned for a minimum number of joints using as few elbows and fittings as possible, although providing for sufficient flexibility to absorb compressor vibrations.

(iii) It is to be so arranged that normal inspection and servicing of the compressor and equipment is not hindered.

(iv) It should be so run that it does not interfere with the removal of compressor-cylinder heads, access plates or any other internal part such as the stator in hermetically-sealed units.

(v) Sufficient clearance is to be provided between the piping and wall or between pipes for insulation.

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In general, the pressure drop should not be greater than that corresponding to 0.9°C change in the saturation temperature for ammonia, and 0.9 to 1.8°C for fluorocarbons. A change of 0.9°C in the saturation temperature at 38°C condenser comes to a pressure drop of approximately 0.227 bar for ammonia, 0.124 bar for R 12, 0.2 bar for R 22 and 0.214 bar for R 502.

4.9.6 Suction Lines

Suction lines are the most critical from the viewpoints of design and construction. The considerations involved are the following:

(i) Correct size for practical pressure drop.

(ii) Capability to return oil to the compressor by entrainment by the suction vapour under minimum loading conditions, especially in the case of fluorocarbons.

(iii) If there are suction risers, gas velocities to ensure oil return have to be increased. Thus a minimum tonnage is prescribed depending on the size of the suction line for oil entrainment in suction risers as given in Table 4.9 for R 22.

Table 4.9 Tonnage of Suction Piping up Suction Risers

Refrigerant	t_0, °C	*Pipe OD*, cm			
		1.27	1.59	1.905	2.86
R-22	– 40	0.09	0.16	0.27	0.79
	–10	0.13	0.24	0.39	1.2
	5	0.18	0.33	0.54	1.6

(iv) Double-suction risers may be used for full-load operation and single risers for part-load operation as shown in Fig. 4.2. Thus, excessive pressure drop at full load is avoided, and oil return at part load is ensured. When the load reduces, the oil cannot be entrained in the beginning which collects in the U-bend and forms a seal. Afterwards, the gas flows only through one riser.

(v) Prevention of drainage of oil from an active evaporator into an idle evaporator. For this purpose, arrangements are made as shown in Figs 4.3 to 4.5. The common-suction line should either be horizontal or pitched down towards the compressor.

It is customary to design suction lines so that the total loss in pressure is equivalent to drops of about 1.8 and 0.9°C in saturation temperatures for fluorocarbons and ammonia respectively. It is also desirable to provide for less pressure drop in low-temperature installations because of increased penalties on the compressor size and performance. Since the pressure loss is also a function of the length of the line and number of fittings, it is obviously desirable to have the shortest possible runs of suction lines. Tables 4.10 and 4.11 give the carrying capacities in tons for R-22 and ammonia, respectively, at different pressure-drop rates.

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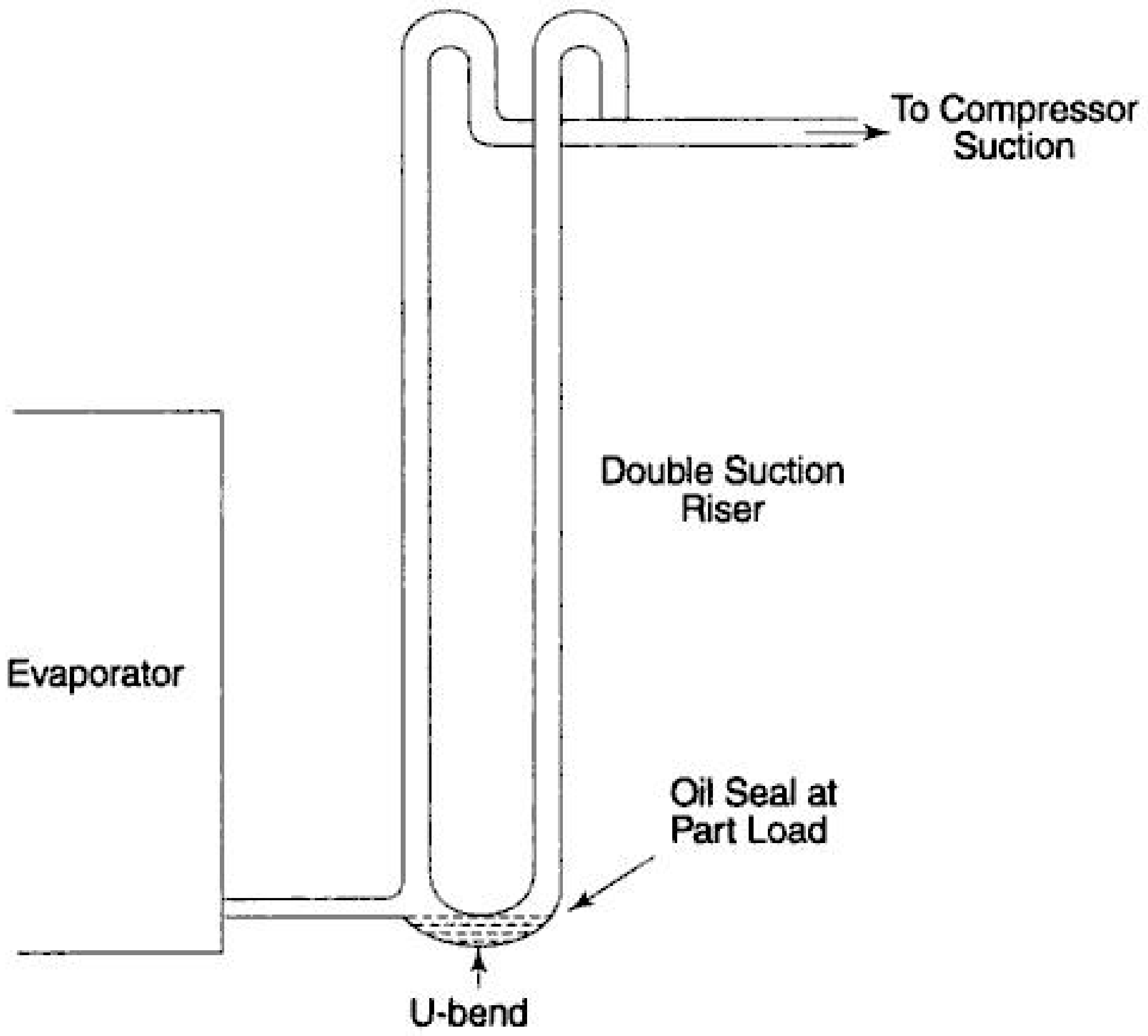

Fig. 4.2 Double-suction Riser

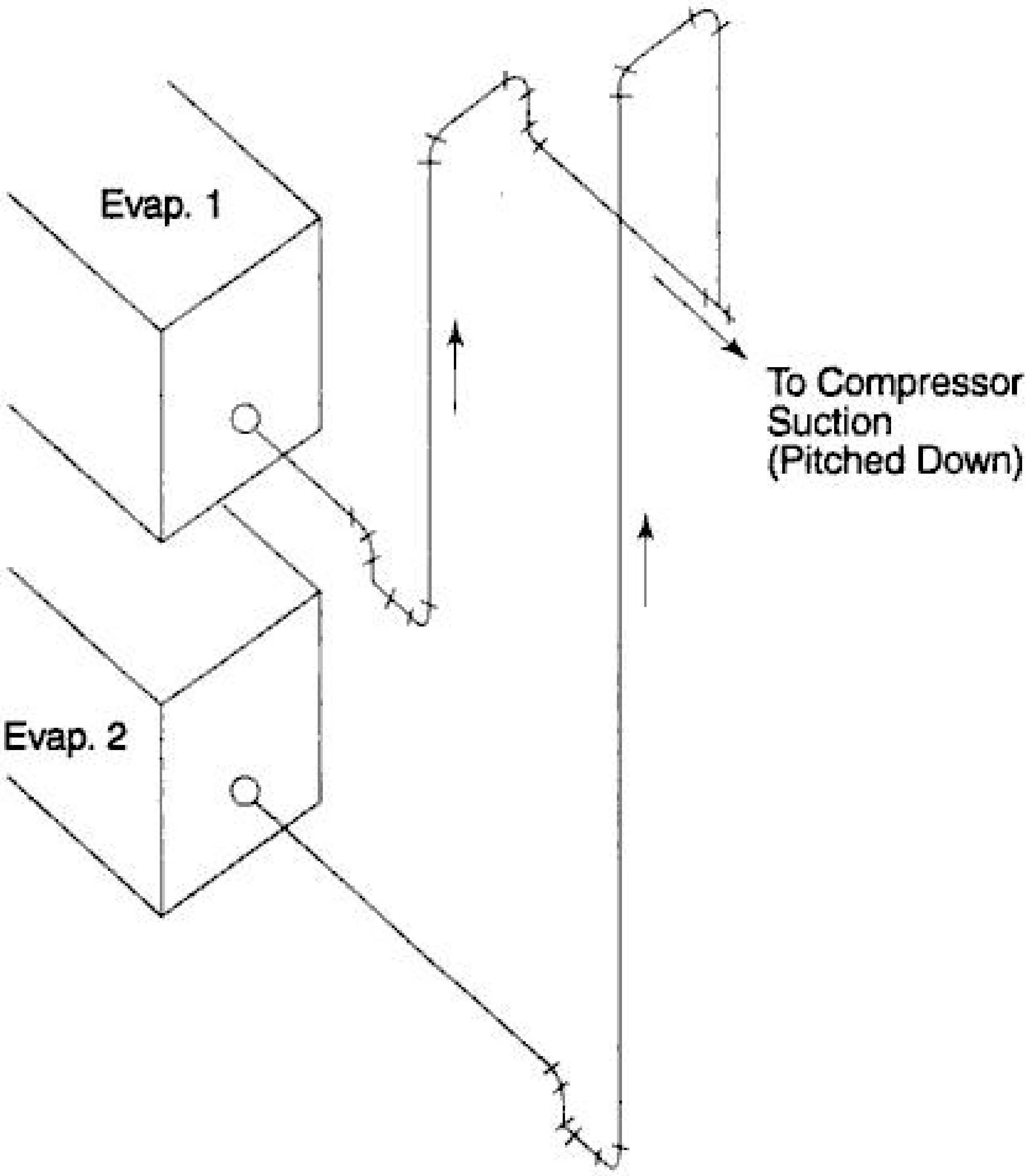

Fig. 4.3 Multiple Evaporators at Different Levels with Compressor Above

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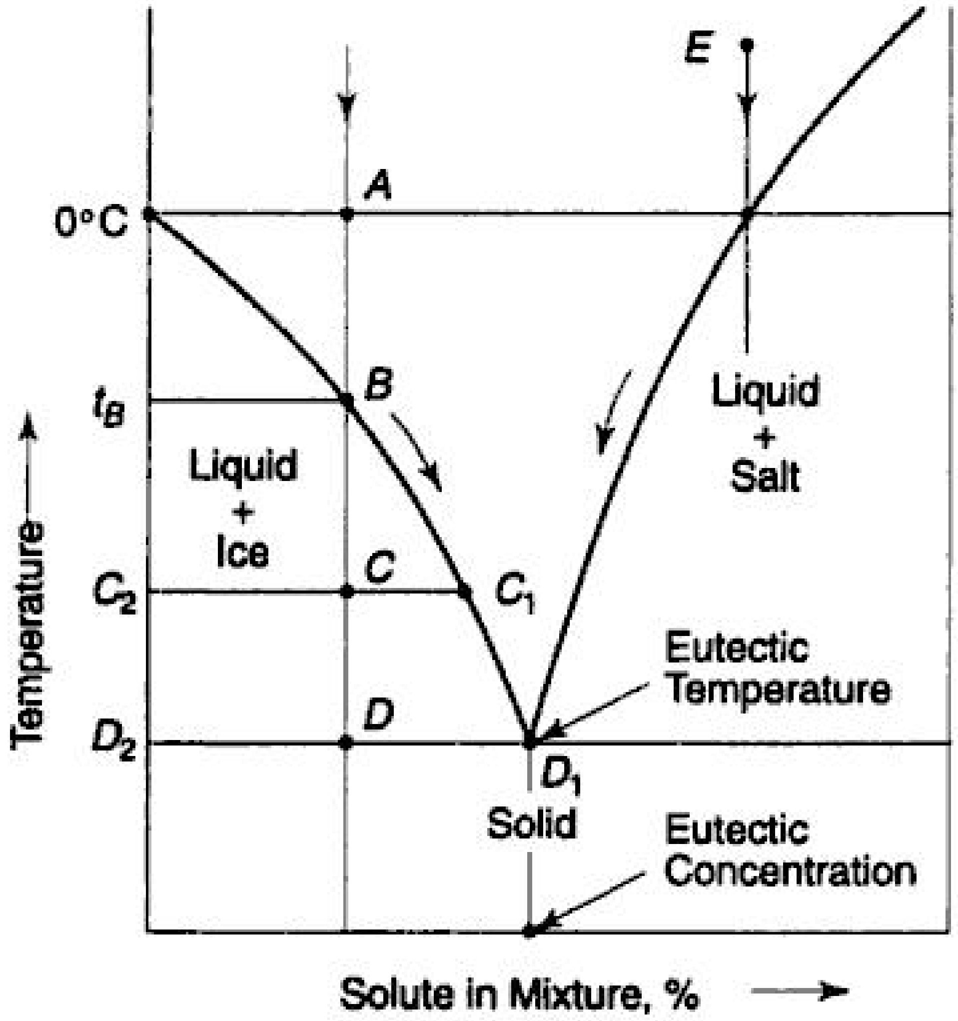

Fig. 4.7 Phase Diagram for Brine

If a brine solution has less than eutectic composition, such as at A, its crystallization temperature or freezing point will be lowered to t_B. If this solution at A is cooled, ice crystals will begin to form at B. As a result, the solution will become richer in salt content. At any point C, the mixture will consist of ice at C_2 and solution at C_1. At point D, the solution will have the composition corresponding to D_1. On further cooling, the solution will freeze as a whole without any separation of ice crystals.

Similarly, if a brine solution of more than eutectic composition such as at E is cooled, it will first separate into salt and solution, until it reaches point D_1 again, whereafter the solution will freeze as a whole.

Brines are mainly used in industrial ice plants, cold storages, skating rinks, etc. Common brines are water solutions of calcium chloride ($CaCl_2$) and sodium chloride (NaCl). Their relevant properties are given in Tables 4.12 and 4.13.

Table 4.12 Properties of $CaCl_2$ Brine

Pure $CaCl_2$ *% Weight*	*Crystallization Temperature* °C	*Specific Gravity at* 15°C	*Specific Heat* kJ/(kg. K)
0	0	1.0	4.1868
5	−1.8	1.044	3.869
10	−5.4	1.087	3.58
15	−10.3	1.133	3.32
20	−18.0	1.182	3.086
25	−28.3	1.233	2.88
29.87 (Eutectic)	−55.0	1.29	2.74

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Table 4.15 Properties Data of R 12 and New Proposed Refrigerants for $t_0 = -25°C$ and $t_k = 55°C$.

Refrigerants	*NBP* °C	p_0 bar	p_k bar	p_2/p_1	p_k-p_0 bar	v_1 m^3/kg	h_{fg} kJ/kg	γ
R 290	– 42.07	2.02	19.07	10.71	17.05	0.2913	404.99	1.126
R 22	– 40.76	2.01	21.74	12.23	19.74	0.1496	223.72	1.166
R 12	– 29.79	1.24	13.61	13.47	12.37	0.1803	163.34	1.126
R 134a	– 26.2	1.07	14.92	17.64	13.85	0.2592	221.83	1.102
R 152a	– 25	0.98	13.32	17.67	12.34	0.4314	328.25	1.134
R 600a	– 11.73	0.59	7.82	21.25	7.23	0.8876	379.49	1.086
R290/R600a		1.4	14.22	13.29	12.82	0.4142	390.13	1.104

Table 4.16 Calculated Performance Parameters of R 12 and New Proposed Refrigerants Using Same Compressor as in R 12 Refrigerator

Refrig- erants	$\dot{m} \times 10^3$ kg/s	η_v	t_{2n} °C	t_{2s} °C	$\dot{Q}_o$ W	$\dot{Q}_k$ W	Q_v kJ/m^3	$\dot{W}_{is}$ W	$\dot{W}_a$ W	$V^* \times 10^3$ m^3/s/TR	W^* W/TR	COP = $\dot{Q}_o / \dot{W}_{is}$	τ_{stc} Nm
R 290	0.5896	0.936		125	155	282	808	88.4		4.35	2080	1.76	3.84
R 22	1.135	0.925		170	178	307	936	101.9		4.76	1995	1.75	4.42
R 12	0.8022	0.772	56.8	135	89	154	555	47.5	44.47	6.39	1786	1.84	2.77
R 134a	0.4437	0.608	60.5	125	65	113	518	33.86	32.46	6.79	1761	1.91	3.10
R 152a	0.2617	0.621	66.8	155	61	106	478	34.92	29.91	7.35	1723	1.75	2.77
R 600a	0.1941	0.894	53.7	110	48	86	260	25.85	24.68	13.54	1819	1.85	1.60
R 290/ R 600a	0.3039	0.612	70.8	116	77	129	609	35.0	28.1	5.78	2066	1.70	2.93

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In fact, the problem does exist in the cases of R 134a and R 152a. The solution to the problem is either to change the motor to a higher torque one, or to increase the starting torque of the existing motor by putting a capacitor in series with the starting winding.

The problem does not occur in the case of R 600a as the difference $(p_k - p_0)$ is small. But, it requires a larger V_p, and hence a higher starting torque of 2.93 Nm as against 2.77 Nm for R 12. But a larger displacement R 12 compressor would have a motor which would have a starting torque greater than 2.93 Nm, even if it is designed for R 12, because its $p_k - p_0$ is much more than that of R 600a.

4.14.4 Motor Ratings

R 134a, R 152a and R 600a require larger displacement compressors. The use of a larger displacement compressor, designed for R 12-without changing the motor installed in it which has a higher rating—would result in increase in energy consumption due to low efficiency as a result of operation under part-load conditions. The motor would have to be changed to a required lower rating motor so that it does not have to run at part loads causing increase in energy consumption.

In the case of R 134a, since the energy consumption is about 50% more than in the case of R 12, the motor rating may have to be increased.

On the other hand, R 22 and R 290 require smaller displacement compressors. The smaller displacement R 12 compressor would have a lower rating motor installed in it. The motor would, therefore, need to be changed to a required higher rating.

4.14.5 Refrigerant Charge

The quantities of refrigerants charged by Ashok Babu in his experiments on 165L refrigerator are given in Table 4.18.

Table 4.18 Quantities of Refrigerants Charged

Refrigerant	*Quantity Charged in Grams*
R 12	120
R 134a	135
R 152a	66
R 600a	60
50% R290/50%/R600a	40

In the experiment, these quantities are about 33% more due to the instruments, connecting lines, etc. The R 12 charge specified by the manufacturer was only 90 g.

Note that refrigerant chanre with hydrocarbons R 290 and R 600a is very small. This is essentially due to their very high latent heat of vaporization, and hence very low mass flow rate.

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$$(\dot{W})_{R12} = (\dot{m}w)_{R12} = 7.68 \times 10^{-4} (55.5 \times 10^3) = 43 \text{ W}$$

$$(\dot{Q}_k)_{R12} = (\dot{m}q_k)_{R12} = 7.68 \times 10^{-4} (192 \times 10^3) = 147 \text{ W}$$

With the same compressor, and hence for the same piston displacement, we have with propane

$$(\dot{m})_{R290} = \frac{\dot{V}_p}{v_1} = \frac{1.247 \times 10^{-4}}{0.27165} = 4.59 \times 10^{-4} \text{ kg/s}$$

$$(\dot{Q}_0)_{R290} = (\dot{m}q_0)_{R290} = 4.59 \times 10^{-4} (279.3 \times 10^3) = 128 \text{ W}$$

$$(\dot{W})_{R290} = (\dot{m}w)_{R290} = 4.59 \times 10^{-4} (135 \times 10^3) = 62 \text{ W}$$

$$(\dot{Q}_k)_{R290} = (\dot{m}q_k)_{R290} = 4.59 \times 10^{-4} (467.5 \times 10^3) = 215 \text{ W}$$

We clearly notice the following advantages in using the hydrocarbon HC 290 in place of CFC 12:

(i) Refrigerating capacity is increased from 43 W to 62 W, viz., by 44%.
(ii) COP *nearly* remains the same.
(iii) Discharge temperature is lowered from 128°C to 120°C.

However, there are implications in charging R 290 directly in a R 12 refrigerator without changing compressor, which are as follows:

(i) Refrigerant mass flow rate requirement decreases, due to higher latent heat and refrigerating effect of R 290

$$279.3 = (q_0)_{R290} > (q_0)_{R12} = 115.8 \text{ kJ/kg}$$

(ii) Refrigerant mass flow rate handled by compressor decreases due to large specific volume of R 290 suction vapour

$$0.27165 = (v_1)_{R290} > (v_1)_{R12} = 0.1624 \text{ m}^3\text{/kg}$$

This reduces mass flow handled by compressor as below:

$$4.59 \times 10^{-4} = (\dot{m})_{R290} < (\dot{m})_{R12} = 7.68 \times 10^{-4} \text{ kg/s}$$

(iii) Conditions (i) and (ii) are complimentary. However, the effect of higher latent heat is greater. As a result, in spite of the larger specific volume and hence smaller mass flow rate handled, the same compressor gives more capacity, and correspondingly increases heat rejected in the condenser for R 290 because of its higher latent heat of vaporization. Thus

$$128 \text{ W} = (\dot{Q}_0)_{R290} > (\dot{Q}_0)_{R12} = 89 \text{ W}$$

$$215 \text{ W} = (\dot{Q}_k)_{R290} > (\dot{Q}_k)_{R290} = 147 \text{ W}$$

(iv) Motor watts increase for R 290 along with higher refrigerating capacity.

$$62 \text{ W} = (\dot{W})_{R290} > (\dot{W})_{R12} = 43 \text{ W}$$

This will result in considerable overloading and overheating of motor. Consequently, it may burn out inspite of the lower discharge temperature of R 290.

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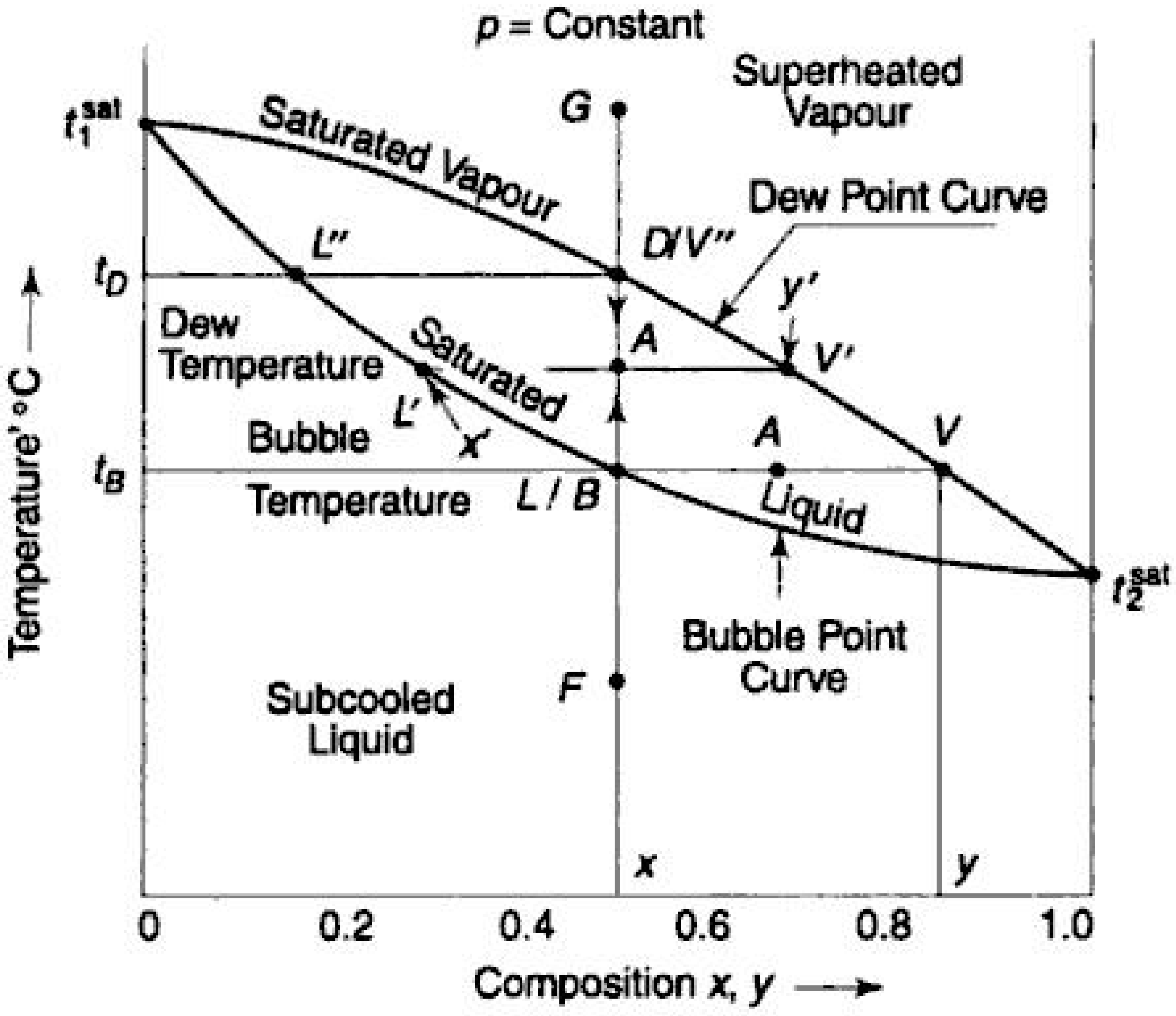

Fig. 4.13 *t-x-y* Diagram of Binary Mixtures

Similarly, the temperature t_D at which a vapour mixture begins to condense is called the *dew point temperature*. A curve passing through all such points is called the *dew point curve*. It is the locus of saturated vapour states. The region below the bubble point curve is the subcooled region and that above the dew point curve is the superheated region. The region between the two curves is the liquid plus vapour region. The states on these curves are the saturation states. Any state A that falls in the liquid plus vapour region comprises of equilibrium states L of saturated liquid and V of saturated vapour. The compositions of the two are ξ^L and ξ^V and the overall composition ξ at A is given in terms of the fraction z of vapour and $(1 - z)$ of liquid as given in Eq. (4.30)

$$\xi = z\xi^V + (1 - z)\xi^L \tag{4.30}$$

It may be noted that there will be a different loop like the one in Fig. 4.13 for each pressure.

Thus, when a subcooled liquid of composition x at F in Fig. 4.13 is heated it will first rise in temperature till it reaches the *bubble temperature* t_B at B/L when it begins to boil. The first vapour bubble that is formed is at V having the composition y. As the boiling proceeds, the temperature rises. Subsequent states during boiling are along the vertical such as point A at the same overall composition with liquid at L' and vapour at V' in equilibrium. When boiling is complete, the vapour is at D/V'' at *dew temperature* t_D. The last drop of liquid to vaporize is at L''.

Further heating will result in superheating of vapour to G. When superheated vapour at G is cooled, the reverse processes of desuperheating to D, condensation to B, and subcooling to F will take place.

It is observed that the *temperature glide/range* during evaporation and condensation is equal to the difference in dew and bubble temperatures $(t_D - t_B)$ for the composition.

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Another criterion of an ideal mixture is that the mixing of the constituents should cause no change in the average intermolecular forces of attraction. This amounts to saying that there is no change on mixing in the internal energy or volume. Accordingly, there is no evolution or absorption of heat on mixing. Thus, the enthalpy of an ideal mixture can be found by adding the enthalpies of the two components at the temperature and pressure of the mixture in proportion to the composition, i.e., $h = \xi_1 h_1 + \xi_2 h_2$.

Thus, from Figs 4.13 and 4.15 the enthalpies at L and V are as given in Eqs (4.36) and (4.37)

$$h^L = \xi_1^{\,L} [h_{f_1} - C_{f_1} (t_1^{\text{sat}} - t_A)] + \xi_2^L [h_{f_2} + C_{f_2} (t_A - t_2^{\text{sat}})] \quad (4.36)$$

$$h^V = \xi_1^{\,V} [h_{g_1} - C_{p_1} (t_1^{\text{sat}} - t_A)] + \xi [h_{g_2} + C_{p_2} (t_A - t_2^{\text{sat}})] \quad (4.37)$$

where C_f and C_p are the specific heats of liquid and vapour respectively at constant pressure. The expressions in brackets represent pure component enthalpies h_1^L and h_2^L for the liquid phase, and h_1^V and h_2^V for the vapour phase.

Example 4.3: Bubble and Dew Temperatures of Propane/Isobutane Mixtures (Illustrating Calculation Procedure)

Determine the bubble and dew temperatures of, by mass, (1) 50% R 290 + (2) 50% R 600a mixtures at 14 bar and 1.4 bar.

Solution: The molecular masses of R 290 and R 600a are $M_1 = 44$ and, $M_2 = 58$.

Given $\xi_1 = 0.5$ and $\xi_2 = 0.5$. From Eqn. (4.29),

$$x_1 \text{ or } y_1 = \frac{M_2\,\xi_2}{M_1\,\xi_1 + M_2\,\xi_2} = \frac{58(0.5)}{44(0.5) + 58(0.5)} = 0.5686$$

Solving Eqn. (4.34)

$$p = x_1\, p_1^{\text{sat}} + x_2\, p_2^{\text{sat}}$$

by iteration for $x_1 = 0.5686$ and $x_2 = 1 - x_1 = 0.4314$ for $p = 14$ bar, and 1.4 bar, we find that

$$(t_B)_{14\text{ bar}} = 54.3°\text{C}$$

$$(t_B)_{1.4\text{ bar}} = -25°\text{C}$$

Solving Eqn. (4.35)

$$p = \frac{p_1^{\text{sat}}\;p_2^{\text{sat}}}{p_1^{\text{sat}} - y_1\left(p_1^{\text{sat}} - p_2^{\text{sat}}\right)}$$

by iteration for $y_1 = 0.5686$ and $y_2 = 0.4314$, we get

$$(t_D)_{14\text{ bar}} = 62.7°\text{C}$$

$$(t_D)_{1.4\text{ bar}} = -16.3°\text{C}$$

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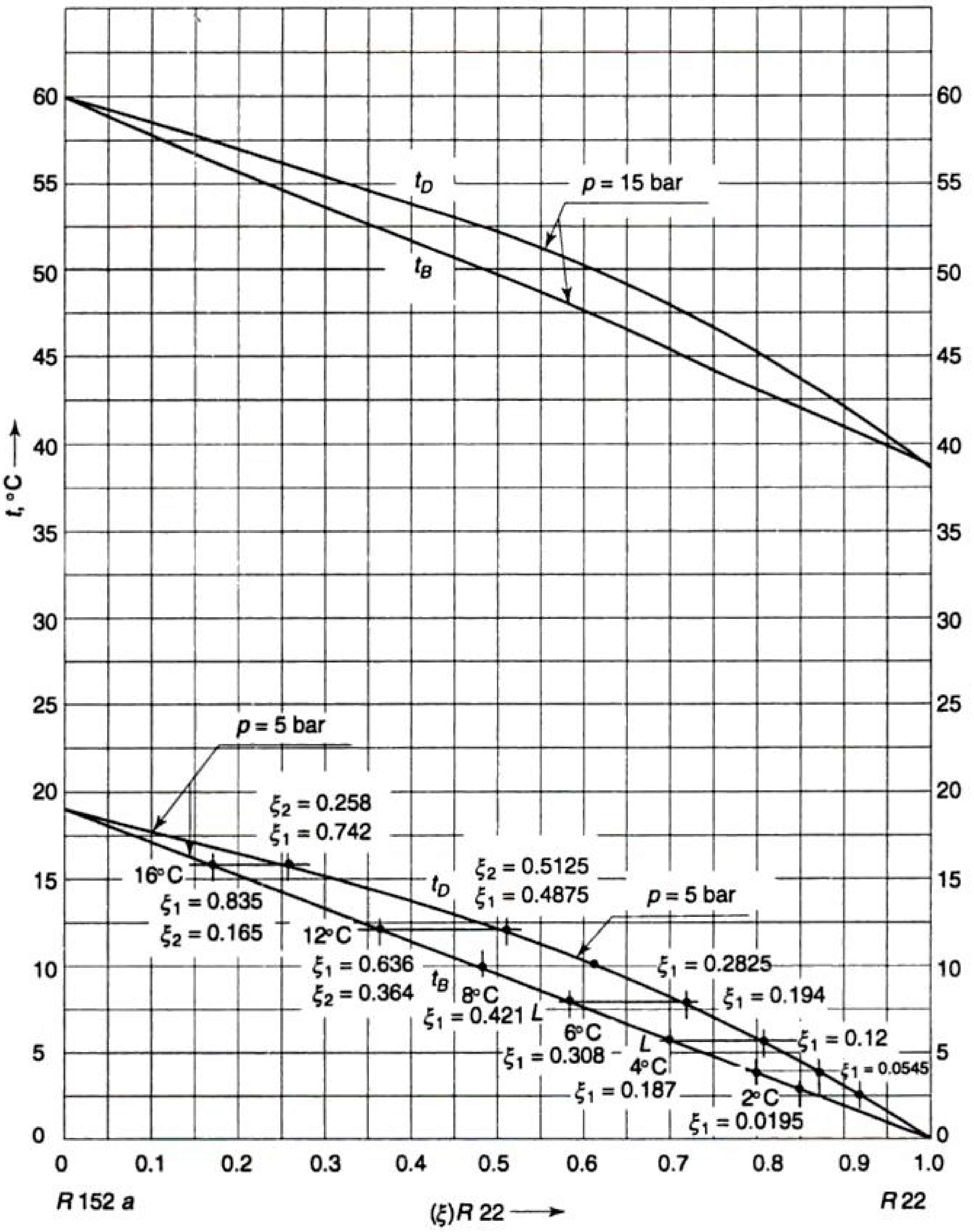

Fig. 4.18 Temperature-Composition Diagram of R 152a/R 22 Mixtures at 15 and 5 bar Pressures

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4.20.3 Calculation of Enthalpy of Mixture in Vapour Phase

A method developed by Agarwal and Arora[2] will now be described.

Figure 4.20 shows the vapour-liquid domes of pure components 1 and 2, and their binary mixture of certain composition on a p-h diagram.

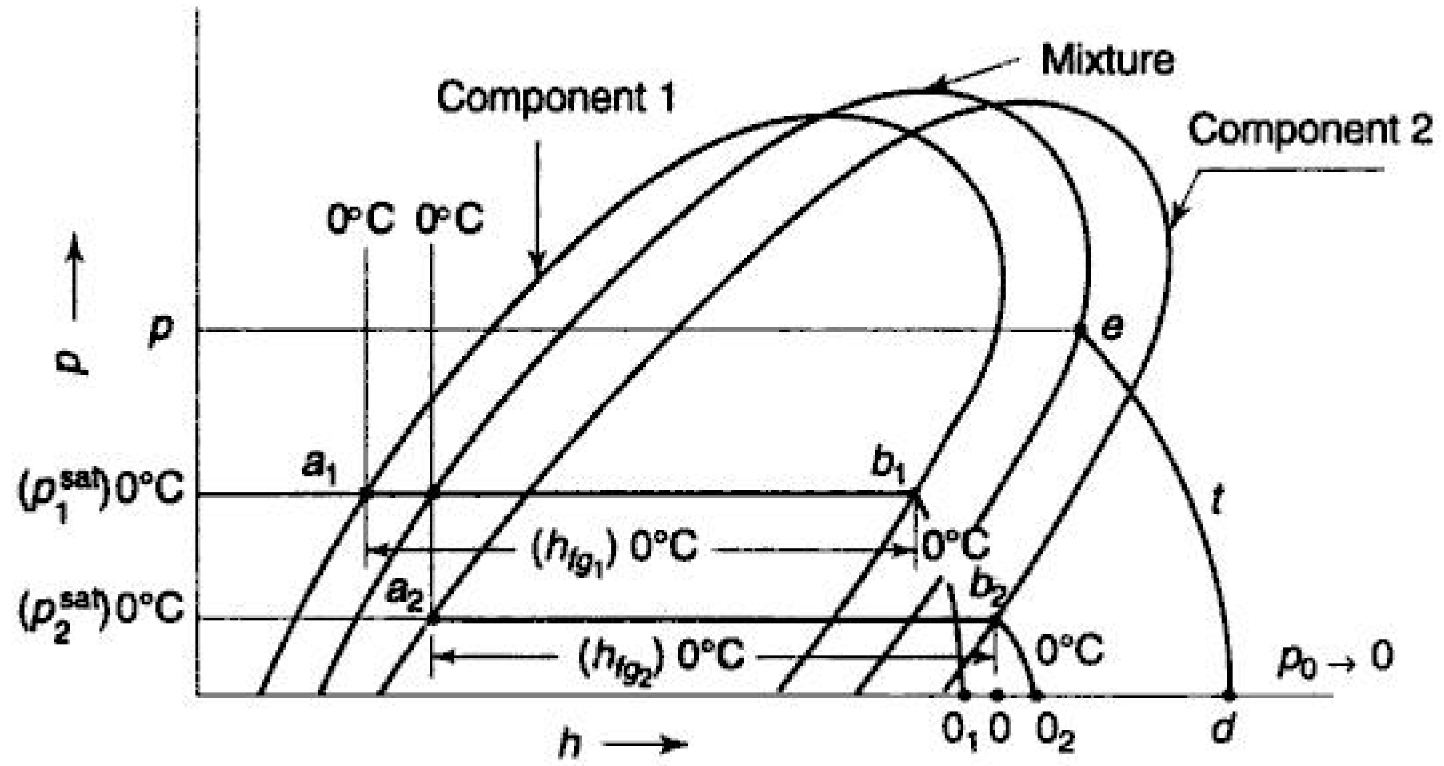

Fig. 4.20 Proposed Method for Vapour Mixture Enthalpy Calculation

The figure illustrates how the enthalpies of saturated liquid and saturated vapour may be calculated. The proposed method assumes values for reference state enthalpies for saturated liquid states of both the pure components 1 and 2 at a_1 and a_2 at 0°C.

Consider now the saturated vapour state e on the dome for the mixture as shown in Fig. 4.20. Let the pressure and temperature of the mixture at e be p and T respectively.

Now, the state at the same temperature T but at zero pressure is shown by point d in the figure. Then, the enthalpy at e is related to ideal gas state enthalpy at d at temperature T by the residual enthalpy term. Thus

$$h_e = h_d + |h - h^{id}|^{p}_{p_0 \to 0}$$

The enthalpy h_d, is related to the ideal gas state enthalpy h_0 at point 0 at t_0 = 0°C (T_0 = 273.15 K) and $p_0 \to 0$ by the relation

$$h_d - h_0 = \int_{T_0}^{T} C_{p_0}\, dT$$

where C_{p_0} is the zero-pressure constant pressure specific heat of the mixture. Now, h_0 is found from

$$h_0 = \xi_1 h_{0_1} + \xi_2 h_{0_2}$$

where h_{0_1} and h_{0_2} are ideal gas state enthalpies of components at point o_1 and o_2. These, in turn are related to real gas saturated vapour state enthalpies h_{b_1} and h_{b_2} by residual enthalpy terms, while h_{b_1} and h_{b_2} are given by

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It is also to be noted that a maximum boiling azeotrope, on account of the negative heat of mixing, will have a higher latent heat of vaporization compared to the molal average latent heat. On the contrary, a minimum boiling azeotrope will have a lower latent heat.

4.22 USE OF MINIMUM AND MAXIMUM BOILING AZEOTROPES

Eiseman[21] has done a study of a large number of azeotropes. Most fluorocarbon mixtures have positive deviations from Raoult's law. Thus, they form minimum boiling azeotropes.

The main property of a minimum boiling azeotrope is its boiling point which is lower than the N.B.P. of even the lower boiling component. It enables the system with an azeotrope to have a higher evaporator pressure as compared to the lower boiling component. This, in general, results in an increased density of the suction vapour and, therefore, higher capacity.

Further, due to the depression of the boiling point of the azeotrope, slope '*b*' of the vapour pressure line of the azeotrope is flatter than the slope of the line of the lower boiling component. Thus, below a certain temperature, say point *A*, in Fig. 4.23, the vapour pressure of the azeotrope is higher, and above this temperature, it is lower than the vapour pressure of the lower boiling component. Accordingly, for the azeotrope, the evaporator pressure p_0 at T_0 is higher, and the condenser pressure p_k at T_k is lower than the values for the lower boiling component. Thus the compression ratio and discharge temperature are considerably reduced.

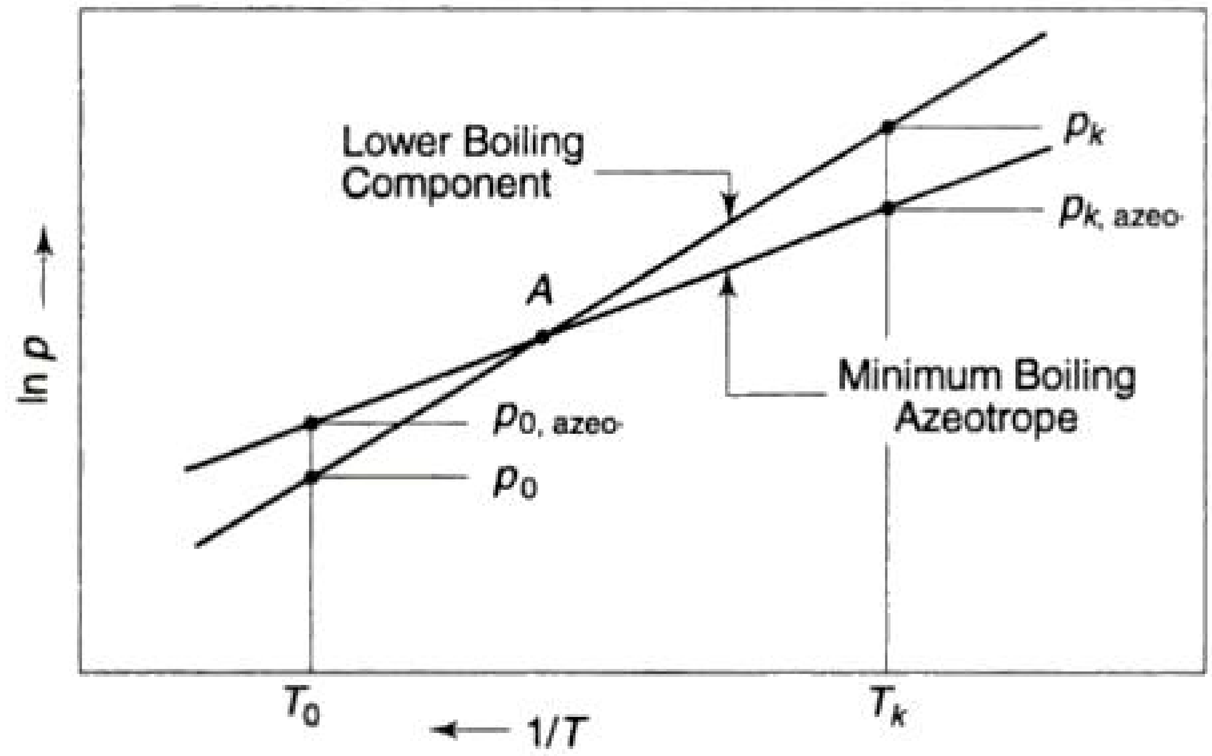

Fig. 4.23 Comparison of ln p-$1/T$ Diagrams of a Minimum boiling Azeotrope and Lower Boiling Component

It is not intended here to describe all the known azeotropes. However, Table 4.27 gives examples of such common minimum boiling azeotropes.

Refrigerant 500:

R 500 was discovered in 1950 as a special refrigerant of the Carrier Corporation under the trade name of Carrene 7. This azeotrope consists of 73.8 per cent R 12 and 26.2 per cent R 152, difluoroethane.

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$$\mathcal{E}_{\text{Carnot}} = \frac{T_0}{T_k - T_0}$$

we find that due to the increase in average temperature T_0 and decrease in average temperature T_k, the COP of the cycle will be increased. The thermodynamic obstacle to the elimination of irreversibility of the heat transfer processes can thus be removed by a judicious selection of a mixture of refrigerants and its composition.

It can be seen from Fig. 4.24 that the use of mixed refrigerant could decrease the work by

area Δw_0 due to non-isothermal evaporation and

area Δw_k due to non-isothermal condensation.

The area $\Delta w_0 = \Delta q_0$ also represents increase in refrigerating effect. It would, therefore, result in increase in COP and hence power saving.

Consider for example a case in which air is cooled from t_{c_1} = 25°C to t_{c_2} = 15°C, and temperature of cooling air in condenser rises from t_{h_1} = 45°C to t_{h_2} = 55°C. These are typical values for a window-type air conditioner. Then if pure refrigerant is used, the practical evaporator and condenser temperatures could be t_0 = 0°C and t_k = 65°C. The corresponding value of Carnot COP would be 4.55.

Now, if a mixture of refrigerants is used for evaporating from t_0 = 0°C to t_0' = 10°C, and condensing from t_k = 60°C to t_k' = 50°C, then the Carnot COP would be 5.55. The saving in power would be 18%.

Note ✍: *It is important to observe that this effect can be utilized only if there is counterflow of the two fluids which is possible only if the mixture is boiling and condensing inside tubes as in window-type air conditioner. Otherwise, if for example, the condensation is outside tubes as in shell and tube condensers, the condensation temperature will rise above the outlet temperature of the coolant ($t_k' \geq 55$°C and $t_k > t_k'$). This will cause increase in average temperature of heat rejection, and hence condenser pressure, work and power consumption.*

Example 4.5 Vapour Compression Cycle of a Hydrocarbon Mixture Refrigerator on *p-h* Diagram

A hydrocarbon refrigerator operating on equal proportions of propane and isobutane by mass works on simple saturation cycle. The condenser and evaporator pressures are 14 bar and 1.4 bar. Show the cycle on *p-h* diagram and determine its discharge temperature and theoretical COP.

Solution

The values of bubble and dew temperatures at these pressures have already been determined in Example 4.3. Accordingly, the *p-h* diagram is drawn as shown in Fig. 4.25.

From tables of properties for 50-50% R 290/R 600a mixture in Appendix

$$h_1 = 545.0 \text{ kJ/kg}, \; h_3 = 341.15 \text{ kJ/kg} = h_4$$

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The two equations can be solved for t_d by iteration. The equilibrium L-V states at 5 bar from Tables 4.23 and 4.25 are plotted on t-ξ and h-ξ diagrams in Figures 4.27 (a) and 4.27(b). It is seen that state d is at a temperature slightly above 4°C. The method of solution by iteration is as follows:

1. Assume t_d.
2. Find ξ_1^L and ξ_1^V.
3. Find z from the composition-balance equation.
4. Verify if enthalpy-balance equation is satisfied.
5. If not, assume another value of t_d.

The solution is

$$t_d = 4.1°\text{C}$$

Theoretical COP of the cycle

$$\varepsilon = \frac{h_a - h_d}{h_b - h_a} = \frac{428 - 258}{455.8 - 428} = \frac{170}{27.8} = 6.1$$

Note ✍: *Ghosh and Sarkar*[22] *obtained a saving of 2.7 to 4.3% in power consumption with 9.87% R152a in R 22/R 152a mixture.*

Example 4.6(b) Estimating Power Saving in Air Conditioners with R 152a/R 22 Mixture.

Find the theoretical COP of R 22 cycle equivalent to the mixture cycle in Example 4.6(a). Estimate the power saving, and discuss the other effects.

Solution

In order to maintain the condenser and evaporator sizes same, we have to design for same LMTD in the two heat exchangers. This means, we have to maintain mean condensation and evaporation temperatures as same. Hence, for the equivalent R 22 cycle,

$$t_k = \frac{t_e + t_c}{2} = \frac{44.75 + 43}{2} = 43.9°\text{C}$$

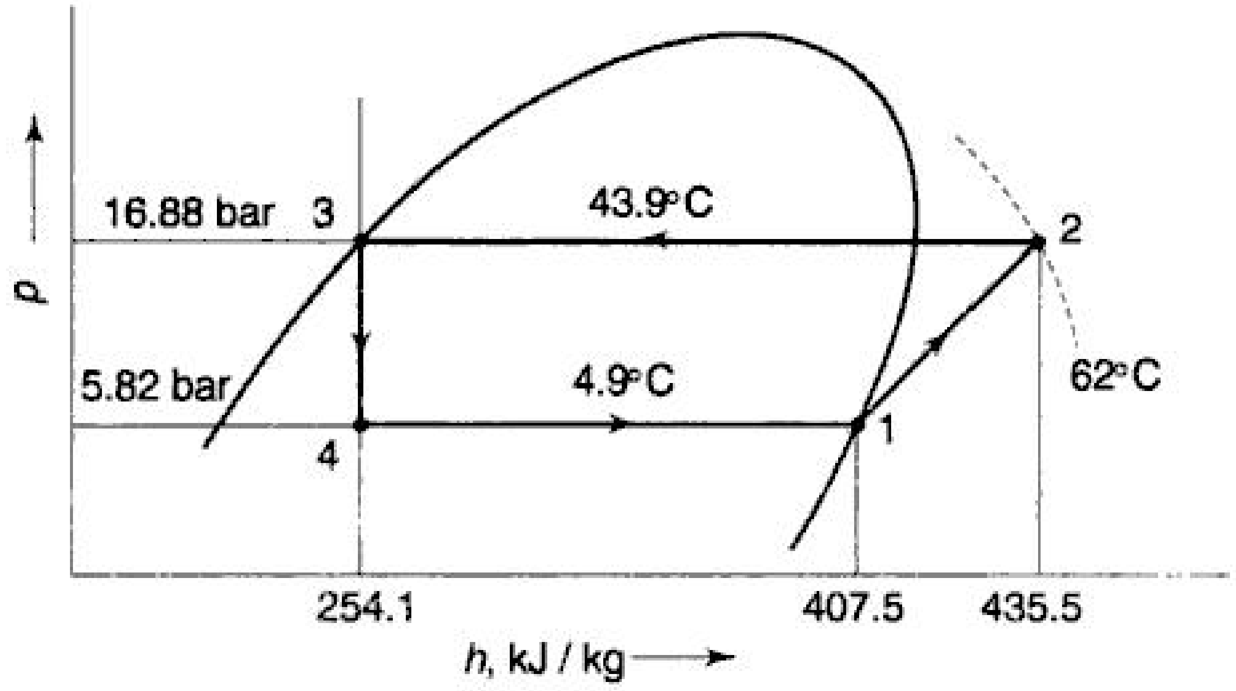

Fig. 4.28 R 22 Cycle for Example 4.6(b)

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4.2 Refrigeration engineers usually presume that if a R 12 car air conditioner compressor is operated with R 134a, its 'cooling capacity' would fall by about 10%. Examine this assumption by a realistic vapour compression cycle analysis.

4.3 Calculate the latent heat of vaporization of R 12 at – 25°C and + 50°C. The p^{sat} versus T^{sat} relationship of R 12 is

$$\ln p^{\text{sat}} = 73.478 - \frac{4200}{T^{\text{sat}}} - 10.97 \ln T^{\text{sat}} - 0.01675\, T^{\text{sat}}$$

where p^{sat} is in bar and T^{sat} in K. Use the specific volumes data from the table of properties of R 12.

Also, find the constants a and n if the h_{fg} versus T^{sat} relationship can be expressed by the equation

$$h_{fg} = a\left(1 - \frac{T^{\text{sat}}}{T_c}\right)^n$$

Critical temperature for R 12 is $T_c = 385.15$ K

4.4 What are azeotropic and non-azeotropic mixtures? Explain, in brief, their advantages giving examples.

4.5 Calculate and compare the specific volume of suction vapour, refrigerating effect, mass flow rate, discharge pressure and temperature, piston displacement, power consumption and COP for a 150 W refrigerating capacity domestic refrigerator operating on simple saturation cycle with – 25°C evaporator and 55°C condenser temperatures for R 290, R 22, R 12, R 134a, R 152a and R 600a. Also compare their pressure drops across capillary.

4.6 If R 12 refrigerator in Prob. 4.5 is charged with propane without changing the hermetic compressor, what would be the motor wattage, isentropic discharge temperature, and heat required to be rejected in condenser? Assume the same operating conditions, and ignore the effect on capillary. At what refrigerating capacity, would the refrigerator be operating with propane? What are the implications of charging R 12 refrigerator with propane?

4.7 (a) A propane refrigerator has to operate at $t_k = 55$°C and $t_0 = -25$°C. For maximum COP, find if the suction state should be in wet or superheat region.

(b) Also find if it is advantageous to use liquid vapour regenerative heat exchanger.

4.8 Draw t-ξ and h-ξ diagrams for R 134a/R 22 mixtures for 2, 4, 6, 10, 15, 20 bar pressures. Do calculations for bubble and dew temperatures and enthalpies assuming ideal mixtures for 10, 20, 30 and 50 per cent R 134a.

4.9 Find the bubble and dew pressures of R 134a/R 22 mixtures at 0, 5, 10, 45, 55 and 60°C temperatures and 10, 20, 30 and 50 per cent R 134a compositions.

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Flash gas removal with multistage compression, therefore, results in power economy, and is always desirable whichever be the refrigerant used.

Another method of obtaining the same result as that of flash gas removal is to employ the flash chamber as a *liquid subcooler* as shown in Fig. 5.2. The liquid subcooler subcools the liquid by the evaporation of the liquid refrigerant in the flash chamber. Figure 5.2 also gives the mass balance of the liquid subcooler. Herein, m_1 is the mass flow rate through first-stage compressor, and m_2 is the mass flow rate through second-stage compressor. The thermodynamic states correspond to those of Fig. 5.1 (b) except that the state of liquid entering the evaporator expansion valve shifts to 8′ from 8.

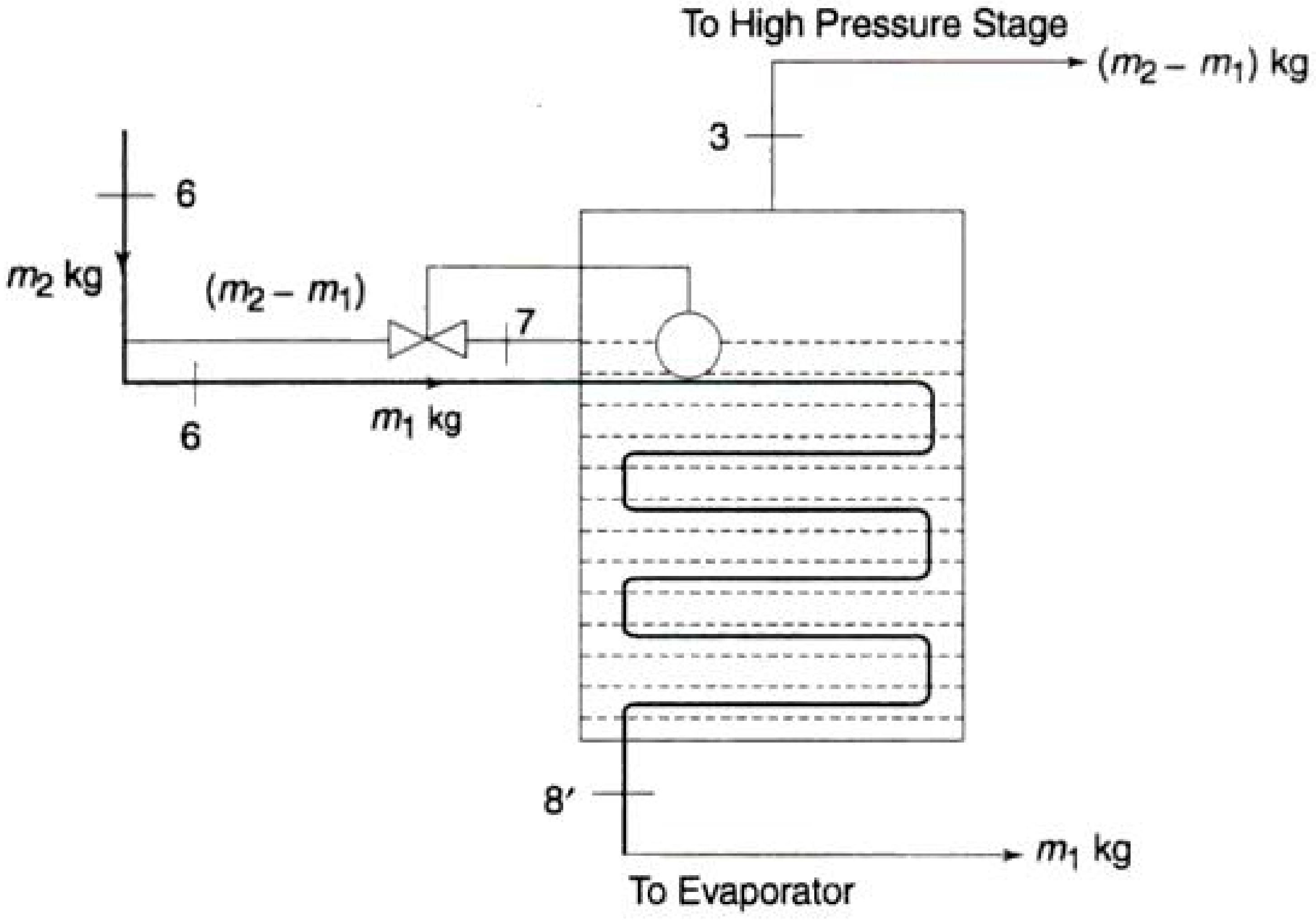

Fig. 5.2 Flash Chamber as a Liquid Subcooler

5.2.2 Flash Intercooling

For flash intercooling, the compressed vapours from the lower stage are led and bubbled through the liquid in the flash chamber as shown in Fig. 5.3(a). The vapours are thus cooled to the saturation temperature at the pressure of the flash chamber and a part of the liquid evaporates which goes to the higher stage along with the vapours from the lower stage. Flash intercooling thus enables the higher stage compression to take place along the steeper isentropic, nearer the saturated vapour line.

Figures 5.3 (a) and (b) show that the discharge vapours from the lower stage at 2 are cooled to 3 by the evaporation of a part of the liquid refrigerant from the flash chamber at 6. Thus, the vapours entering the high-stage compressor m_2 comprise these vapours in addition to the vapours m_1 from the low-stage.

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densing temperature—has a flash intercooling system with a liquid subcooler. The vapour leaving the evaporator is at –30°C and entering the first-stage compressor is at –15°C. The vapour leaving the flash chamber is superheated by 10°C in the suction line to the second-stage compressor. Water intercooling is done to cool the vapour to 45°C. Adiabatic efficiencies of both compressors are 0.75. The volumetric efficiencies of first- and second-stage compressors are 0.65 and 0.77 respectively. Find the piston displacements, discharge temperatures and power requirements of the two compressors.

Solution

See Fig. 5.5. Reference state is - 40°C at which $h_f = 0$, $s_f = 0$.

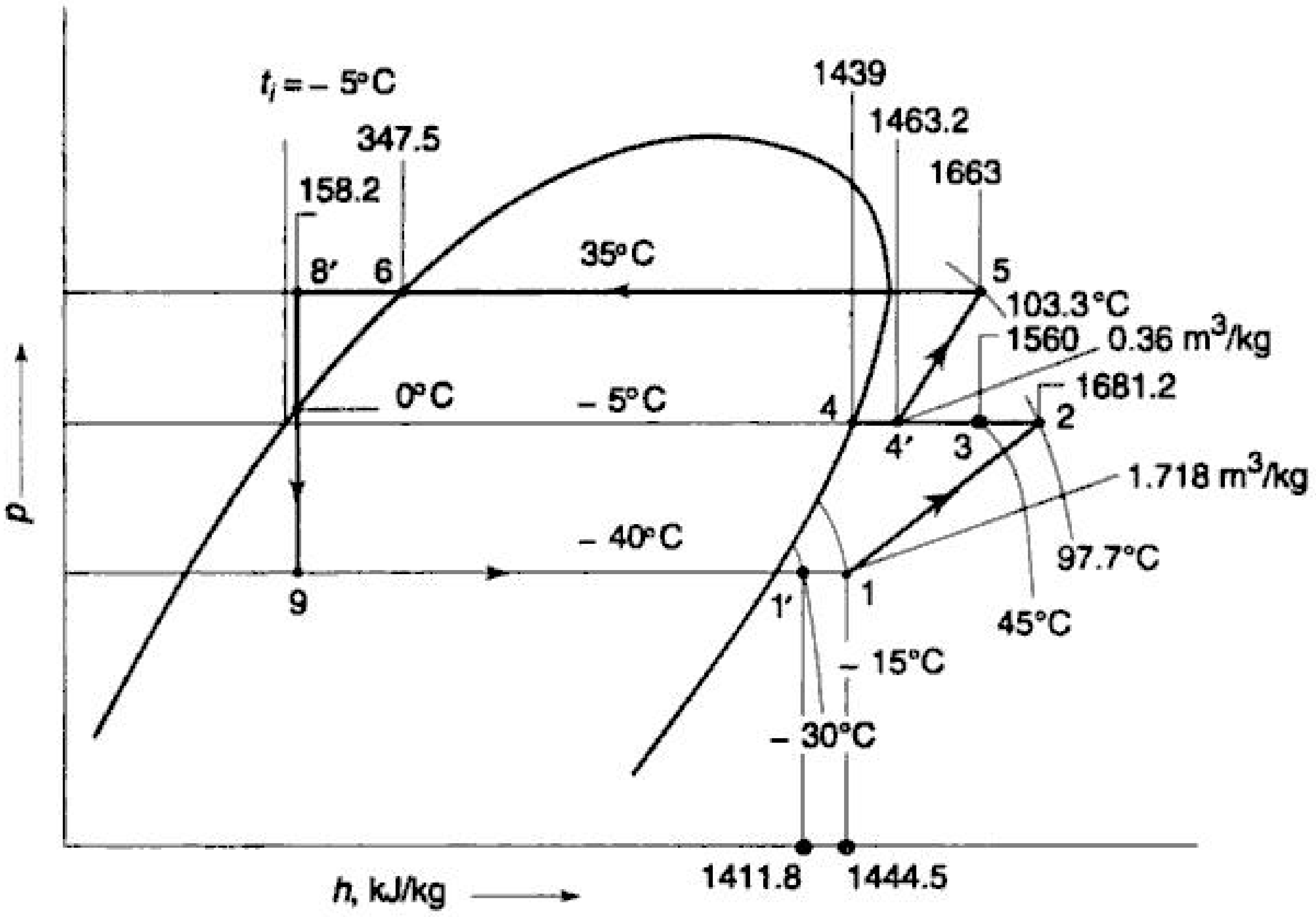

Fig. 5.5 Figure for Example 5.1

$$p_k = 1.35 \text{ MN/m}^2 \text{ (at } 35°\text{C)}$$

$$p_0 = 0.0718 \text{ MN/m}^2 \text{ (at } - 40°\text{C)}$$

$$p_i = \sqrt{1.35 \times 0.0718 \times \left(\frac{273 + 35}{273 - 40}\right)} = 0.355 \text{ MN/m}^2$$

The corresponding saturation temperature is

$$t_i = - 5°\text{C}$$

First Stage

Enthalpy of vapour leaving the evaporator

$$h_1' = 1390 + \{ - 30 - (- 40)\} \left(\frac{1499 - 1300}{50}\right)$$

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The enthalpy of the vapour mixture entering the compressor is

$$h_1 = \frac{m_1 h_7 + m_2 h_6}{m_1 + m_2} \tag{5.3}$$

and the net work done is given by

$$W = (m_1 + m_2)(h_2 - h_1) \tag{5.4}$$

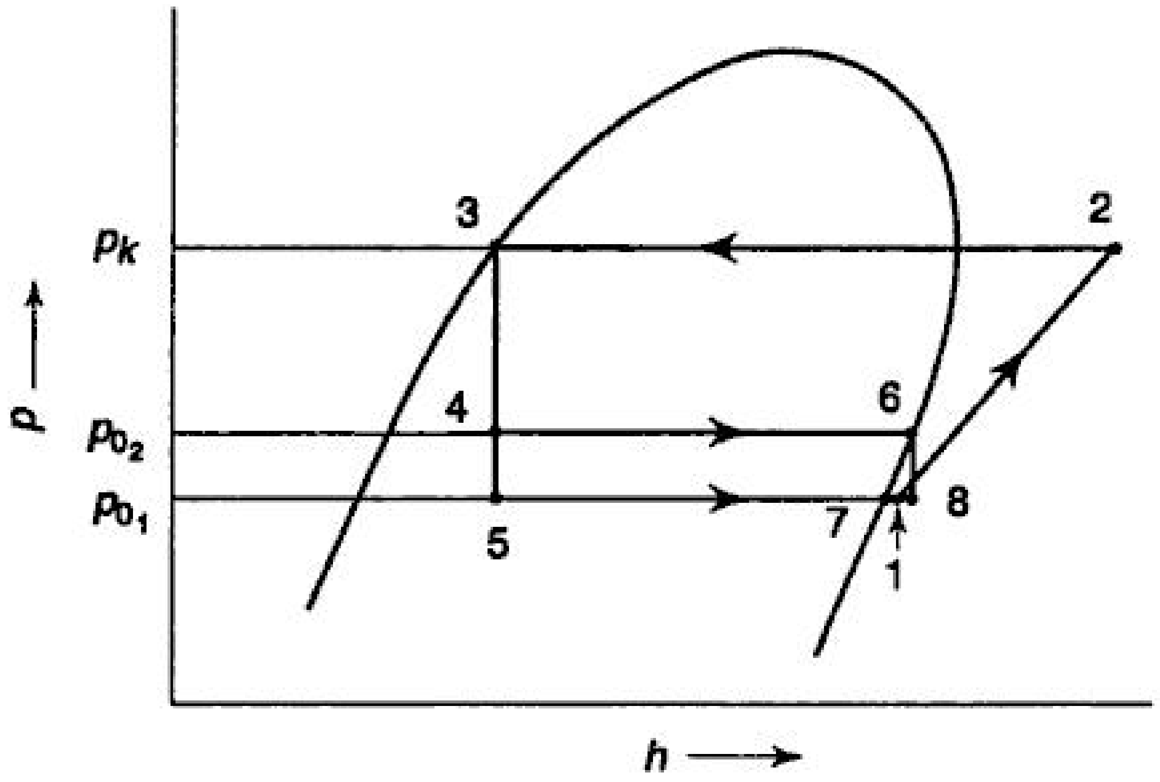

Fig. 5.6(b) Thermodynamic Cycle for the System of Fig. 5.6(a)

5.3.2 Single Compressor—Multiple Expansion Valves

The operation of a two-evaporators single-compressor system with *multiple arrangement* of expansion valves is shown in Figs 5.7(a) and (b). The only advantage of the arrangement is that the flashed vapour at the pressure of the high temperature evaporator is not allowed to go to the lower temperature evaporator, thus improving its efficiency. To gain thermodynamic advantage from this, it will be necessary to use individual compressors for each evaporator, thus eliminating the throttle losses of the back pressure valves.

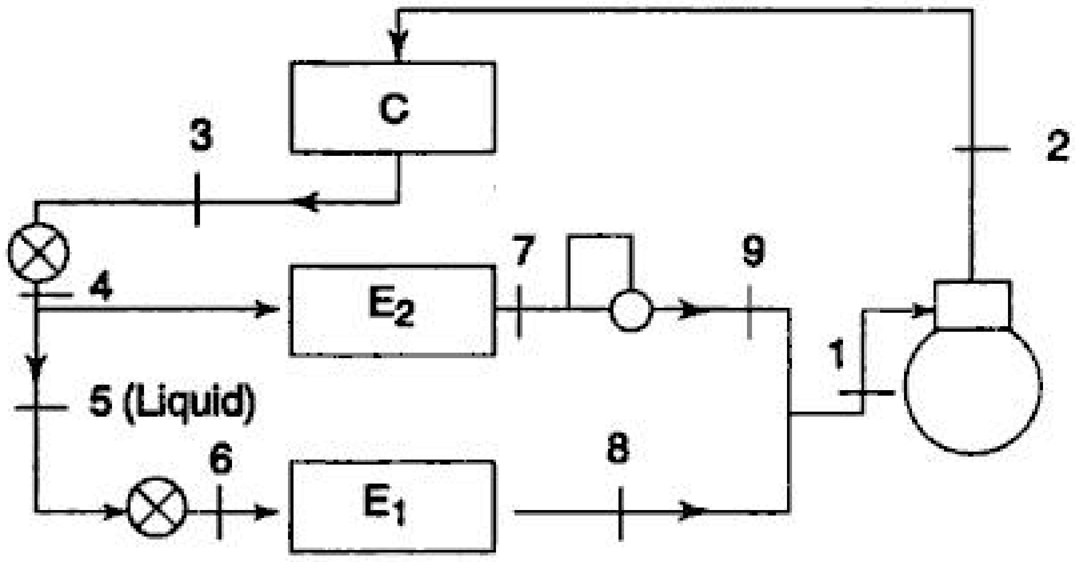

Fig. 5.7(a) System with Two Evaporators and Single Compressor with Multiple Expansion Valves

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(i) Solidification temperature of the refrigerant.
(ii) Extremely low pressures in the evaporator and large suction volumes if a high-boiling refrigerant is selected.
(iii) Extremely high pressures in the condenser if a low-boiling refrigerant is selected.
(iv) Very high pressure ratio p_k/p_0 and, therefore, a low coefficient of performance.
(v) Difficulties encountered in the operation of any mechanical equipment at very low temperatures.

We know that multistage compression is employed when low evaporator temperatures are required and when the pressure ratio p_k/p_0 is high. Refrigerant 22 is used in a two-stage system up to –50°C and in a three-stage system up to about –65°C.

If vapour compression systems are to be used for the production of low temperatures, the common alternative to stage compression is the *cascade system* in which a series of refrigerants, with progressively lower boiling points, are used in a series of single-stage units. The system provides a solution to all the problems mentioned above except the last one.

The cascade system combines two or more vapour compression units as shown in Fig. 5.10. The high temperature cascade produces refrigeration at a certain low temperature t_{0_2}. The low temperature cascade produces refrigeration at a still further low temperature t_{0_1}, using the refrigerating effect of high temperature cascade at temperature t_{0_2} for rejecting heat in its condenser at temperature t_{k_1}, which in the limit is equal to t_{0_2}. In practice, however, there is a certain overlap between these temperatures, i.e. the temperature t_{k_1} is about 5°C higher than the temperature t_{0_2}.

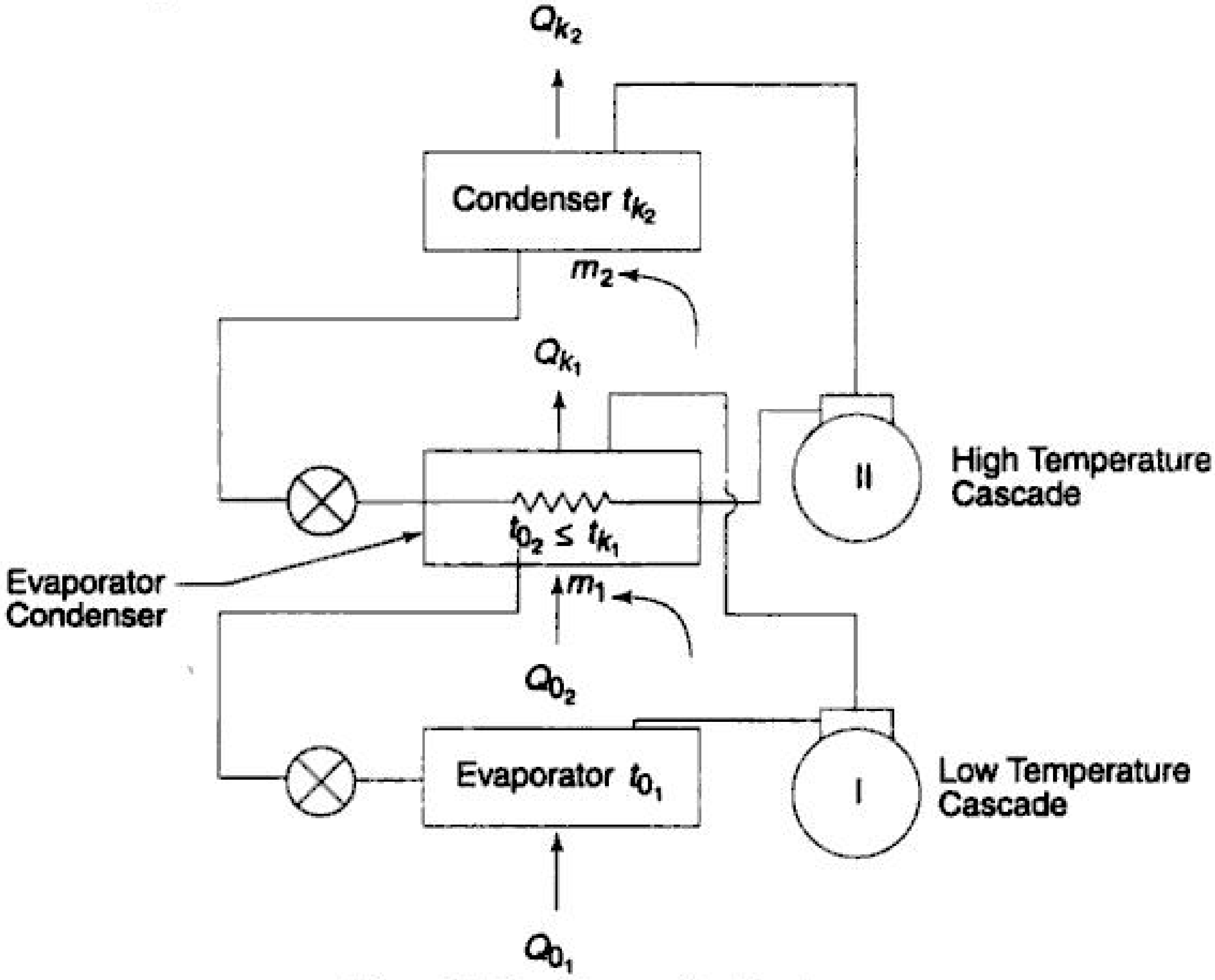

Fig. 5.10 Cascade System

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Another problem in the process of manufacture is the blocking of the expansion device by the formation of dry ice. This is eliminated by first producing liquid carbon dioxide at a pressure slightly above the triple point pressure and then reducing its pressure to one atmosphere in a snow chamber. This makes it necessary to have two snow chambers, one receiving liquid carbon dioxide at a pressure higher than the triple point pressure, and the other reducing pressure to form solid carbon dioxide and vapour, both working alternately. Such a method is known as the *pressure snow chamber method*.

Thus if 6 bar is chosen as the pressure of liquid formation in the snow chamber, then the various stage pressures would be nearly equal to 69, 20, 6 and 1.01325 bar respectively.

Further, the condensing temperature of 28°C is impractical when the cooling water temperature itself is 30°C or more in summer. Also, even with 28°C, the COP of the system would be very low as it is close to the critical temperature. The high pressure stage of the carbon-dioxide cycle is, therefore, replaced with a separate ammonia circuit in cascade with the carbon dioxide circuit.

The schematic diagram of a system using the pressure snow chamber method and ammonia in the cascade circuit is shown in Fig. 5.12. The condensing pressure of carbon dioxide is generally kept at 15 to 20 bar. The *p-h* diagram of the carbon dioxide circuit is shown in Fig. 5.13. In the position shown in Fig. 5.12, high pressure liquid carbon dioxide is being prepared in snow chamber II at 6 bar, whereas snow chamber I is shown to be reducing in pressure from 6 bar to 1.01325 bar. Dry ice is formed at 12 and the vapour leaves at 13. During this cycle of operation, expansion valve A is open and B is closed. Also, shut-off valves C and F are open and D and E are closed.

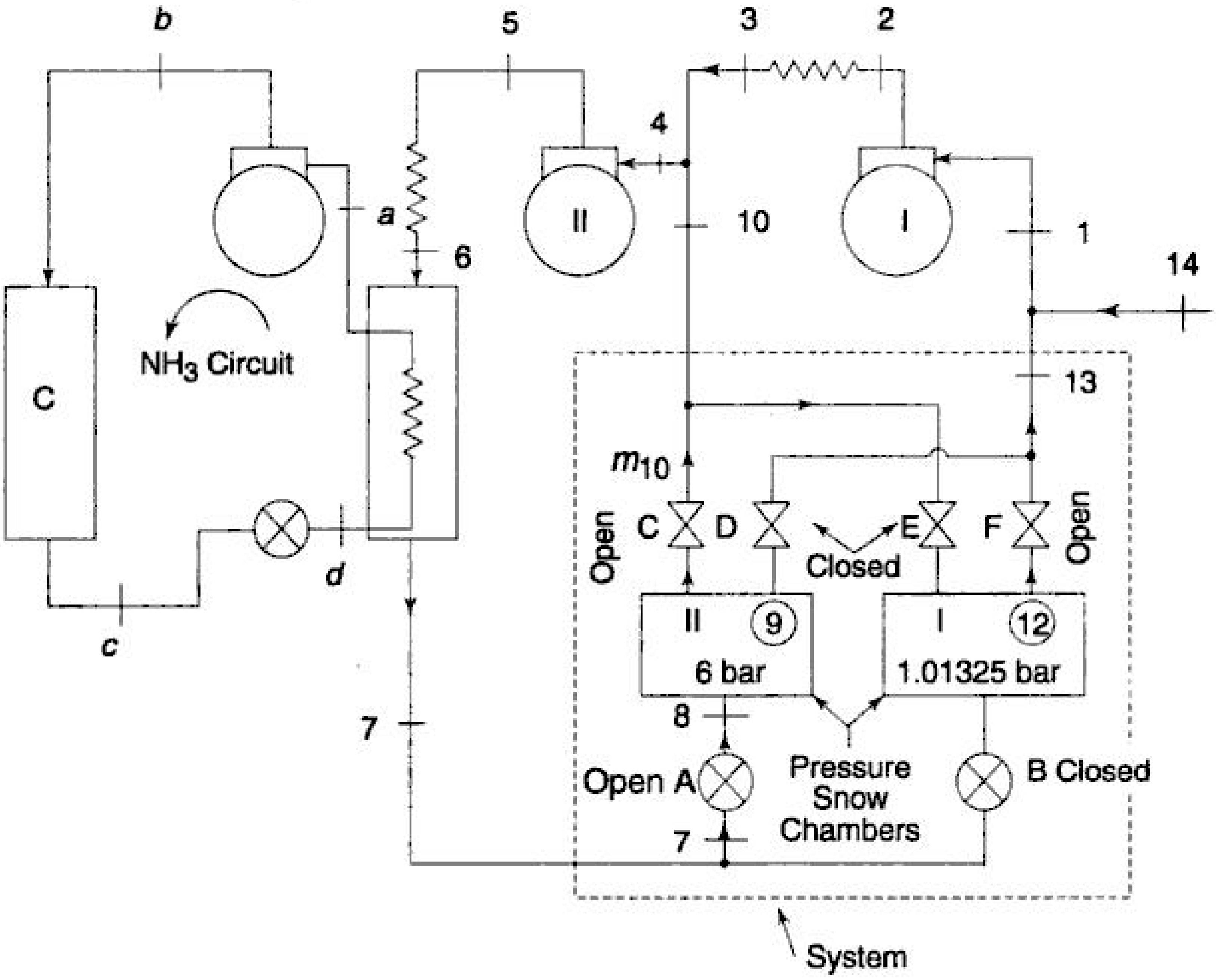

Fig. 5.12 Pressure Snow Chamber Method with Ammonia in the Cascade Circuit for Manufacture of Dry Ice

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$$= \frac{500}{3600}\,(0.66)\,(1680 - 1450)$$

$$= 21.08 \text{ kW}$$

Total power requirement

$$= 12.31 + 30.6 + 21.08 = 64 \text{ kW}$$

5.7 SYSTEM PRACTICES FOR MULTI-STAGE SYSTEMS

Rotary vane or centrifugal compressors are commonly used for the booster stage of a multistage system where large gas volumes are handled. There are also some applications in which oil-free and dry-cylinder compressors are used where the effects of oil in the refrigerant are of great consideration.

When LP stage temperatures are below –70°C, the pull-down load may be three times the normal. Compressor motors are, therefore, selected for about 150 per cent above the normal loading.

Some thought must also be given for sizing the condenser for the maximum amount of heat rejection that is expected during the pull-down period.

One problem in low temperature evaporators is the return of oil to the compressor. Another is the pressure drop through the evaporator. A D-X (direct-expansion) type evaporator is the most common because of its improved ability to return oil as well as the smaller charge of the refrigerant required with it. Sometimes an oil separator is used even with R 12 in low-temperature applications such as freeze-drying. The effect of pressure drop in evaporators at very low pressures is very serious because of its large magnitude which is a result of the large increase in volume of the refrigerant during vaporization. Hence, pumped systems such as a *flash-cooler type* evaporator can be used, in which the refrigerant is used like brine under pressure by a pump from a flash chamber so that vaporization does not take place inside the evaporator, but only on return to the flash cooler. Or a *recirculation type* evaporator may be used.

Receivers should not be used as the temperature of the surroundings may make it difficult for the liquid to enter into it.

It may be necessary to raise the temperature of the low-stage suction gas for maintaining the lubricating ability of the oil in the compressor. In cascade systems, this can be best done through a heat exchange between this gas and the high pressure stage liquid.

It is desirable to charge the system with a *fade-out charge* such that on shut down, all liquid in the system will evaporate into gas without excessively increasing the system pressure. In practice, however, the system may not have enough volume to permit this. In that case, an expansion tank may be provided and connected to the evaporating side of the system.

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6.1.1 Work in Reciprocating Compressor[1]

The *p-v* diagram for the *machine cycle* of a reciprocating compressor is shown in Fig. 6.1 along with the skeleton diagram of the cylinder and piston mechanism. When the piston is in the extreme left position of the *inner dead centre* (IDC), the volume occupied by the gas is $V_c = V_3$ called the *clearance volume*, i.e., the volume between the IDC position of the piston and the cylinder head. As the piston moves outward, the clearance gas expands to 4, where the pressure inside the cylinder is equal to the pressure at the *suction flange* of the compressor. As the piston moves further, the *suction valve S* opens and the vapour from the evaporator is sucked in till the extreme right position of the *outer dead centre* (ODC) is reached. At this point the volume occupied by the gas is V_1. The *stroke* or *swept volume* or *piston displacement* is

$$V_p = (V_1 - V_3) = \frac{\pi D^2}{4} L$$

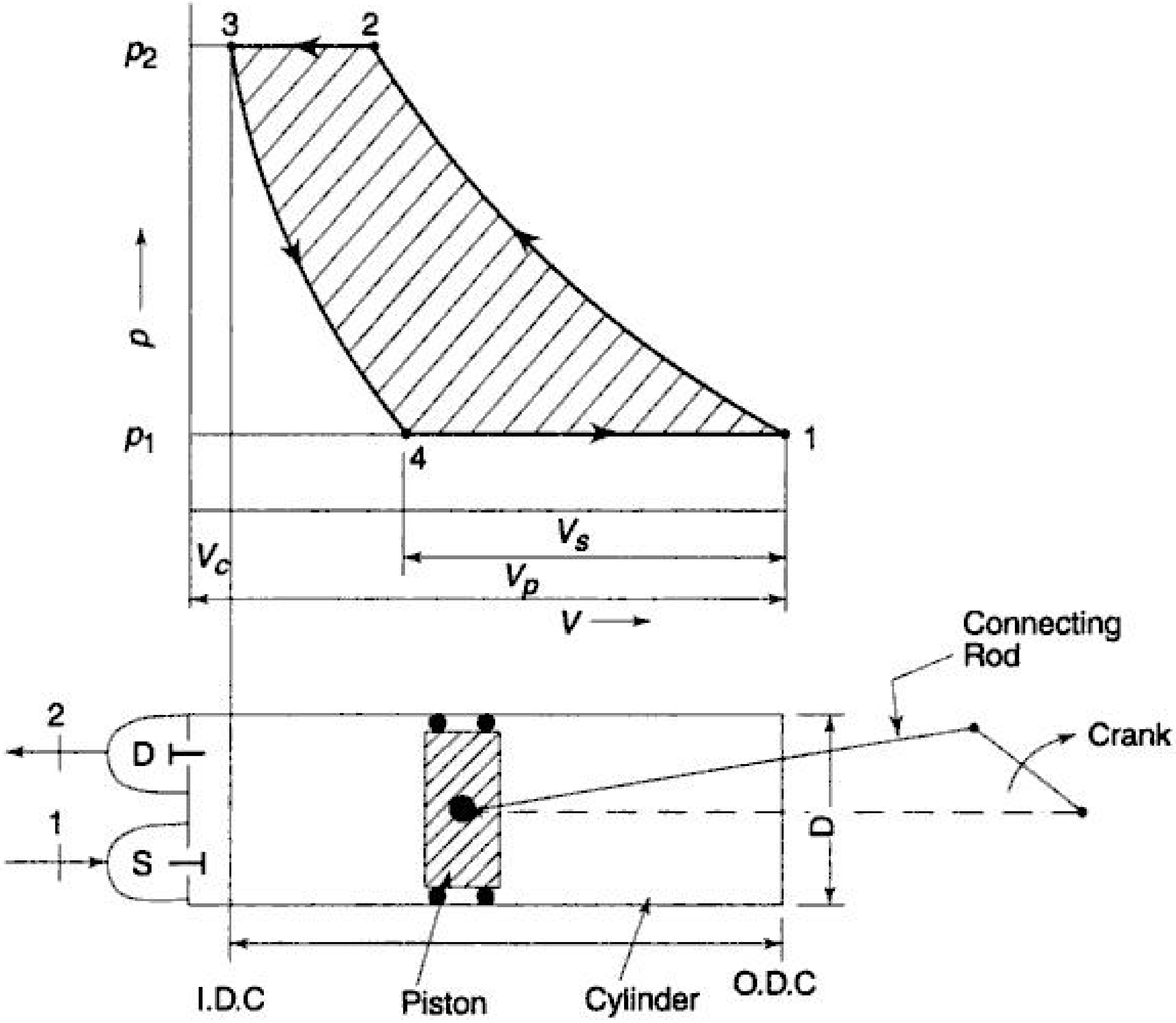

Fig. 6.1 Cylinder and Piston Mechanism and *p-V* diagram of a Reciprocating Compressor

where *D* is the *bore* or diameter and *L* is the *stroke*, i.e., the distance travelled by the piston between I.D.C. and O.D.C. of the cylinder. At 1, the suction valve closes as the piston moves inwards and the compression begins. At 2, the pressure in the cylinder is equal to the pressure at the *discharge flange* of the compressor. A further inward movement of the piston results in the pressure in the cylinder exceeding the condenser pressure. This opens the discharge valve D and the vapour from the cylinder flows into the condenser till the piston again reaches the IDC position. Gas equal to the clearance volume V_c remains in the cylinder and the cycle is repeated.

The work done for compression for the machine cycle is given by the cyclic integral of pdV. Hence

$$W = \oint p\mathrm{d}V = \int_1^2 p\mathrm{d}V + \int_2^3 p\mathrm{d}V + \int_3^4 p\mathrm{d}V + \int_4^1 p\mathrm{d}V$$

$$= \int_1^2 p\mathrm{d}V + p_2(V_3 - V_2) + \int_3^4 p\mathrm{d}V + p_1(V_1 - V_4)$$

$$= \text{Area 1-2-3-4}$$

It will be seen that this area is also expressed by the term $-\oint V\mathrm{d}p$. Hence

$$W = \oint_1^2 p\mathrm{d}V = -\oint_1^2 V\mathrm{d}p = m\oint_1^2 p\mathrm{d}v = -m\oint_1^2 v\mathrm{d}p$$

where m is the mass of the suction vapour. Thus, the specific work in a reciprocating compressor is given by

$$w = \oint_1^2 pdv = -\oint_1^2 v\mathrm{d}p \tag{6.1}$$

where 1 and 2 are the limits of integration from suction state 1 to the discharge state 2 as indicated in Fig. 6.1.

6.1.2 Work in Centrifugal Compressor[1]

In a steady-flow process, the gas enters the centrifugal compressor, passes over the blades in a centrifugal field and is subjected to momentum change, leaving finally, through a diffuser at the discharge pressure. From the steady-flow energy equation

$$q = (h_2 - h_1) + w \tag{6.2}$$

and from the combined First and Second Laws for reversible process,

$$q = \int_1^2 T\mathrm{d}s = \int_1^2 (\mathrm{d}h - v\mathrm{d}p) = (h_2 - h_1) - \int_1^2 v\mathrm{d}p \tag{6.3}$$

Comparing the two expressions, we have for work

$$w = -\int_1^2 v\mathrm{d}p \tag{6.4}$$

It is thus seen that the work of compression is the same for both reciprocating and centrifugal compressors and is given by the expression $-\int_1^2 v\mathrm{d}p$, integrated between the suction and discharge states. Equation (6.3), therefore, represents the energy equation for both compressors, viz.,

In centrifugal compressors, however, the process is nearly adiabatic. But an adiabatic compression process (no cooling) will normally be accompained with friction. Such a process can also be represented by the polytropic law and will be of the type (b), viz., *polytropic with friction* as represented by the line 1 – 2*f* in Figs 6.2 and 6.3.

An actual compression process in reciprocating compresors will be accompanied with both cooling and friction. Such a process can also be represented by the polytropic law with an appropriate value of the index of compression *n*. The discharge state after compression may be either to the left or to the right of the point 2*s*, depending on the degree of cooling and friction.

To compare the performance of a compressor, we define the following efficiencies

$$\textit{Isothermal efficiency, } \eta_T = \frac{\text{Isothermal work}}{\text{Actual work}}$$

$$\textit{Adiabatic efficiency, } \eta_a = \frac{\text{Isentropic work}}{\text{Actual work}}$$

Adiabatic effeciency is the most commonly used term.

The minimum work of compression, with cooling, is isothermal work. Often isothermal efficiency is, therefore, used to express the performance of reciprocating compressors which are invariably cooled.

The minimum work of compression, without cooling, is isentropic work. Adiabatic efficiency is, therefore, used to express the performance of centrifugal compressors in which it is not possible to arrange cooling during compression.

6.3 VOLUMETRIC EFFICIENCY OF RECIPROCATING COMPRESSORS

Volumetric efficiency η_v is the term defined in the case of positive displacement compressors to account for the difference in the displacement or swept volume V_p in-built in the compressor and volume V_s of the suction vapour sucked and pumped. It is expressed by the ratio

$$\eta_v = \frac{V_s}{V_p}$$

6.3.1 Clearance Volumetric Efficiency

The clearance or gap between the IDC position of the piston and cylinder head is necessary in reciprocating compressors to provide for thermal expansion and machining tolerances. A clearance of (0.005 L + 0.5) mm is normally provided. This space, together with the volume of the dead space between the cylinder head and valves, forms the clearance volume. The ratio of the clearance volume V_c to the swept volume V_p is called the *clearance factor C*, i.e.,

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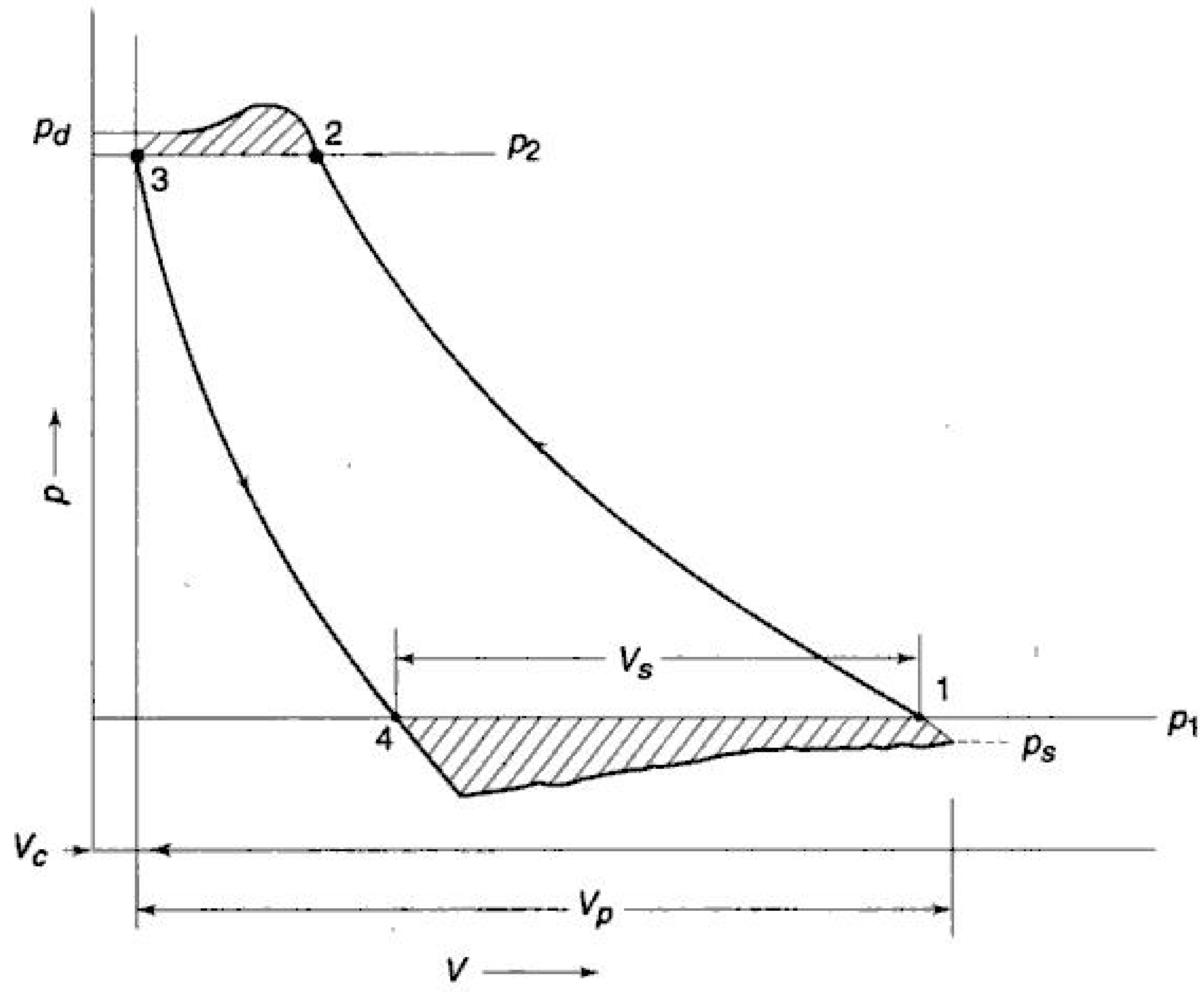

Fig. 6.5 Effect of Valve Pressure Drops

Considering the effect of pressure drop at the discharge valve as well, it can be shown that the expression for volumetric efficiency is

$$\eta_V = (1 + C)\left(\frac{p_s}{p_1}\right)^{1/n} - C\left(\frac{p_d}{p_1}\right)^{1/m} \tag{6.15}$$

6.3.5 Leakage Loss

The effect of leakage past piston rings and under the suction valve elements is normally accounted for by allowing 1.5 per cent leakage per unit of the compression ratio r, viz., p_2/p_1.

Old worn out compressors tend to have more leakage and hence they lose their cooling capacity.

6.3.6 Overall Volumetric Efficiency

Considering the effect of wire-drawing at the valves, polytropic compression, re-expansion, and leakage, we may write the expression for the overall or total volumetric efficiency as follows

$$\eta_V = (1 + C)\left(\frac{p_s}{p_1}\right)^{1/n} - C\left(\frac{p_d}{p_1}\right)^{1/m} - 0.015\, r \tag{6.16}$$

Some authors give slightly different expressions for the total volumetric efficiency.

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$$= \eta_V \frac{\dot{V}_p}{v_1} (h_1 - h_4) \tag{6.22}$$

The refrigerating effect varies very little with the suction pressure. The capacity, therefore, varies almost according to the mass flow rate of the refrigerant which in turn is proportional to η_V/v_1. The capacity decreases as p_0 decreases not only because of increase in v_1, but also because of decrease in η_V. The capacity $\dot{Q}_0 \to 0$ as $p_0 \to p_{0_{\min}}$. Thereafter it increases as shown in Fig. 6.6.

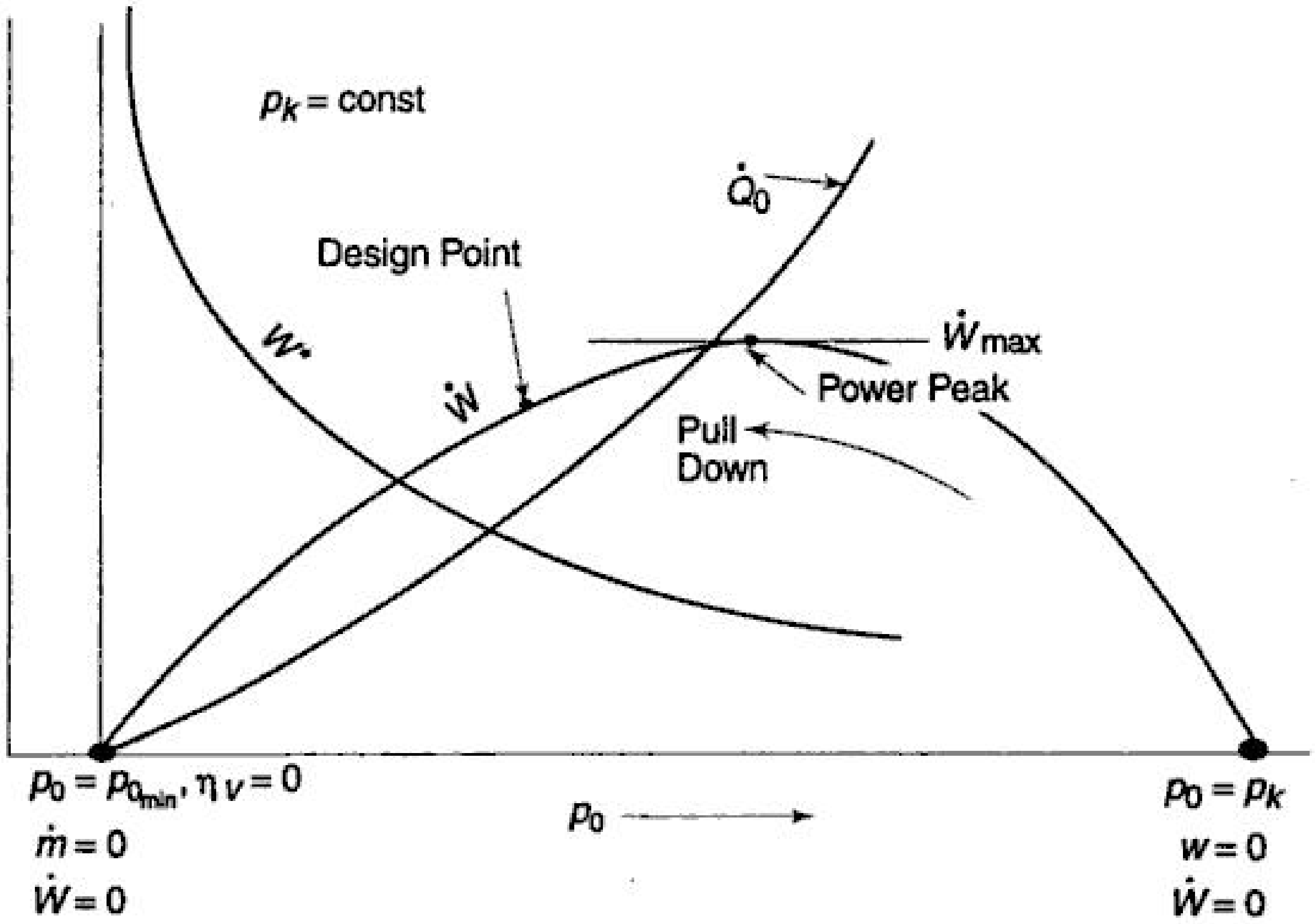

Fig. 6.6 Performance Characteristics of a Reciprocating Compressor as a Function of Evaporator Pressure

It is seen that the piston displacement per ton is given by

$$V^* = \frac{211}{q_0} \cdot \frac{v_1}{\eta_V} \text{ m}^3\text{/min (TR)}$$

Hence

$$\dot{Q}_0 = \left(\frac{\dot{V}_p}{V^*} \right) \propto \frac{1}{V^*} \tag{6.23}$$

Thus, the capacity is inversely proportional to the piston displacement per ton. It was shown in Sec. 3.5.1 that a drop of 5°C in the evaporator temperature increases V^* from 0.124 to 0.149 m^3/min/TR. The corresponding decrease in capacity was found to be 16.8 per cent.

The power consumption of the compressor is given by

$$\dot{W} = \dot{m} w$$

$$= \eta_V \frac{\dot{V}_p}{v_1} \cdot \left(\frac{h_2 - h_1}{\eta_m \eta_a} \right) \tag{6.24}$$

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***Note* ✍:** *With a decrease in the evaporator temperature and pressure, although the specific work increases, the total power consumption of compressor does not increase since the mass flow rate decreases. The power consumption at 4°C evaporator temperature is 4.45 kW (Capacity 7.5 TR). The power consumption at – 2°C evaporator temperature is 4.2 kW only (Capacity 5.8 TR).*

6.6.1 Calorimetric Method of Measuring Refrigerating Capacity of Small Compressors

Figure 6.7 illustrates the construction and schematic arrangement of a calorimeter used to measure the refrigerating capacity of small compressors.

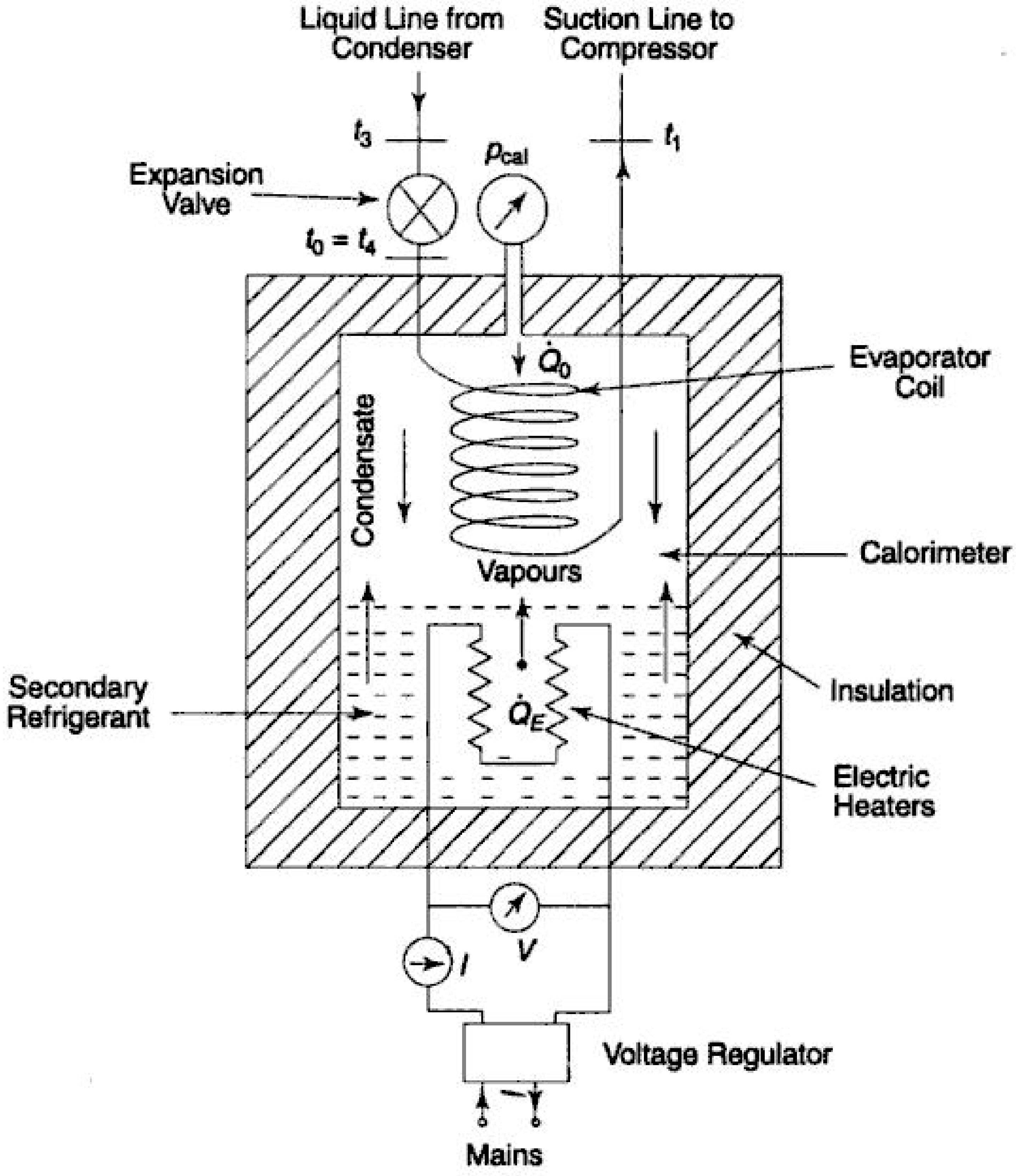

Fig. 6.7 Calorimeter for Measuring Refrigerating Capacity of Small Compressors

The calorimeter encloses the evaporator in the form of a coil, and a set of electric heaters dipped below the level of the liquid column of a secondary refrigerant. When the heaters are on, vapours of secondary refrigerant rise above. They get condensed by coming in contact with the cold surface of the evaporator coil. The condensate returns back to the sump of liquid below.

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minimize the pressure drop, their weight should be small. Reed valves with varying clamping arrangements are commonly employed in refrigeration.

The lubrication systems in refrigerant compressors vary. Both *splash* and *forced feed* systems are employed. *Shaft seals* are used in open-type compressors. Normally *stationary* seals with *bellows* are employed. Recently, *rotary* synthetic seals, tightly fitted to the shaft in a carbon nose, have gained popularity. They are much less expensive.

6.9 ROTARY COMPRESSORS

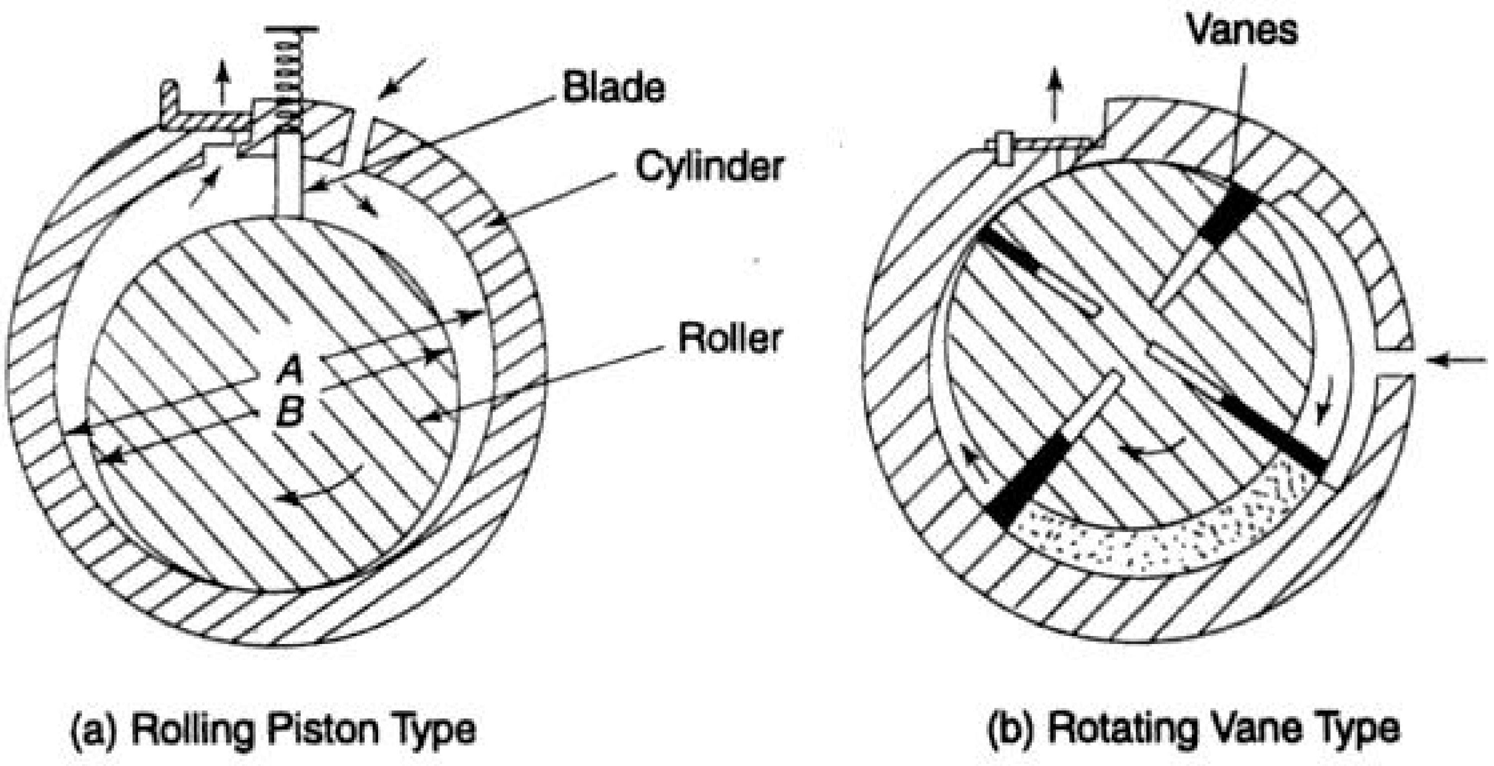
Rotary compressors are positive displacement, direct-drive machines. There are essentially two designs of this compressor:

(i) Rolling piston type.

(ii) Rotating vane type.

In the rolling piston type, shown in Fig. 6.8(a) the roller is mounted on an eccentric shaft with a single blade, which is always in contact with the roller by means of a spring. The theoretical piston displacement is

$$V_p = \frac{\pi H(A^2 - B^2)}{4} \tag{6.25}$$



(a) Rolling Piston Type

(b) Rotating Vane Type

Fig. 6.8 Rotary Compressor

where *A* and *B* are respectively the diameters of the cylinder and rolling piston and *H* is the height of the cylinder.

In the rotating vane type, as shown in Fig. 6.8 (b) with four vanes, the rotor is concentric with the shaft. The vanes slide within the rotor but keep contact with the cylinder. The assembly of rotor and the vanes is off-centre with respect to the cylinder.

In both designs, the whole assembly is enclosed in a housing (not shown in the figures), filled with oil and remains submerged in oil. An *oil film* forms the seal

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Flow through Inlet Casing (Process i-1)

In this process the fluid is accelerated, but there is no energy transfer. Hence, the stagnation enthalpy remains constant but there is a drop in the static pressure and enthalpy.

$$h_{io} = h_i + \frac{C_i^2}{2} = h_1 + \frac{C_1^2}{2} = h_{10} \tag{6.26}$$

(Subscript 0 refers to stagnation state.)

Flow through Impeller (Process 1-2)

In this, work is done by the impeller. Energy is transferred to the fluid and the velocity, pressure and enthalpy are increased. The *energy equation* for the process is

$$w = (h_2 - h_1) + \frac{C_2^2 - C_1^2}{2} = h_{20} - h_{10} \tag{6.27}$$

Flow through Diffuser and Volute Casing (Processes 2-3 and 3-4)

There is no energy transfer, but kinetic energy is converted into static enthalpy in both processes.

Diffuser: $$h_{20} = h_2 + \frac{C_2^2}{2} = h_3 + \frac{C_3^2}{2} = h_{30} \tag{6.28}$$

Volute casing: $$h_{30} = h_3 + \frac{C_3^2}{2} = h_4 + \frac{C_4^2}{2} = h_{40} \tag{6.29}$$

Combining the two equations for the process from 2 to 4

$$h_{20} = h_2 + \frac{C_2^2}{2} = h_4 + \frac{C_4^2}{2} = h_{40} \tag{6.30}$$

and further combining with Eq. (6.27), we obtain

$$w = h_{20} - h_{10} = h_{40} - h_{10} = \Delta h_0 = (h_4 - h_1) + \frac{C_4^2 - C_1^2}{2} \tag{6.31}$$

which represents the overall energy balance for a centrifugal compressor stage.

6.11.2 Application of Momentum Equation to a Centrifugal Stage

Figure 6.12 represents the impeller of a stage of a centrifugal compressor and Fig. 6.13 shows the velocity triangles at the impeller inlet and outlet. In Fig. 6.12, C is the fluid velocity, C_{rel} is the relative velocity of the fluid with respect to the impeller and u is the impeller tip *speed*. C_u and C_r are the tangential and radial components of the fluid velocity as shown in Fig. 6.13.

Now the torque I is given by the rate of change of angular momentum of the fluid. Thus, per unit mass of the fluid, we have

$$I = C_{u_2} r_2 - C_{u_1} r_1 \tag{6.32}$$

where r_2 and r_1 are the radii of the impeller at the outlet and inlet respectively.

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From Fig. 6.16, the actual work done is

$$\Delta h_0 = h_{40} - h_{10} = (h_4 + C_4^2/2) - h_{10} \tag{6.41}$$

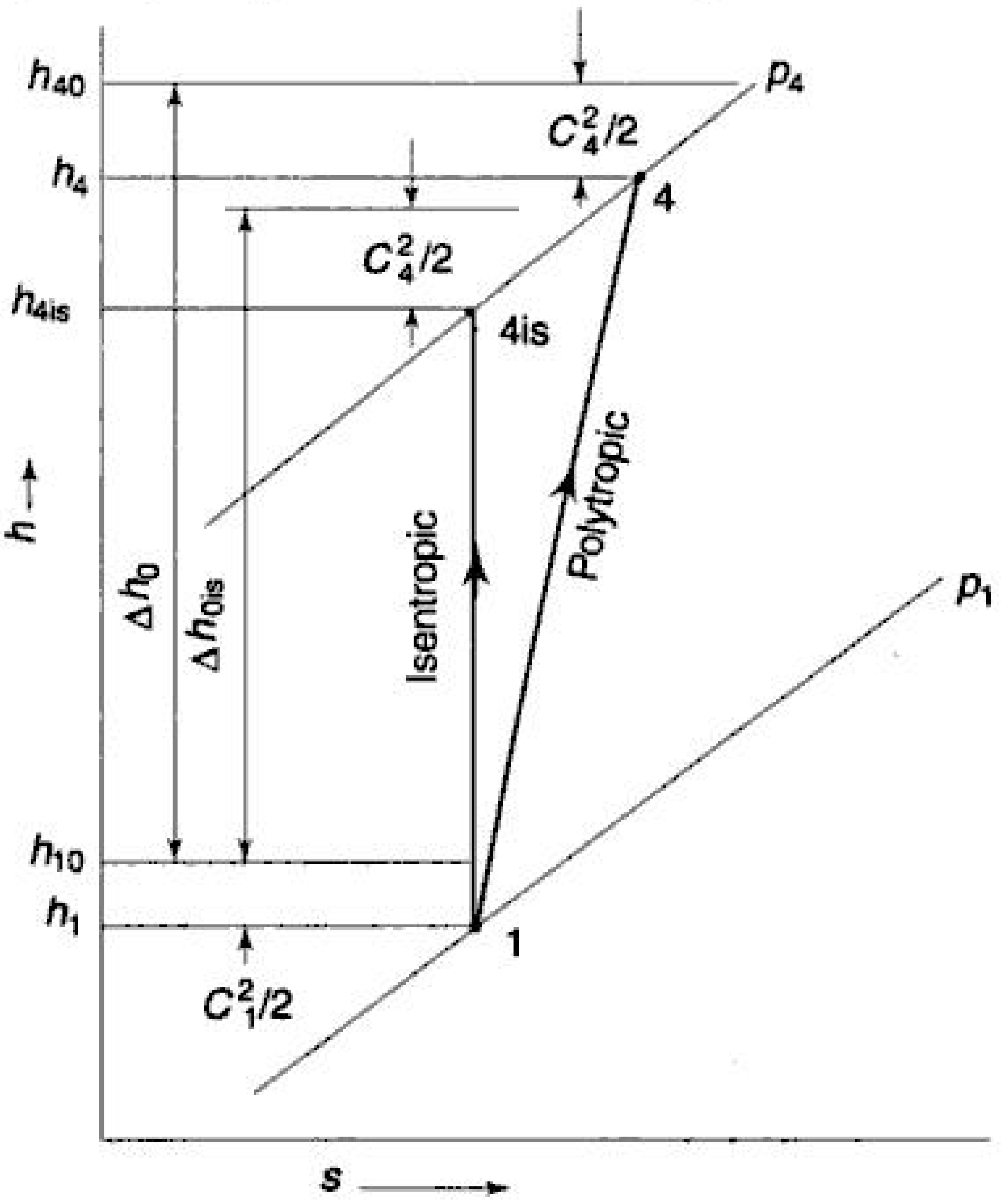

Fig. 6.16 Comparison of Isentropic and Polytropic Compression Processes on a Mollier Diagram

Also, 4*is* is the point after isentorpic compression to the same pressure p_4. Considering same kinetic energies at 4*is* and 4, we have for work after isentropic compression

$$\Delta h_{0is} = (h_{4is} + C_4^2/2) - h_{10} \tag{6.42}$$

Enthalpy h_{4is} can be found from the isentropic relationslip.

6.11.5 Pressure Ratio Developed in a Centrifugal Stage

Combining steady-flow energy Eq. (6.31) and momentum Eq. (6.33) we have

$$w = \Delta h_0 = C_{u_2} u_2 - C_{u_1} u_1 = (h_4 - h_1) + \frac{C_4^2 - C_1^2}{2} \tag{6.43}$$

whence

$$h_4 - h_1 = w - \frac{C_4^2 - C_1^2}{2}$$

$$= (C_{u_2} u_2 - C_{u_1} u_1) - \frac{C_4^2 - C_1^2}{2} \tag{6.44a}$$

For the isentropic process, since $dh = v dp$, we can write for the enthalpy change in terms of the pressure ratio p_4/p_1 developed in the compressor stage

$$h_4 - h_1 = \int_1^4 v\,dp = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[(p_4/p_1)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{6.44b}$$

Combining Eqs (6.44a) and (6.44b), we have

$$h_4 - h_1 = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= (C_{u_2} u_2 - C_{u_1} u_1) - \frac{C_4^2 - C_1^2}{2} \tag{6.45}$$

whence we get for the pressure ratio developed

$$r = \frac{p_4}{p_1} = \left\{\frac{\gamma - 1}{\gamma} \frac{1}{p_1 v_1}\left[(C_{u_2} u_2 - C_{u_1} u_1) - \frac{C_4^2 - C_1^2}{2}\right] + 1\right\}^{\frac{\gamma}{\gamma - 1}} \tag{6.46}$$

The pressure ratios developed separately in the impeller and diffuser (including volute casing) respectively are similarly given by,

$$h_2 - h_1 = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= \frac{u_2^2 - u_1^2}{2} + \frac{C_{\text{rel}_1}^2 - C_{\text{rel}_2}^2}{2} \tag{6.47}$$

$$h_4 - h_2 = \frac{\gamma}{\gamma - 1} p_2 v_2 \left[\left(\frac{p_4}{p_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = \frac{C_2^2 - C_4^2}{2} \tag{6.48}$$

6.11.6 Compressor with Radial Blades

A radial blade compressor is simple in construction. In this, the outlet blade angle β_2 is 90°. It can be seen from the outlet velocity triangle in Fig. 6.17 that $C_{u_2} = u_2$ for a radial blade compressor. Then, from Eqs (6.43) and (6.44a),

$$w = \Delta h_0 = u_2{}^2 - C_{u_1} u_1 = (h_4 - h_1) + \frac{C_4^2 - C_1^2}{2} \tag{6.49}$$

Under assumptions of no prewhirl and negligible kinetic energies at the inlet and outlet, we obtain

$$w = \Delta h_0 = h_4 - h_1 = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = C_{u_2} u_2 = u_2^2 \tag{6.50}$$

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Expressing the radial velocity and head developed in dimensionless from we have

Flow coefficient, $\phi = \dfrac{C_r}{u}$

Head coefficient, $\mu = \dfrac{\Delta h_0}{u_2^2} = \dfrac{C_{u_2}}{u_2}$

and from Eq. (6.54)

$$\mu = 1 - \phi_2 \cot \beta_2 \tag{6.55}$$

This relation also shows that, for radial blades and with no pre-whirl, μ is equal to unity, i.e., the head developed is equal to u_2^2.

6.13.1 Surging

Consider A in Fig. 6.18 as the point of operation at full load. When the refrigeration load decreases, the point of operation shifts to the left until point B of maximum head is reached. If the load continues to decrease to the left of B, say to C, the pressure ratio developed by the compressor becomes less than the ratio required between the condenser and evaporator pressure, viz.,

$$\frac{p_4}{p_1} < \frac{p_k}{p_0}$$

Hence some gas flows back from the condenser to the evaporator, thus increasing the evaporator pressure and decreasing p_k/p_0. The point of operation suddenly shifts to A. As the refrigeration load is still less, the cycle will repeat itself. This phenomenon of reversal of flow in centrifugal compressors is called *surging*. It occurs when the load decreases to below 35 per cent of the rated capacity and causes severe stress conditions in the compressor as a result of *hunting*. Figures 6.15 and 6.18 show the *surge line* drawn through the points of maximum head at different speeds.

6.13.2 Capacity Control of Centrifugal compressors

Centrifugal compressors require high tip speeds u_2 to develop the necessary pressure ratio. The high tip speed is achieved by employing either a large diameter impeller or high rpm or both. Because of large u_2, the velocities in general including the flow velocity C_r are high. Also, there must be a reasonable width of the shrouds to minimize friction and achieve high efficiency. Thus, because of the sufficiently large flow area (diameter D and width of shrouds b) required and large flow velocity, the satisfactory volume that can be handled by a centrifugal compressor is about 30–60 cubic metres per minute. A single centrifugal compressor, therefore, can be designed for a minimum capacity approximately of the order of 250 TR with R 11 and 150 TR with R 113 for the purpose of air conditioning. These are the two common refrigerants which have been used with centrifugal compressors in the past.

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$$= 18.5 \text{ kJ/kg}$$

For actual compression

$$w = \Delta h_0 = \frac{\Delta h_{0_{\text{isen}}}}{\eta_P}$$

$$= \frac{18.4}{0.773} = 24.1 \text{ kJ/kg}$$

(a) *For backward curved blades*

$$\phi_2 = (1 - \sin \beta_2) \tan \beta_2$$

$$= (1 - \sin 32°) \tan 32° = 0.294$$

$$\mu = (1 - \phi_2 \cot \beta_2) = \frac{\Delta h_0}{u_2^2}$$

$$= (1 - 0.294 \times \cot 32°) = 0.528$$

Hence

$$u_2^2 = \frac{\Delta h_0}{\mu} = \frac{24.1 \times 10^3}{0.528} = 4.564 \times 10^4 \text{ m}^2/\text{s}^2$$

$$u_2 = 213.2 \text{ m/s}$$

The required diameter of the impeller is

$$D_2 = \frac{60 u_2}{\pi N} = \frac{(60)(213.2)}{(3600)} = 1.131 \text{ m}$$

(b) *For radial blades*

$$\beta_2 = 90°, C_{u_2} = u_2, \Delta h_0 = C_{u_2} u_2 = u_2^2, \mu = 1$$

Hence

$$u_2 = \sqrt{\Delta h_0} = \sqrt{24.1 \times 10^3} = 155.2 \text{ m/s}$$

$$D_2 = \frac{60\,(155.2)}{3600\,\pi} = 0.824 \text{ m}$$

(c) *Enthalpy of liquid from condenser at 36°C*

$$h_f = 231.4 \text{ kJ/kg}$$

Refrigerating effect

$$q_0 = 392.5 - 231.4 = 161.1 \text{ kJ/kg}$$

Minimum mass flow rate

$$\dot{m} = \frac{40/60}{0.308} = 2.1645 \text{ kg/s}$$

Radial flow velocity

$$C_{r_2} = \phi_2 u_2 = 0.294\ (215.6) = 63.39 \text{ m/s}$$

Impeller diameter

$$D_2 = \frac{60\, u_2}{\pi N} = \frac{60\,(215.6)}{\pi (2950)} = 1.4 \text{ m}$$

Width of shrouds at outlet

$$b_2 = \frac{\dot{m}\, v_2}{\pi D_2\, C_{r_2}} = \frac{\dot{Q}_0}{q_0} \frac{v_2}{\pi D_2\, C_{r_2}} = \frac{600 \times 3.5167\ (0.0168)}{(157.5)\,\pi\,(1.4)\ (63.39)}$$

$$= 0.81 \times 10^{-3} \text{ m } (0.81 \text{ mm})$$

From outlet velocity triangle, absolute velocity $C_2 = 141.8$ m/s
Accoustic velocity and Mach number at outlet

$$a_2 = \sqrt{\gamma R T_2} = \sqrt{1.16\ (96.13)\ (273 + 73)} = 184.7 \text{ m/s}$$

$$M_2 = \frac{C_2}{a_2} = \frac{141.8}{184.7} = 0.77$$

Note: *Width of shrouds of 0.81 mm is very small. The efficiency of compressor will be very poor, not 0.85. The solution is to use tunnel-diffuser passages. It will be interesting to see what the demensions will be with R134a.*

6.15 COMPARISON OF PERFORMANCE OF RECIPROCATING AND CENTRIFUGAL COMPRESSORS

The advantages of the centrifugal compressor over the reciprocating compressor are high efficiency over a large range of load and a large volume of the suction vapour and hence a larger capacity for its size.

The centrifugal compressor has also many other advantageous features. The most important is the *flat head-capacity characteristic* as compared to that of a reciprocating compressor as shown in Fig. 6.19 for typical compressors at a constant condensing temperature of 38°C and constant rpm. It is seen that the variation in the evaporator temperature is only 2 to 7.5°C for a load variation of 100 to 240 TR for a centrifugal compressor, whereas it is – 11 to 6°C for the same load variation for a reciprocating compressor.

Another advantageous feature is the *non-overloading characteristic* as shown in Fig. 6.20. It is seen that for a centrifugal machine, there is a decrease in the power requirement with an increase in the condensing temperature. This is due to the fact that the flow rate (refrigerating capacity) decreases as the head required increases (Fig. 6.18), while the power consumption represents the product of the two quantities. Thus there is no overloading of the motor with increasing condenser temperature. This feature is accompanied by a rapid fall in the flow rate and hence capacity as shown in Fig. 6.21. The power requirement decreases although the horsepower per ton increases. For reciprocating compressors, there is a small increase in the power requirement with an increase in the condensing temperature with consequent overloading of the motor. This is accompanied with a very small decrease in the refrigerating capacity. This latter aspect of a reciprocating compressor is, however, an advantage so that the capacity of a reciprocating machine is not affected much by an increase in the condensing temperature followed by adverse ambient conditions.

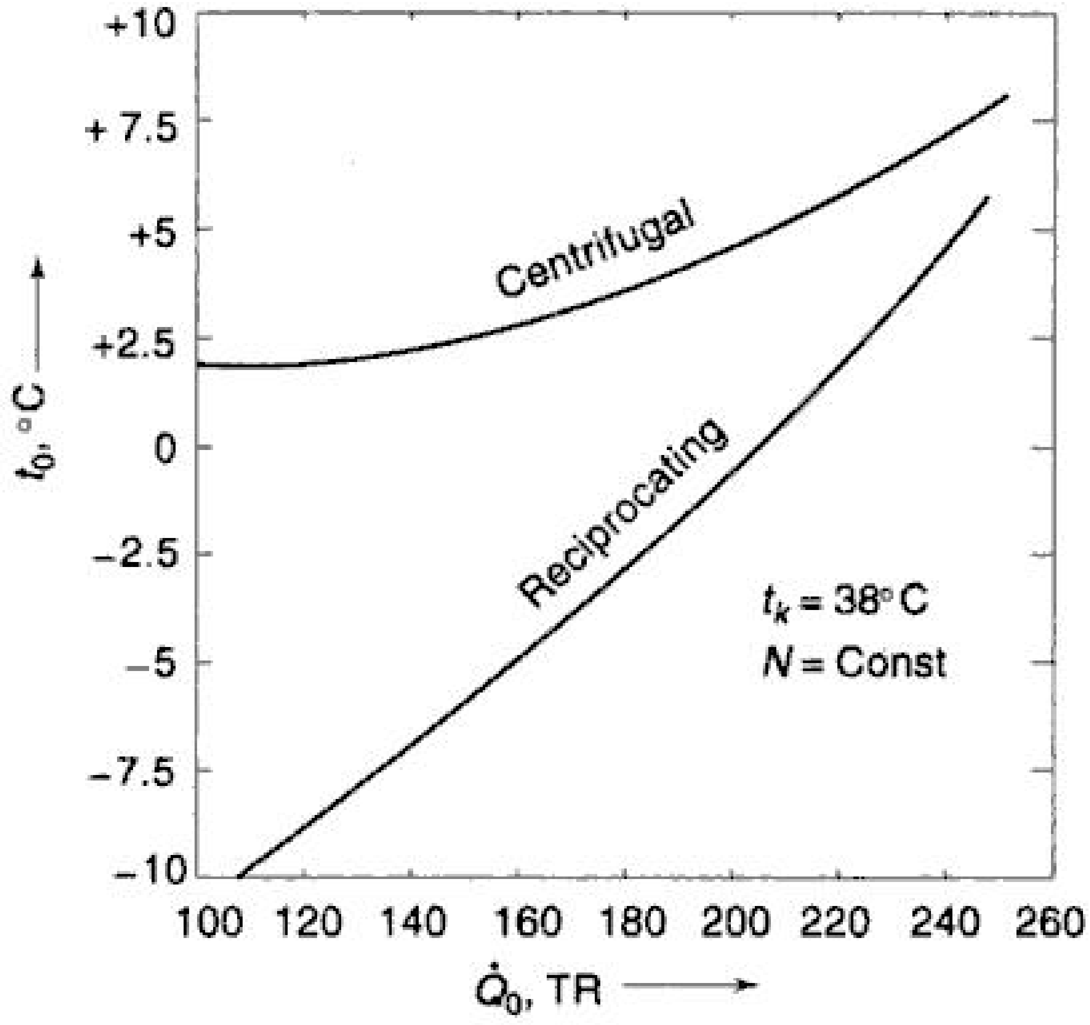

Fig. 6.19 Nature of Capacity Variation with Evaporator Temperature t_0 for Centrifugal and Reciprocating Compressors

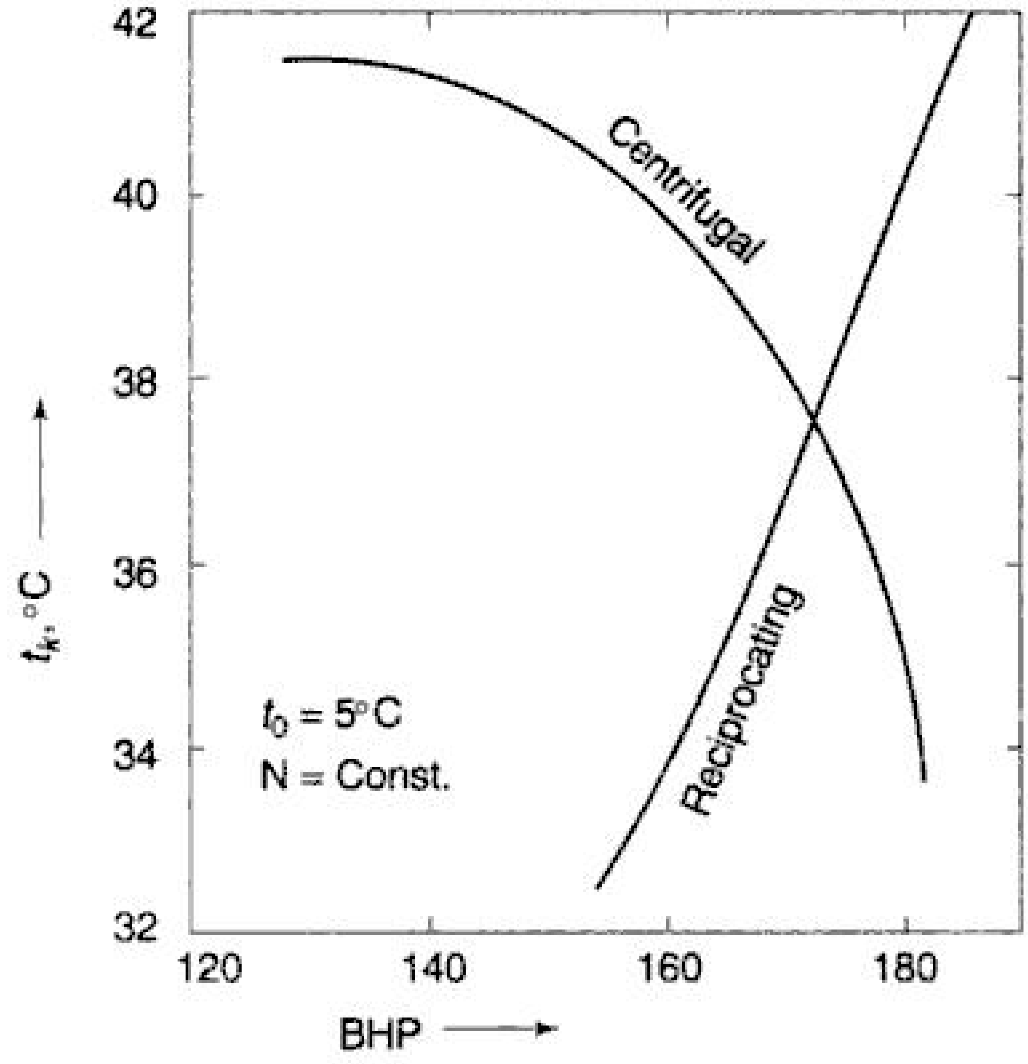

Fig. 6.20 Non-overloading Characteristic of Centrifugal Compressor as Compared to that of Reciprocating Compressor

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Chapter 7

Condensers

7.1 HEAT REJECTION RATIO

It is ultimately in the condenser that heat is rejected in a vapour-compression refrigeration machine. The vapour at discharge from the compressor is superheated. Desuperheating of the vapour takes place in the discharge line and in the first few coils of the condenser. It is followed by the condensation of the vapour at the saturated discharge temperature or condensing temperature t_k. In some condensers, subcooling may also take place near the bottom where there is only liquid. However, the sensible heat of the desuperheating and subcooling processes is very small as compared to the latent heat of the condensation process.

The loading on the condenser per unit of refrigeration is called the *heat rejection ratio*.

Since

$$Q_k = Q_0 + W$$

we have for the heat rejection ratio

$$\frac{Q_k}{Q_0} = 1 + \frac{1}{\mathcal{E}_c} \tag{7.1}$$

Thus the ratio depends on the COP which in turn depends on the condenser and evaporator temperatures. In actual air-conditioning equipment for R 12 and R 22, operating at $t_k = 40°\text{C}$ and $t_0 = 5°\text{C}$, the heat rejection ratio is approximately 1.25 to 1.35.

7.2 TYPES OF CONDENSERS

The type of a condenser is generally characterized by the cooling medium used. Thus there are three types of condensers:

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7.3.1 Desuperheating

In condenser design, normally no separate calculations are made for the desuperheating section. Though the heat transfer coefficient is lower for this section, the temperature differential is greater as compared to that of the condensing section. The two factors tend to cancel the effects of each other. Thus, the practice is to take the refrigerant-side temperature equal to the condensing temperature and the heat-transfer coefficient equal to the condensing coefficient for the whole length of the condenser.

7.3.2 Condensing Heat Transfer Coefficient

Refrigerants are usually good wetting agents. In condensation, therefore, they follow a *film-wise* pattern. Nusselt, using his laminar liquid film theory, derived the following expression for the overall coefficient of heat transfer for condensation on a vertical surface of height x

$$h = 0.943 \left[\frac{k_f^3 \rho_f (\rho_f - \rho_g) g h_{fg}}{\mu_f x \Delta t} \right]^{1/4} \tag{7.4}$$

where k_f, ρ_f and μ_f are the thermal conductivity, density and viscosity of the condensate film. These are evaluated at the average film temperature $t_f = \frac{1}{2}(t_k + t_w)$. And Δt is the temperature differential $(t_k - t_w)$ where t_w is the tube-wall temperature. Density ρ_g of the vapour is very small compared to ρ_f and can be neglected.

Note that the condensate film acts as a resistance to heat transfer.

Outside Horizontal Tubes

A similar expression for the average coefficient of heat transfer for vapour condensing on the outside of horizontal tubes of diameter D_0, as in shell and tube condensers, is

$$h_0 = 0.725 \left[\frac{k_f^3 \rho_f^2 g h_{fg}}{N D_0 \mu_f \Delta t} \right]^{1/4} \tag{7.5}$$

where D_0 is the outside diameter of tubes and N is the number of tubes in a vertical row.

Equations (7.4) and (7.5) show that the heat-transfer coefficient increases with thermal conductivity. A large density results in a thinner growth of the condensate film. The heat-transfer coefficient, therefore, increases with the density. Similarly, a high latent heat implies a lower condensation rate and hence a thinner film and higher heat-transfer coefficient. On the other hand, a higher viscosity results in a thicker film and hence a lower heat transfer coefficient. Similarly, a higher temperature differential results in higher condensation rate, a thicker film and a lower heat-transfer coefficient.

Further, a large vertical tube length x or a horizontal tube diameter D_0 cause a thicker growth of film before it drains of from the tube and hence a lower average heat-transfer coefficient. Finally, the condensate film formed on the lower tubes in a vertical bank is thicker, giving a lower average value of the coefficient.

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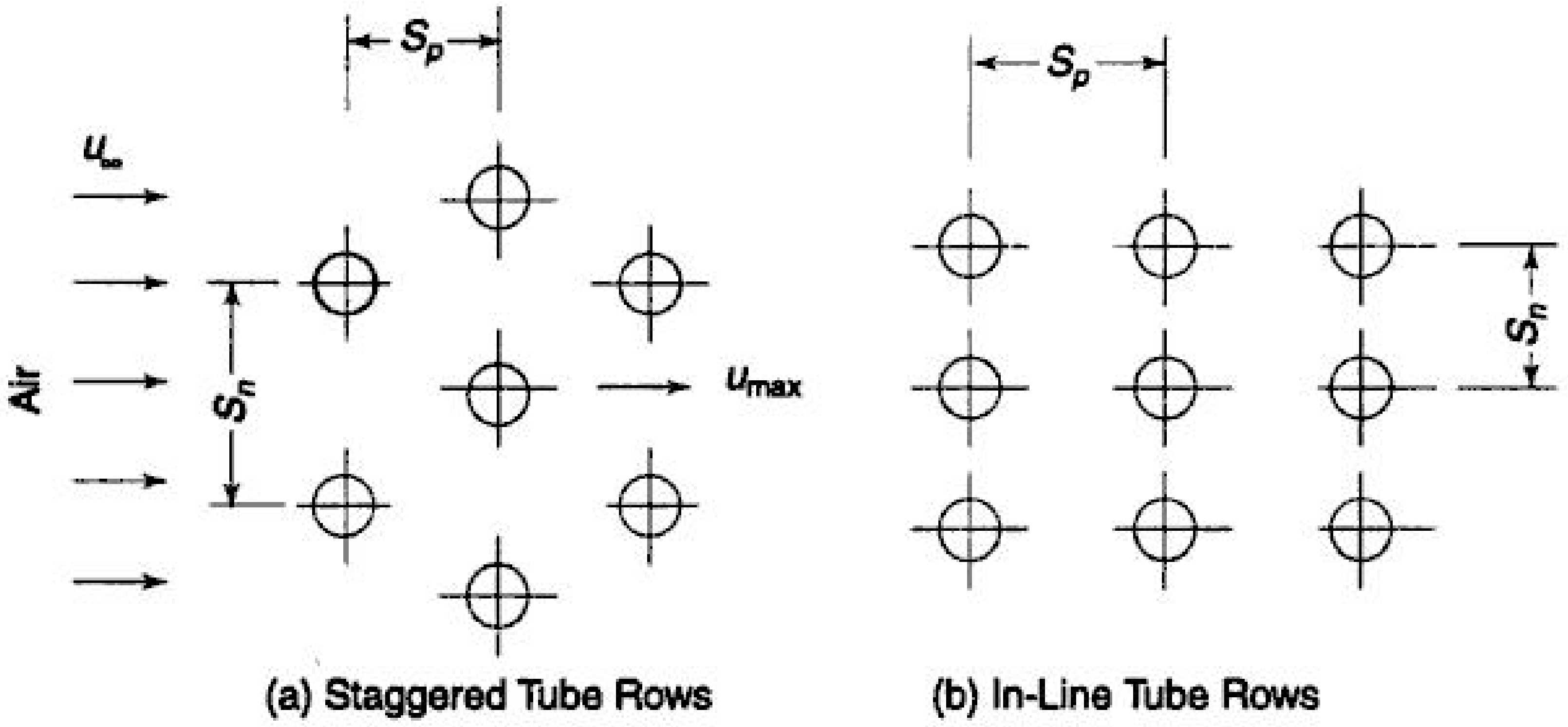

(a) Staggered Tube Rows (b) In-Line Tube Rows

Fig. 7.5 Tube Arrangements

In the case of tube banks, u_∞ is the face velocity (FV) measured at a distance from the tube

$$u_\infty = \frac{\text{Volume flow rate}}{\text{Face area}} = \frac{\dot{Q}_v}{(FA)} = (FV)$$

The Reynolds number is, however, based on the maximum velocity occurring in the tube bank which is

$$u_{\max} = u_\infty \frac{S_n}{S_n - D}$$

For the case of forced convection of air over finned tubes, Table 20.1 gives the values of h_{air} or f_g as a function of (FV). These values can be correlated by the relation

$$h_{\text{air}} = 38\,(FV)^{0.5} \tag{7.17}$$

For the case of natural convection of air on horizontal tubes, the following simplified relations may be used.

For laminar flow, $10^4 < \text{Gr}_f\,\text{Pr}_f < 10^9$

$$h_{\text{air}} = 1.32\left(\frac{\Delta T}{D_0}\right)^{1/4} \tag{7.18}$$

For turbulent flow, $\text{Gr}_f\,\text{Pr}_f > 10^9$

$$h_{\text{air}} = 1.24\,(\Delta T)^{1/3} \tag{7.19}$$

In these equations Gr_f is the Grashoff's number, a characteristic of natural convection, expressed by

$$\text{Gr}_f = \frac{g\beta\Delta T D^3}{\nu^2} \tag{7.20}$$

where $\beta = \frac{1}{T_f}$. T_f is the mean temperature of the film and ν is the kinematic viscosity.

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$$\text{Re} = \frac{Du_w\rho}{\mu} = \frac{(0.0137)\,(1.644)\,(1000)}{7.65 \times 10^{-4}} = 24{,}070$$

Water-side coefficient

$$\text{Nu} = \frac{h_i D}{k} = 0.023\,(\text{Re})^{0.8}\,(\text{Pr})^{0.4} = 141.4$$

$$h_i = 141.4 \times \frac{0.623}{0.01} = 6432 \text{ Wm}^{-2}\text{ K}^{-1}$$

Area Calculations

Consider 1m length of tube.

Fin surface area

$$A_f = 748\,\pi\left[\frac{2(0.0179^2 - 0.0159^2)}{4} + (0.0179)\,0.001\right]$$

$$= 0.12144 \text{ m}^2$$

Bare tube surface area

$$A_b = \pi\,(0.0159)\,(1 - 0.748)$$

$$= 0.01258 \text{ m}^2$$

Total tube surface area

$$A_t = A_f + A_b = 0.0134 \text{ m}^2$$

Outside and inside tube surface areas

$$A_0 = \pi\,D_0 = 0.04993 \text{ m}^2$$

$$A_i = \pi\,D_i = 0.04302 \text{ m}^2$$

Outside to inside surface area of the tube

$$\frac{A_0}{A_i} = \frac{D_0}{D_i} = \frac{1.59}{1.37} = 1.1607$$

Extended surface to outside and inside tube surface area ratios

$$A_t/A_0 = \frac{0.0134}{0.04993} = 2.684$$

$$A_t/A_i = \frac{0.0134}{0.04302} = 3.046$$

Fin efficiency as calculated (circular fins)

$$\eta_f = 0.992$$

Overall heat transfer coefficient based on total extended surface side area A_t

$$\frac{1}{U_t} = \frac{1}{h_0}\,\frac{A_t}{(\eta_f\,A_f + A_b)} + \frac{1}{h_f}\,\frac{A_t}{A_i} + \frac{1}{h_i}\,\frac{A_t}{A_i}$$

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At this temperature for air

$$\mu = 2 \times 10^{-5} \text{ kg. m}^{-1} \text{ s}^{-1}$$

$$k = 0.03 \text{ Wm}^{-1} \text{ K}^{-1}$$

$$\mathbf{Re} = \frac{D_E \rho u}{\mu} = \frac{0.65 \times 1.2 \times 6}{2 \times 10^{-5}} = 234{,}000$$

$$\mathbf{Nu} = 0.193\ (0.71)^{1/3}\ (234{,}000)^{0.618} = 358$$

$$h_0 = \frac{k}{D_E}\ \mathbf{Nu} = \frac{0.03}{0.65}\ (358) = 16.4 \text{ Wm}^{-2} \text{ K}^{-1}$$

Overall heat transfer coefficient

Neglecting the metal-wall resistance, the overall heat-transfer coefficient based on the total fin-side surface area is given by

$$\frac{1}{U_t A_t} = \frac{1}{h_i A_i} + \frac{1}{h_0 A_t \eta_f}$$

$$\frac{1}{U_t} = \frac{1}{h_i} \frac{A_t}{A_0} \frac{A_0}{A_i} + \frac{1}{h_0} \frac{1}{\eta_f}$$

$$= \frac{1}{2643}\ (20) \left(\frac{1.27}{1.12}\right) + \frac{1}{16.4\ (0.9)}$$

$$\Rightarrow \qquad U_t = 13.1 \text{ Wm}^{-2} \text{ K}^{-1}$$

Finned surface area

Temperature rise of air

$$\Delta t_a = \frac{\dot{Q}_k}{\dot{Q}_v\, \rho\, C_p} = \frac{22.2 \times 60}{120 \times 1.2 \times 1.005} = 9.25°\text{C}$$

Air leaving temperature

$$t_{a_2} = 40 + 9.25 = 49.25°\text{C}$$

Log mean temperature difference

$$\Delta t_m = \frac{(55 - 40) - (55 - 49.25)}{\ln \dfrac{55 - 40}{55 - 49.25}} = 9.62°\text{C}$$

Finned surface area

$$A_t = \frac{\dot{Q}_k}{U_t\, \Delta t_m} = \frac{22.2 \times 10^3}{(13.1)\ (9.62)} = 176 \text{ m}^2$$

Note ✍: *Temperature drop through films may be checked.*

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Chapter 8

Expansion Devices

8.1 TYPES OF EXPANSION DEVICES

An *expansion device* in a refrigeration system normally serves two purposes. One is the thermodynamic function of expanding the liquid refrigerant from the condenser pressure to the evaporator pressure. The other is the control function which may involve the supply of the liquid to the evaporator at the rate at which it is evaporated. The latter has an important role and determines the efficiency with which the evaporator surface is utilized.

An expansion device is essentially a restriction offering resistance to flow so that the pressure drops, resulting in a throttling process. Basically there are two types of expansion devices. These are:

(i) Variable-restriction type.
(ii) Constant-restriction type.

In the variable-restriction type, the extent of opening or area of flow keeps on changing depending on the type of control. There are two common types of such control devices, viz., the *automatic expansion valve* and the *thermostatic expansion valve*.

In addition, there are the float-valves which are also variable restriction type devices. The float valves again are of two types: The *high-side float* and the *low-side float*.

The high-side float maintains the liquid at a constant level in the condenser and the low-side float maintains the liquid at a constant level in the evaporator.

The constant-restriction type device is the *capillary tube* which is merely a long tube with a narrow diameter bore.

The capillary and bulb are generally filled with the same liquid refrigerant (power fluid) as in the refrigeration system. Force F_0' corresponds to the saturation pressure p_0' of the refrigerant at temperature t_0' at the exit end of the evaporator as shown in Fig. 8.4.

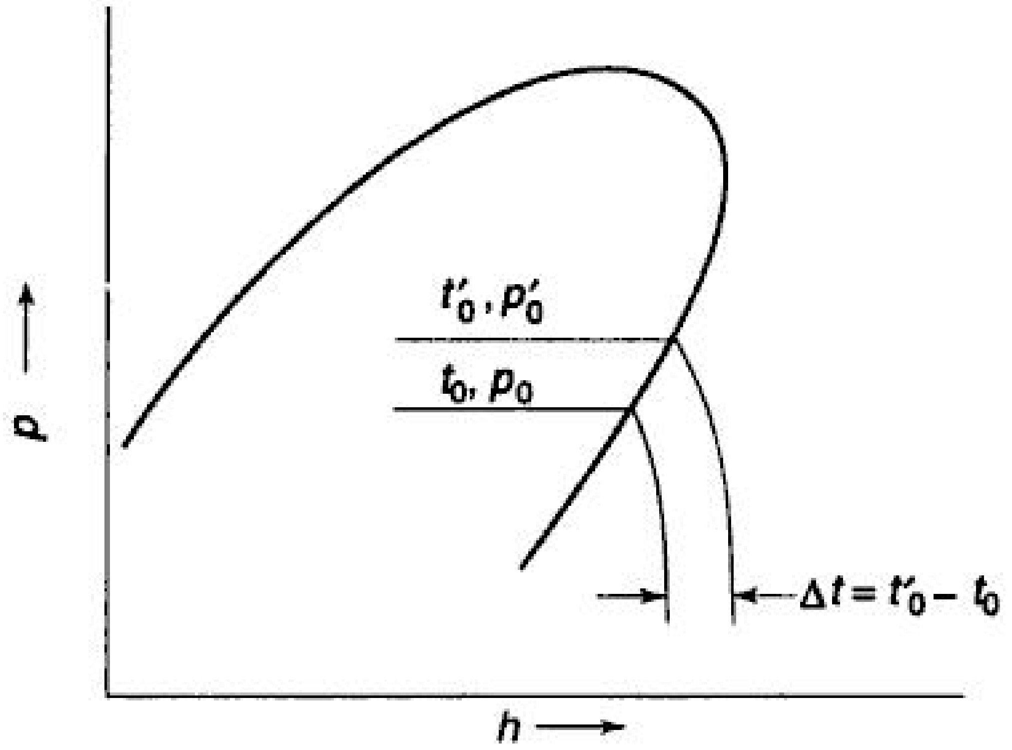

Fig. 8.4 Degree of Superheat with Thermostatic-Expansion Valve

During off-cycle, the temperatures at the evaporator inlet and outlet are the same so that $(F_0' - F_0)$ is zero. Since the follow-up spring is adjusted with some initial compression, the valve remains closed as

$$F_0 + F_{14} > F_0'$$

or,
$$F_{14} > (F_0' - F_0) = f(t_0' - t_0)$$

When the compressor starts, F_0 drops. A temperature difference $(t_0' - t_0)$ is created between the outlet and inlet of the evaporator, i.e., a degree of superheat results. The corresponding pressure difference $(p_0' - p_0)$ between the outside and inside of the bellow causes a downward force $(F_0' - F_0)$ on the bellow and the valve opens. Due to compression, F_{14} also increases. The opening of the valve is determined by the equilibrium position at which

$$(F_0' - F_0) = F_{14}$$

By adjusting the initial compression of the follow-up spring and hence the force F_{14}, the force $(F_0' - F_0)$ and hence the degree of superheat can be controlled. Normally, the setting of the spring is done in the factory to allow the valve to start opening at a superheat of 5°C.

8.3.1 Application of Thermostatic-Expansion Valves

The performance characteristic of the thermostatic-expansion valves is most suitable for application in air conditioning and refrigerant plants.

When the cooling load increases, the refrigerant evaporates at a faster rate in the evaporator than the compressor can suck. As a result, the pressure p_0 and the degree of superheat in the evaporator increase. The increase in superheat causes the valve to open more and to allow more refrigerant to enter the evaporator. At the same time, the increase in suction pressure p_0 also enables the compressor to deliver increased refrigerating capacity.

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fluid is a slightly higher boiling substance. It has lower pressures. It is seen that the degree of superheat Δt almost remains constant at different evaporator temperatures. Such a thermostatic expansion valve is called *cross-charged.*

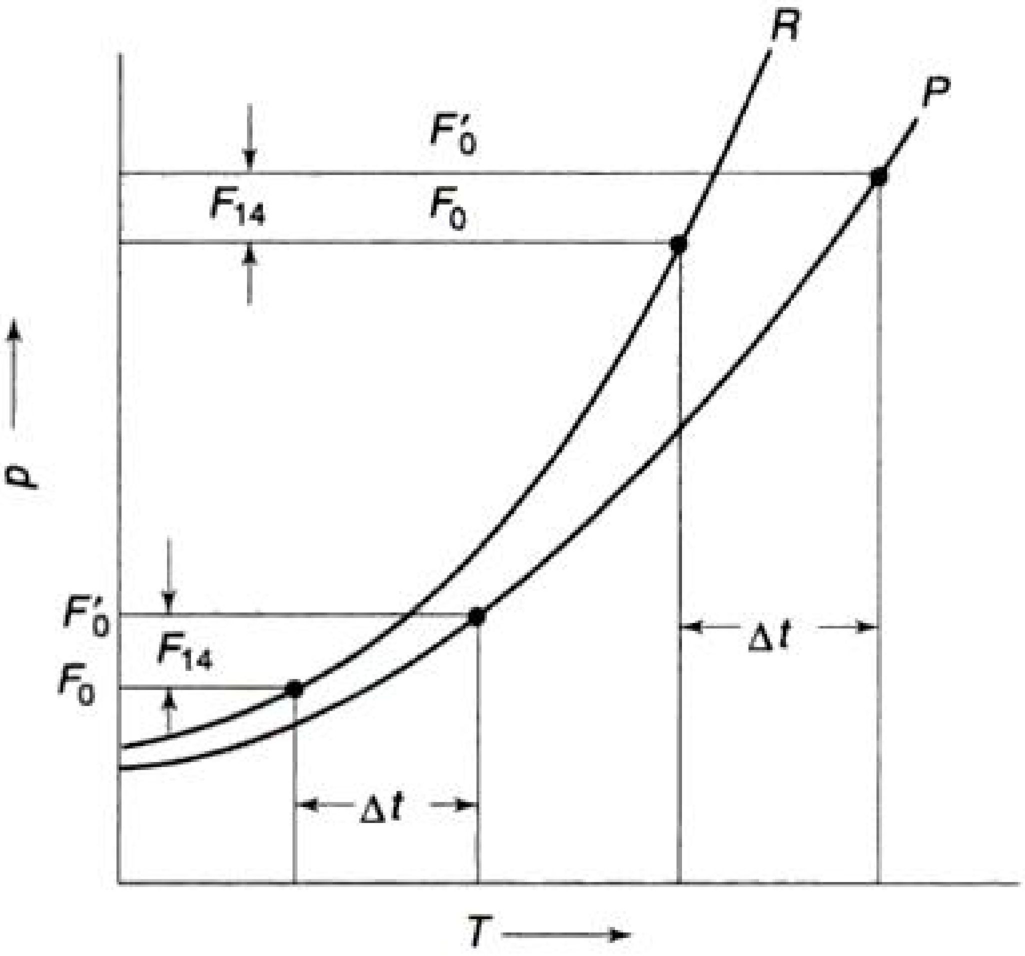

Fig. 8.8 Maintaining Uniform Superheat by the Use of Cross-Charged Expansion Valve

8.3.4 Thermostatic Charge and Fade-Out Point

The charge of the power fluid contained inside the bulb and the capillary tube of a thermostatic-expansion valve, limits the maximum suction pressure up to which the expansion valve modulates.

Consider the thermodynamic change of state of the power fluid as its temperature varies. The process take place along a constant-volume line as shown in Figs 8.9 and 8.10. Initially, at some stage, the fluid may exist at state A with the liquid and vapour in equilibrium at temperature t_A. With increase in temperature, the pressure increases according to the p-T relationship of the fluid until point B is reached. After B, at which all the liquid has turned into vapour, any further increase in temperature results in an insignificant increase in the pressure (according to the constant-volume process for a gas) inside the bulb and capillary of the expansion valve. As shown in Fig. 8.9, the increase in temperature equal to $t_C - t_B$ increases the pressure only by $p_D - p_B$. Thus at B, the pressure of the power fluid almost reaches a limiting value. The valve, therefore, does not open any wider with increase in the temperature, and the suction pressure stays constant at its value at B. The point B is called the *fade-out point.*

It can be seen that the fade-out point depends on the specific volume of the power fluid, i.e., the mass of the refrigerant charged in the power element, as the total volume of the element is fixed. Thus, the limiting value of the suction pressure can be increased by decreasing the specific volume, i.e., increasing the charge of the power fluid. Similarly, the limiting suction pressure can be lowered

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$$v_1 = v_{f_1} + x_1(v_{g_1} - v_{f_1}) \tag{8.2}$$

(iii) Calculate the velocities from the continuity equation at both the ends of the element

$$u_k = \frac{\dot{m}v_k}{A} \text{ and } u_1 = \frac{\dot{m}v_1}{A}$$

or

$$\frac{u}{v} = \frac{\dot{m}}{A} = G = \text{Constant} \tag{8.3}$$

where G is mass velocity.

(iv) For Fanno-line flow, an iteration procedure is necessary. This is done by applying the correction to enthalpy since $h_1 \neq h_k$. Thus

$$h_1 = h_k - \frac{u_1^2}{2} \tag{8.4}$$

The calculations for quality, specific volume, velocity and enthalpy may be repeated until the final value of h_1 is equal to its value in the preceding iteration.

***Note* ✍:** *The procedure appears time-consuming but, fortunately, it converges very fast.*

(v) Determine the pressure drop due to the acceleration, Δp_A, from the momentum equation

$$A\mathrm{d}p = -\dot{m}\mathrm{d}u$$

whence

$$\Delta p_A = \frac{\dot{m}}{A}(u_k - u_1) = G(u_k - u_1) \tag{8.5}$$

(vi) Determine the pressure drop due to the friction, Δp_F, from

$$\Delta p_F = \Delta p - \Delta p_A \tag{8.6}$$

(vii) Equate the required frictional pressure drop to

$$\Delta p_F = \frac{\rho f \,\Delta L u^2}{2D} \tag{8.7}$$

where $\rho = \dfrac{1}{v}$

ΔL = length of the element

Substituting $\dot{m} = \rho u A,\ \rho u = \dfrac{\dot{m}}{A} = G$, we have (8.8)

$$\Delta p_F = \frac{G}{2D} f u \,\Delta L = Y f u \,\Delta L \text{ (say)} \tag{8.9}$$

where

$$Y = \frac{G}{2D} \tag{8.10}$$

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$${}_k\Delta p_1 = 18.543 - 15.331 = 3.212 \text{ bar}$$

Acceleration pressure drop

$$\Delta p_A = G\,\Delta u = (5.83 \times 10^3)\,(10.474 - 5.317) = 32.8 \times 10^3 \text{ N/m}^2$$

Friction pressure drop

$$\Delta p_F = \Delta p - \Delta p_A = 3.212 \times 10^5 - 0.328 \times 10^5$$
$$= 2.884 \times 10^5 \text{ N/m}^2$$

Mean friction factor

$$f = \frac{0.0202 + 0.02}{2} = 0.0201$$

Mean velocity

$$u = \frac{5.317 + 10.474}{2} = 7.82 \text{ m/s}$$

Inremental length $${}_k\Delta L_1 = \frac{\Delta p_F}{Yfu} = \frac{2.884 \times 10^5}{(1.2674 \times 10^6)\,(0.0201)\,(7.82)} = 1.389 \text{ m}$$

The calculations for various sections are given in Table 8.4.

Table 8.4 Capillary Tube Length Calculations for Example 8.2

Sections	Δp	Δp_A	Δp_F	ΔL
	bar	*bar*	*bar*	*m*
k-1	3.212	0.328	2.884	1.389
1-2	3.416	0.574	2.842	0.706
2-3	2.818	0.857	1.961	0.276
3-4	2.292	1.283	1.009	0.087
4-5	0.969	0.844	0.125	0.008

Total length required = $\Sigma \Delta L$ = 2.466 m

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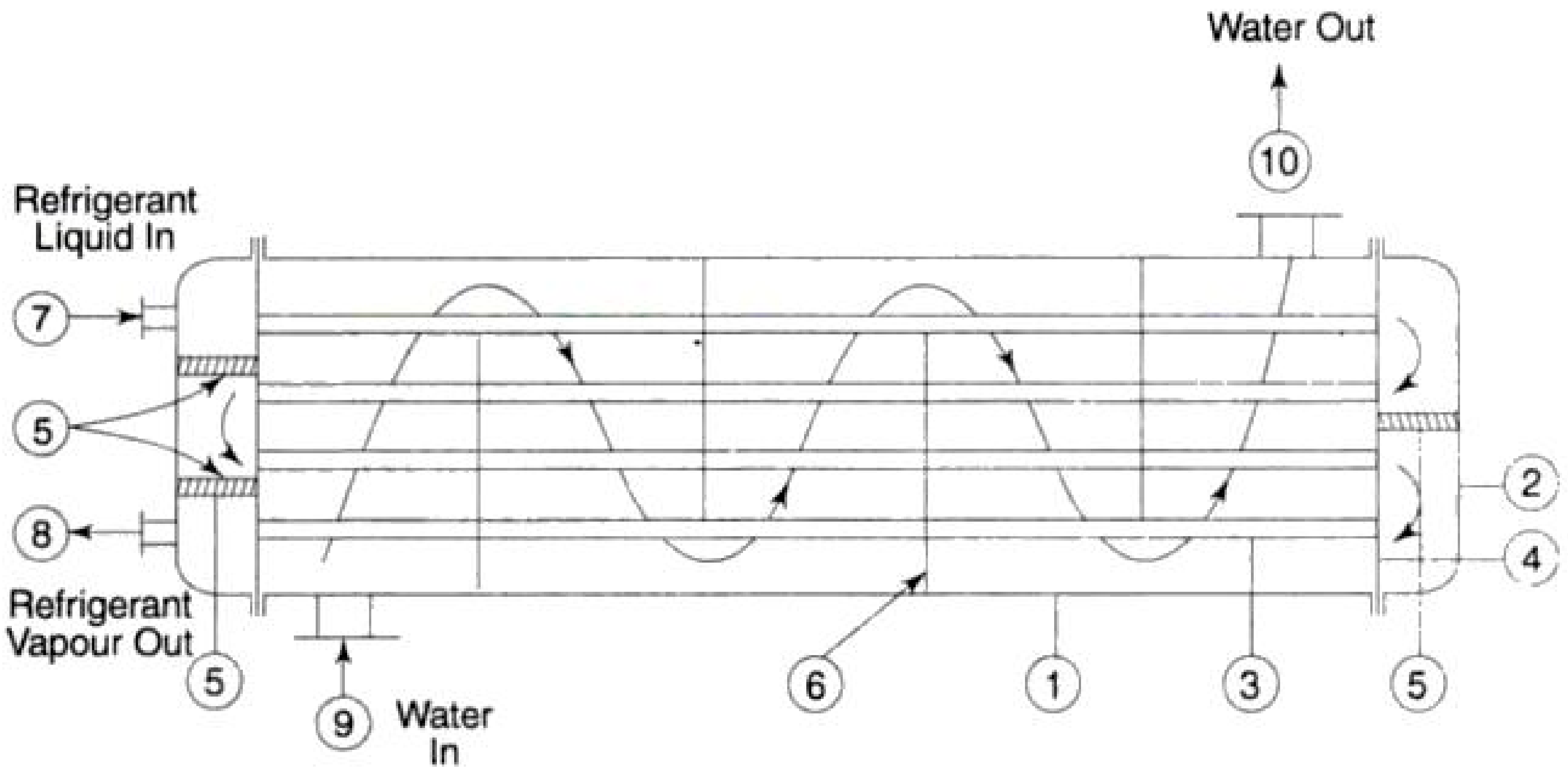

1 Shell
2 Headers
3 Tubes
4 Tube Sheet
5 Partitions
6 Baffles
7 Refrigerant Inlet (from Expansion Valve)
8 Refrigerant Outlet
9 Water Inlet
10 Water Outlet

Fig. 9.2 Direct-Expansion Chiller

Another common type of dry evaporator is the direct-expansion (D-X) coil as shown in Fig. 9.3 with fins on the air side used in air-conditioning equipment. The coil shown in the figure has 4 rows of tubing, and only one tube-circuit. Large D-X evaporators have more than one refrigerant circuit otherwise the pressure drop will become too large.

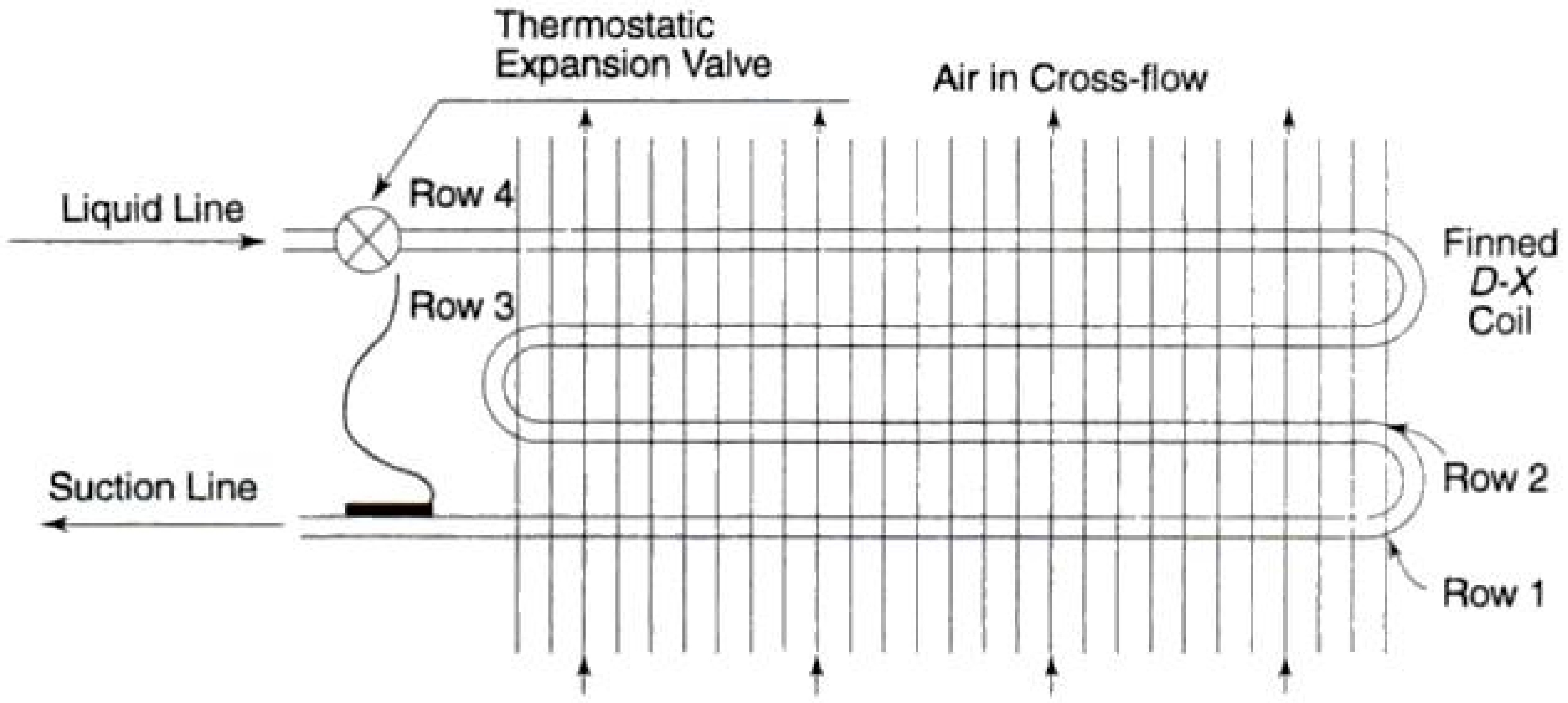

Fig. 9.3 Illustration of a Dry Evaporator, Direct-Expansion Cooling Coil with Thermostatic-Expansion Valve

The name direct-expansion is derived from the fact that the refrigerant expands directly inside the tubing, and evaporates, thus cooling the medium outside. To facilitate the return of oil to the compressor, D-X evaporators are fed from the top by a thermostatic-expansion valve as shown in Figs 9.2 and 9.3.

The transitions occur as the flow proceeds from the inlet end to outlet end of the evaporator.

The major portion of the heat transfer occurs in the horizontal tube through the wetted part of the tube wall. The heat-transfer coefficient increases when the flow becomes annular, but when most of the liquid is evaporated and the dry-out point is reached, the coefficient falls off rapidly to a low value.

To a large degree, the complete length of a single evaporator tube can be divided into three convenient parts, viz., stratified flow at the inlet, first slug, and then annular flow in the middle, and mist flow at the outlet.

A significant feature of boiling heat transfer is the dependence of the heat-transfer coefficient on quality. The nature of variation is shown in Fig. 9.5. For R11, h is maximum approximately at $x \leq 0.8$. For water it is maximum at $x = 0.4$. For other fluorocarbons the peak occurs at $0.4 < x < 0.8$. If this is true, a substantial improvement in the evaporator capacity is achieved with wet operation, represented by line *CD* in Fig. 9.6 as against dry operation with superheat as from *A* to *B*. The average value of h-when the quality changes from *C* to *D*—is much higher than the same when it changes from *A* to *B* as in normal evaporators without recirculation. This is achieved by recirculation (see Sec. 9.4.3).

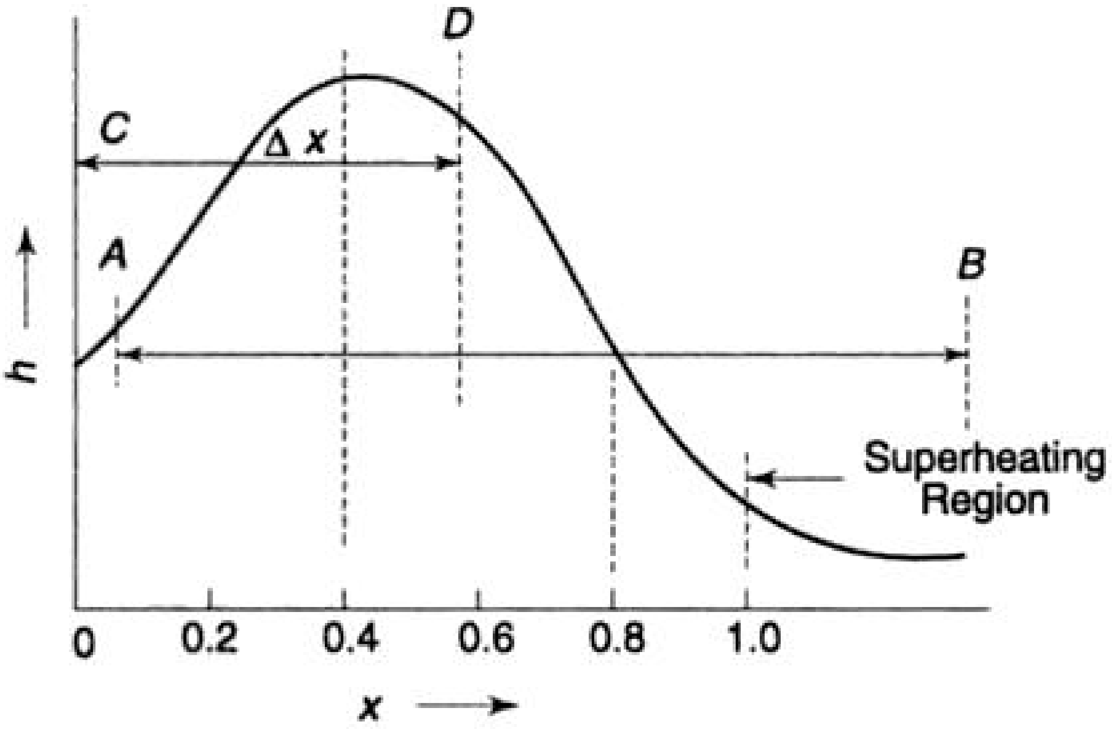

Fig. 9.5 Variation of Flow-Boiling Heat Transfer Coefficient with Dryness Fraction of Refrigerant

9.2.4 Forced Convection Boiling Correlations

Boiling of liquids has been of considerable interest to nuclear, chemical and mechanical engineers and, consequently, a lot of work has been done to obtain general correlations for predicting heat-transfer coefficients and pressure drops for flows with boiling. However, as verified by Andersen *et al.*, most of these do not satisfactorily predict the heat-transfer coefficients and pressure drops for evaporating refrigerants. Only Bo Pierre's[2] correlation predicts somewhat satisfactory values for the average heat-transfer coefficient but this does not give the local heat-transfer coefficient. His correlations are given below.

$$\text{Nu}_f = 0.0009\ (\text{Re}_f^2\ K_f)^{0.5} \tag{9.4}$$

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Solution

Heat transfer in each pass

$$\dot{Q} = \frac{(20)\,(3.5167)}{8} = 8.792 \text{ kW}$$

Temperature drop of water in each pass

$$\Delta t_w = \frac{11.1 - 7.2}{8} = \frac{3.9}{8} = 0.49°\text{C}$$

Mass flow rate of water

$$\dot{m}_w = \frac{(20)\,(2.5167)}{(4.1868)\,(3.9)} = 4.307 \text{ kg/s}$$

For Eighth Pass

Entering water temperature, $t_{w_1} = 11.1°\text{C}$

Leaving water temperature, $t_{w_2} = 11.1 - 0.49 = 10.61°\text{C}$

Mean temperature difference

$$\Delta t_m = \frac{0.49}{\ln (11.1 - 2.2)/(10.61 - 2.2)} = 8.85°\text{C}$$

Assume mean tube wall temperature $t_m = 8.8°\text{C}$. Then the refrigerant-side film coefficient is

$$h_i = 230\,(8.8 - 2.2) = 1518 \text{ W. m}^{-2}\text{ K}^{-1}$$

Overall heat-transfer coefficient based on outside tube surface area

$$\frac{1}{U_0} = \frac{1}{h_0} + \frac{1}{h_i}\,\frac{A_0}{A_i}$$

$$= \frac{1}{4650} + \frac{1}{1518}\left(\frac{1.905}{1.704}\right)$$

$$\Rightarrow \qquad U_0 = 1051 \text{ W.m}^{-2}\text{ K}^{-1}$$

Heat transfer area of the pass

$$A_0 = \frac{\dot{Q}}{U_0\,\Delta t_m}$$

$$= \frac{8.792 \times 10^3}{(1051)\,(8.85)} = 0.945 \text{ m}^2$$

$$A_i = 0.945 \times \frac{1.704}{1.905} = 0.845 \text{ m}^2$$

Check for the assumed valve of t_m

$$t_m = 2.2 + \frac{\dot{Q}}{h_i\,A_i} = 2.2 + \frac{8.792 \times 10^3}{(1518)\,(0.845)} = 9.05°\text{C}$$

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Combining Eqs (9.13) and (9.14), since $\mathrm{d}\dot{Q}_I = \mathrm{d}\dot{Q}_{II}$, we have

$$\frac{t_s - t_r}{h - h_s} = \frac{\eta_f \ f_g \ A_t}{U_i \ C_p \ A_i} = \text{const.} \tag{9.15}$$

where

t_r = Refrigerant temperature

t_s = Temperature of the wetted surface

h = Enthalpy of air

h_s = Enthalpy of saturated air at the temperature of the wetted surface

U_i = Overall heat-transfer coefficient based on inside tube surface area accounting for the thermal resistance of refrigerant side film, metal wall and outside condensate layer

C_p = Specific heat of air.

A detailed discussion on cooling and dehumidifying coils is given in Chapter 20.

Example 9.2 Simulation of Flooded Chiller

The word 'simulation' means recreation and validation of design and performance.

In an R 22 flooded chiller, the design conditions are as follows:

Evaporation temperature, $t_0 = 2°\text{C}$

Chilled water inlet temperature, $t_{w_1} = 11.1°\text{C}$

Chilled water circulation rate, $\dot{m}_w = 24.332$ kg/s

Construction details of the chiller are as follows:

Tube OD, $D_0 = 0.01704$ m Tube ID, $D_i = 0.01379$ m

Tube length, $L = 3.69$ m

Number of passes, $n = 2$

Number of tubes per pass = 114

As the heat transfer coefficient in pool boiling is very much lower than water-side coefficient, integral fin tubes with the following specifications are used:

Number of fins = 748 per metre

Diameter over fins, $D_{\text{fin}} = 0.01905$ m

Fin thickness, $b = 0.8$ mm

Estimate the capacity of the chiller in kW, accurate to the first approximation.

Solution

First Approximation

Let the temperature drop across the outside refrigerant film be $\Delta T_0 = 5°\text{C}$, and let the water leaving temperature be $t_{w_2} = 7°\text{C}$. Surface tension of R 22 at 2°C from Eq. (4.24) and constants given is

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9.4.1 Influence of Refrigerant Vapour Pressure Characteristic

The effect of using an augmentation method is to increase the heat transfer rate. But, it increases the pressure drop as well. The effect of increased pressure drop is either

(i) to cause a lowering of the compressor suction pressure and thus to decrease the capacity and performance of the whole system, or
(ii) to cause a decrease in the temperature potential (at the same suction pressure) for heat transfer and thus nullify, to some extent, the increase in the heat transfer obtained. Figures 9.8 and 10.7 illustrate this case. Augmentation of heat-transfer coefficient decreases the size of the evaporator, whereas decrease in ΔT increases its size.

A proper optimal balance between the two is, therefore, necessary. It is obvious that a refrigerant which causes a small change in the saturation temperature for a given change in the saturation pressure, will have a smaller fall in the temperature potential due to increased pressure drop in the evaporator. A qualitative estimate of the suitability of various refrigerants for use with augmentation techniques is thus provided by Table 9.2, which gives the temperature change corresponding to a saturation pressure change of 0.01 bar. It can be clearly seen that R 22 is the most suitable refrigerant for augmentation since it has the least saturation temperature change.

Table 9.2 Temperature Change of Refrigerants for 0.01 bar Pressure Drop

Temperature Range, °C	R 11	R 12	R 134a	R 21	R 22	R 717
0 to 5	0.535	0.0933	0.0876	0.312	0.0569	0.0585
– 10 to – 20	0.991	0.1439	0.1475	5.65	0.0893	0.098
– 35 to – 40	2.49	2.93	0.335	14.9	1.792	2.26

It can be anticipated that with R 11 and R 21, no substantial improvement in net heat transfer unit pressure drop may be possible while with R 12, R 134a, R 717 and R 22, significant improvements (in that order) may be obtained.

9.4.2 Augmentation Techniques

A large number of methods for the augmentation of heat transfer exist. These methods either increase the coefficient or increase the effective surface area, or do both. The most common of them are as follows:

(i) Turbulence promoters.
(ii) Swirl flow generators.
(iii) Extended surfaces.
(iv) Inner fins.
(v) Recirculation.
(vi) Roughening of surfaces.

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$$(R_{\text{fouling}})_0 = 0.00009 \text{ m}^2\text{K/W}$$

$$k_{\text{tubes}} = 0.386 \text{ kW/m.K}$$

Tubes are roughened with roughness parameter

$$R_p = 0.5\ \mu\text{m}$$

Determine the number of tubes required. Assume equal number of tubes in each pass. Use *Slipcevic correlations.* Assume fin efficiency $\eta_f = 1$.

Solution

Evaporator pressure, $p_0 = 5.283$ bar (at 2°C)
Critical pressure of R 22, $p_c = 49.36$ bar

$$\phi_p = \frac{p_0}{p_c} = \frac{5.283}{49.36} = 0.107$$

From Example 9.2, per metre length of tube, we have

$$\phi_a = \frac{A_t}{A_0} = \frac{0.13998}{0.0548} = 2.5544$$

Hence,

$$n_1 = 0.75\ (2.5544)^{-0.144}\ (0.107)^{0.088\ (2.5544)^{-0.25}} = 0.54$$

$$n_2 = 0.75\ (2.5544)^{-0.13}\ (0.107)^{-0.28} = 1.235$$

$$L = \left[\frac{\sigma}{g\,(\rho_f - \rho_g)}\right]^{0.5} = \left[\frac{0.0115}{9.81\,(1279 - 22.573)}\right]^{0.5} = 9.656 \times 10^{-4} \text{ m}$$

$$\varepsilon = \frac{R_p}{L} = \frac{0.5 \times 10^{-6}}{9.656 \times 10^{-4}} = 5.178 \times 10^{-4}$$

First Approximation

Assume number of tubes, $n = 228$
Length of tubes, $l = 3.69$ m
Area/m length of tube $= 0.14 \text{ m}^2$

$$q_t = \frac{\dot{Q}}{A_t\, nl} = \frac{422 \times 10^3}{0.14 \times 228\,(3.69)} = 3583 \text{ W/m}^2$$

$$\text{Re}_r = \frac{q_t\, L}{h_{fg}\, \mu_f} = \frac{(3583)\,(9.656 \times 10^{-4})}{(203.7 \times 10^{-3})\,(0.231 \times 10^{-3})} = 0.0735$$

$$Ph = \frac{C_f\, T}{h_{fg}} = \frac{1.77\,(273 + 2)}{203.7} = 1.589$$

$$P_{r_f} = 2.798$$

$$\text{Nu} = \frac{h_0\, L}{k_f} = 230\ (\phi_a)^{0.1}\ \text{Re}_r^{n_1} \left(\frac{Ph}{P_{r_f}}\right)^{n_2} \varepsilon^{0.133}$$

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First Approximation

Assume a refrigerating capacity of 140.7 kW. Then

$$\dot{m}_r = \frac{140.7}{409.7 - 253.7} = 0.9 \text{ kg/s}$$

Refrigerant flow area in a pass with n number of tubes

$$A = n\,\frac{\pi D_i^2}{4} = n\,\frac{\pi}{4}\,(0.0158)^2 = 1.96 \times 10^{-4}\, n \text{ m}^2$$

Refrigerant mass velocity

$$G = \frac{\dot{m}_r}{A} = \frac{0.9}{1.96 \times 10^{-4}\, n} = \frac{4590}{n} \text{ kg/s.m}^2$$

Shell-side heat-transfer coefficient (Kern's correlation[9])

$$\text{Nu} = \frac{h D_0}{k} = 0.36\left(\frac{D_e\, G_s}{\mu}\right)^{0.55} (\text{Pr})^{1/3} \left(\frac{\mu}{\mu_{\text{wall}}}\right)^{0.14} \tag{9.33}$$

Neglecting viscosity variation between bulk and wall fluid, we have

$$\text{Nu} = \frac{h D_0}{k} = 0.36\left(\frac{D_e\, G_s}{\mu}\right)^{0.55} (\text{Pr})^{1/3}$$

Now, tube pitch, $P_T = 0.0222$

Tube OD, $D_0 = 0.0191$ m

Equivalent diameter for triangular pitch

$$D_e = \frac{4}{\pi D_0}\left[0.86\, P_T^2 - \frac{\pi D_0^2}{4}\right] \tag{9.34}$$

$$= 9.15 \times 10^{-3} \text{ m}$$

Calculation of mass flow rate of water

$$\dot{Q}_0 = \dot{m}_w\, C_p\, (t_{w_1} - t_{w_2})$$

Assuming a net refrigerating capacity of 140.7 kW, we get

$$140.7 = \dot{m}_w\,(4.1868)\,(11.1 - 7.2)$$

$$\Rightarrow \qquad \dot{m}_w = 8.617 \text{ kg/s}$$

Cross-flow area S_s is defined as

$$S_s = D_s\,(P_T - D_0) \cdot \frac{P_B}{P_T} \tag{9.35}$$

We have shell dia, $D_s = 0.406$ m

Baffle pitch, $P_B = \dfrac{\text{Tube length}}{\text{No. of baffles}} = \dfrac{2.213}{22} = 0.1$ m

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Length of forced convection boiling section

$$L_A = \frac{(\dot{Q}_{\text{VI}})_A}{U_i\, n\, \pi\, D_i\, \Delta T} = \frac{27.1 \times 10^3}{1302\,(32)\,\pi\,(0.0158)\,(9.15 - 2.43)}$$
$$= 1.948 \text{ m}$$

Section B Forced convection Dittus-Boelter equation section

$$L_B = 2.213 - 1.948 = 0.265 \text{ m}$$

$$(x_B)_{\text{mean}} = \frac{0.975 + 1}{2} = 0.9875$$

assuming $x_{\text{out}} \cong 1$.
Properties of liquid-vapour mixture in B

$$\mu = x\,\mu_g + (1 - x)\,\mu_f$$
$$= 0.9875\,(1.2 \times 10^{-5}) + 0.0125\,(2.31 \times 10^{-4})$$
$$= 1.473 \times 10^{-5} \text{ kg/m.s}$$

Similarly,

$$C_p = 0.606 \text{ kJ/kg.K}$$
$$k = 0.01035 \text{ W/m.K}$$

Heat transfer calculations using Dittus-Boelter equation for L_B.

$$\text{Re} = \frac{G\,D_i}{\mu} = \frac{143.4\,(0.0158)}{1.473 \times 10^{-5}} = 154{,}000$$

$$\text{Pr} = \frac{C_p\,\mu}{k} = \frac{(0.606 \times 10^3)\,(1.473 \times 10^{-5})}{0.01035} = 0.862$$

$$h_i = 0.023\,(154{,}000)^{0.8}\,(0.862)^{0.4}\left(\frac{0.01035}{0.0158}\right)$$
$$= 200.6 \text{ W/m}^2\text{.K}$$
$$U_i = 188.6 \text{ W/m}^2\text{.K}$$
$$(\dot{Q}_{\text{VI}})_B = 0.53 \text{ kW}$$

Total heat transfer in Pass VI

$$\dot{Q}_{\text{VI}} = (\dot{Q}_{\text{VI}})_A + (\dot{Q}_{\text{VI}})_B = 27.1 + 0.53 = 27.63 \text{ kW}$$

$$x_{\text{out}} = 0.827 + \frac{27.63}{(0.9)\,(203.7)} = 0.978 < 1 \text{ (assumed)}$$

Repeat calculations do not change the value of $(\dot{Q}_{\text{VI}})_B$.
Seventh Pass ($n = 32$, $G = 143.4$ kg/s.m^2), similarly

$$\dot{Q}_{\text{VII}} = 4.5 \text{ kW},\ x_{\text{out}} = 1.002 \text{ (superheated vapour)}$$

Eighth Pass ($n = 34$, $G = 135$ kg/s.m^2)

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Chapter 10

Complete Vapour Compression System

10.1 THE COMPLETE SYSTEM

The four components of the vapour-compression system, viz., the compressor, condenser, expansion device and evaporator, each have their individual characteristics which are the functions of evaporator and condenser temperatures. These have been discussed earlier in Chapters 6 to 9. Then, the performance of the complete system can be obtained by superimposition or the simultaneous solution of the characteristics of the individual components. This can be done either graphically or analytically.

Often, the system may also include ancillary components as well, such as cooling tower, a boiler, ducting, etc. As the number of components increases, an analytical approach provides the only feasible answer. However, the graphical solution gives a better visual appreciation and comprehension about the behaviour of various components and the system.

***Note* ✍:** *The performance of the complete system also depends upon the mass of total refrigerant charge in the system.*

10.2 GRAPHICAL METHOD[4]

The graphical method involves the use of tables and charts from the catalogue data of individual components and successive graphical analysis, taking two components at a time as illustrated by the following examples.

10.2.1 Performance Characteristics of the Condensing Unit

The performance characteristics of a *condensing unit* (combination of compressor and condenser) are obtained by superimposing the characteristics of the two as shown in Fig. 10.1.

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$$h = a_0 + a_1\left(\frac{T}{100}\right) + a_2\left(\frac{T}{100}\right)^2 + \cdots + a_6\left(\frac{T}{100}\right)^6 \qquad (10.1)$$

Likewise, property correlations can be established for specific heat, pressure-temperature relationship, etc. Another form of representation is the exponential form

$$y - b = ax^m \qquad (10.2)$$

Yet another is like the ln p_{sat} versus $1/T_{\text{sat}}$ relationship.
When a variable is a function of two variables, e.g., the pressure ratio developed by a centrifugal compressor is a function of speed N and refrigerating capacity $\dot{Q}_0$ (see Fig. 6.18), then we may express as follows:

$$\frac{p_k}{p_0} = a + b\dot{Q}_0 + c\dot{Q}_0^2 \qquad (10.3)$$

where

$$a = A_0 + A_1N + A_2N^2$$

$$b = B_0 + B_1N + B_2N^2$$

$$c = C_0 + C_1N + C_2N^2$$

This representation requires nine data points to find the constants. The *least square method* may be used to find the constants if a large number of data points are available.

10.3.2 Mathematical Modelling

In mathematical modelling no catalogue data are necessary. An example of mathematical modelling is provided by analytical expressions/equations which are derived to represent the performance of heat exchangers. Thus for a counter flow heat exchanger, as shown in Fig. 10.4 in which subscripts c and h refer to cold and hot fluids respectively, we have for the rate of heat transfer by LMTD method,

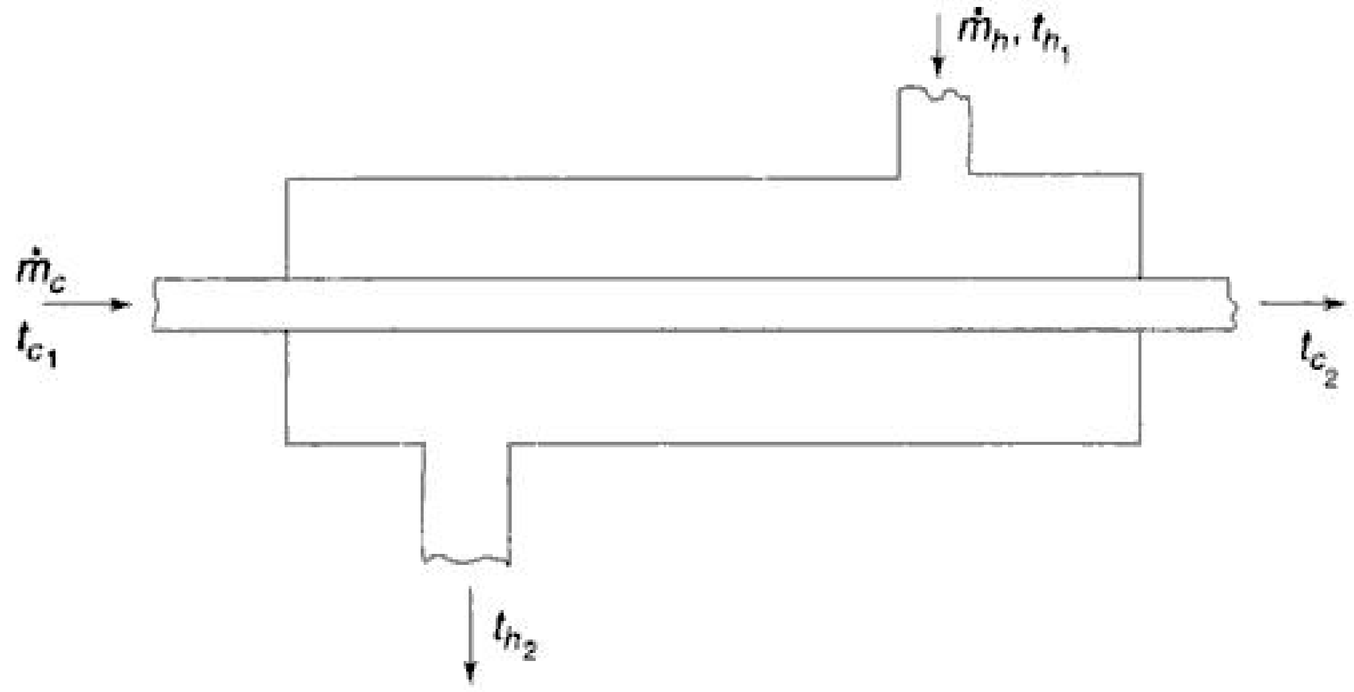

Fig. 10.4 Schematic Representation of a Simple Heat Exchanger

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Example 10.1 The performance data of a chilled water central air-conditioning plant in the range of operation is given below:

Condenser

$$(UA)_k = 20{,}700 \text{ W °C}^{-1}$$

Cooling water flow rate, $\dot{m}_w = 8.05$ kg/s

Evaporator

$$(UA)_0 = 23{,}840 \text{ W°C}^{-1}$$

Chilled water flow rate, $\dot{m}_b = 7.2$ kg/s

Compressor

Refrigerating capacity,

$$\dot{Q}_0 = 116{,}300 + 2910\,(t_0 - 5) - 1165\,(t_k - 43) \text{ W}$$

Power consumption,

$$\dot{W} = 13{,}490 + 1165\, t_0 + 350\, t_k \text{ W}$$

If the cooling water enters the condenser at 30°C and chilled water enters the evaporator at 11°C, find the temperatures of the cooling water leaving the condenser and the chilled water leaving the evaporator, and the refrigerating capacity and power consumption of the plant.

Solution

The mathematical expression for power, cooling and chilled water temperatures, t_w and t_b and refrigerating capacity are:

$$\begin{aligned}
\dot{W} &= \dot{Q}_k - \dot{Q}_0 \\
&= (8.05)\,(4187)\,(t_w - 30) - 7.2\,(4187)\,(11 - t_b) \\
t_w &= 30 + (t_k - 30)\,[1 - e^{-20{,}700/(8.05)\,(4187)}] \\
t_b &= 11 - (11 - t_0)\,[1 - e^{-23{,}840/(7.2)\,(4187)}]
\end{aligned}$$

$$\dot{Q}_0 = (7.2)\,(4187)\,(11 - t_b)$$

The simplified expressions are

$$\begin{aligned}
\dot{W} &= 33{,}705\, t_w + 30{,}146\, t_b - 1{,}342{,}756 \\
t_w &= 0.459\, t_k + 16.23 \\
t_b &= 0.547\, t_0 + 4.99
\end{aligned}$$

$$\dot{Q}_0 = 331{,}610 - 30{,}146\, t_b$$

The functions can be written as

$$\begin{aligned}
f_1 &= 33{,}705\, t_w + 30{,}146\, t_b - 1{,}342{,}756 - \dot{W} = 0 \\
f_2 &= 0.459\, t_k + 16.23 - t_w = 0 \\
f_3 &= 0.547\, t_0 + 4.99 - t_b = 0 \\
f_4 &= 331{,}610 - 30{,}146\, t_b - \dot{Q}_0 = 0
\end{aligned}$$

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PROBLEMS

10.1 For a Freon 12 compressor, the following performance data at a condensing temperature of 35°C are given:

Evaporator Temperature t_0(°C)	*Refrigerating Capacity* $\dot{Q}_0\,(W)$	*Power Consumption* $P(W)$
−20	2560	1800
−15	3325	2000
−10	4190	2200

Fit suitable equations for the refrigerating capacity and power consumption as a function of the evaporator temperature and constant condensing temperature of 35°C and find their numerical values at an evaporator temperature of 0°C.

10.2 A refrigeration plant serves as a water chiller. Data on the individual components are as follows:

Evaporator: $UA = 30595\ \text{W°C}^{-1}$
Chilled water flow rate, $\dot{m}_c = 6.795$ kg/s
Condenser: $UA = 26375\ \text{W°C}^{-1}$
Cooling water flow rate, $\dot{m}_w = 7.55$ kg/s

Compressor: The refrigerating capacity and power consumption as functions of evaporating and condensing temperature are given by the equations

$$\dot{Q}_0\ (\text{kW}) = -\,768 + 5.1\,T_0 - 2T_k$$
$$P(\text{kW}) = -\,0.732\,T_0 + 0.001746T_0^2 + 0.31\,T_k$$

Determine the condensing and evaporating temperatures, refrigerating capacity and power consumption for inlet-chilled water temperature of 2°C and inlet cooling water temperature of 30°C.

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(i) T_1 as the highest refrigeration temperature, and
(ii) T_3 as the lowest ambient temperature.

Point 1 on the diagram is fixed by the temperature T_1 and also the pressure p_0 which is generally equal to the surrounding atmospheric air pressure. Point 3 is fixed because of the limitations of the ambient temperature T_3 to which the gas can be cooled. Pressure p_k can, however, be varied within wide limits, starting from $p_{k,\,\min}$ onwards as shown in Fig. 11.3. With the compressor discharge pressure equal to $p_{k,\,\min}$, the refrigerating capacity of the machine is zero. The air is alternately compressed and expanded between points $2_{\min}$ and 1. The net work is also zero and hence the COP is indeterminate.

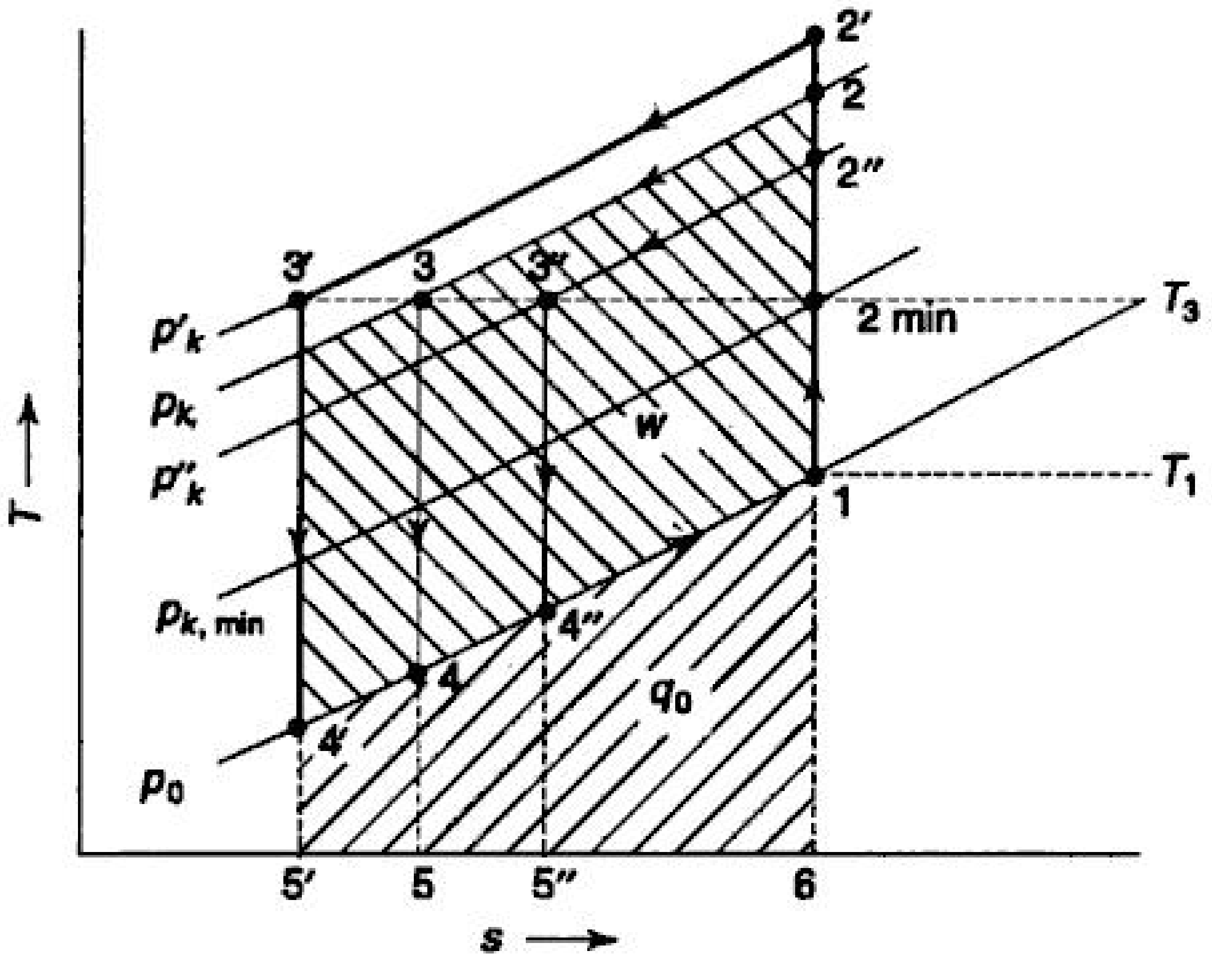

Fig. 11.3 Effect of Discharge Pressure p_k on the Performance of the Gas Cycle

However, as the pressure p_k is increased, although the refrigerating effect (area under the curve 4-1) and hence the capacity of the refrigerating machine increases, the work of the cycle also increases. For example, when the discharge pressure is p_k, the refrigerating effect is 1-4-5-6 and the net work is 1-2-3-4. When the discharge pressure is increased to p_k', the increase in the refrigerating effect is 4-4′-5′-5 and that in the net work is 2-2′-3′-4′-4-3-2. If is evident that the increase in work is much more than the increase in the refrigerating effect. As a result, the COP decreases with increasing p_k. Looking at Table 11.1, we find that a compression ratio of 3 to 4 in a single stage is reasonable.

11.2.2 Polytropic and Multistage Compression

As discussed in Sec. 6.2.3, polytropic compression with cooling would reduce the net work of the cycle by reducing the average temperature of the compression process and the value of the compression index from γ to n. Then, the expression for compressor work becomes

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$$T_2 = T_2' = T_1 + \frac{C^2}{2C_p} \tag{11.13}$$

The above relation can be simplified to

$$\frac{T_2'}{T_1} = 1 + \frac{C^2}{2C_p\,T_1} = 1 + \frac{C^2}{\dfrac{2\gamma R}{(\gamma-1)}\,T_1} = 1 + \frac{\gamma-1}{2}\,\frac{C^2}{a^2}$$

$$= 1 + \frac{\gamma-1}{2}\,M^2 \tag{11.14}$$

where $a = \sqrt{\gamma R T_1}$ is the acoustic velocity and M is the Mach number of the flight.

The temperature $T_2 = T_2'$ is the stagnation temperature of the ambient air. The stagnation pressure after isentropic diffusion, p_2', is given by the relation

$$\frac{p_2'}{p_1} = \left(\frac{T_2'}{T_1}\right)^{\frac{\gamma}{\gamma-1}} \tag{11.15}$$

The irreversible compression in the ram, however, results in air reaching point 2 instead of point 2′, i.e., at the same stagnation temperature but at a reduced stagnation pressure p_2 which is obtained from the knowledge of the ram efficiency η_R defined by

$$\eta_R = \frac{\text{Actual pressure recovery}}{\text{Ideal pressure recovery}}$$

$$= \frac{p_2 - p_1}{p_2' - p_1} \tag{11.16}$$

The rest of the cycle is also shown in Fig. 11.5. the temperature after the cooler (process 3-4), T_4 is shown to be higher than the stagnation temperature T_2 of the ambient air. It implies that the air cannot be cooled to a temperature below T_2 unless some method is devised to extract the kinetic energy of the ambient air. Another point to note is the pressure after expansion, p_5, which is slightly above the cabin pressure. It is also higher than the ambient air static pressure existing at the altitude at which the aircraft is flying. The refrigerating effect produced is

$$q_0 = C_p\,(T_i - T_5) \tag{11.17}$$

where T_i is the room temperature maintained inside the cabin. The net work of the cycle is the difference of work for process 2-3 of the compressor and process 4-5 of the expander plus the ram air work, $C_p\,(T_2 - T_1)$, which is derived directly from the engine.

The ambient air temperature varies with the altitude of the flight of the aircraft. Generally, the temperature drops by 0.64°C per 100 m of height from the sea level temperature.

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$$= \frac{294.1}{\left(\frac{5 \times 0.504}{1.01325}\right)^{0.286}} = 226.6 \text{ K}$$

Power required for refrigeration (excluding ram work)

$$\dot{W}_2 = |\dot{W}_C| - \dot{W}_E$$
$$= \dot{m} C_p (T_3 - T_2 - T_4 + T_5)$$
$$= 1\ (1.005)\ (466 - 294.1 - 294.1 + 226.6) = 104.4 \text{ kW}$$

Additional power for refrigeration

$$\dot{W}_1 = \dot{W}_2 - \dot{W}_1$$
$$= 104.4 - 65.34 = 39.06 \text{ kW}$$

Refrigerating capacity

$$\dot{Q}_0 = \dot{m} q_0 = \dot{m} C_p (T_{\text{cabin}} - T_5)$$
$$= (1)\ (1.005)\ (300 - 226.6) = 73.77 \text{ kW}$$

(b) Ram pressure

$$p_2 = p_1 + \eta_R (p_2 - p_1)$$
$$= 0.341 + 0.84\ (0.504 - 0.341) = 0.478 \text{ bar}$$

Power requirement for pressurization (excluding ram work)

$$\dot{W}_1 = \frac{(1)\ (1.005)\ (294.1)}{0.82} \left\{\left(\frac{1.01325}{0.478}\right)^{0.286} - 1\right\} = 86.4 \text{ kW}$$

Temperature after isentropic compression

$$T_3' = 466.0 \text{ K}$$

Temperature after actual compression

$$T_3 = T_2 + \frac{T_3' - T_2}{\eta_C}$$
$$= 294.1 + \frac{466 - 294.1}{0.82} = 503.7 \text{ K}$$

Temperature after heat exchanger

$$T_4 = T_3 - \varepsilon (T_3 - T_2)$$
$$= 503.7 - 0.8\ (503.7 - 294.1) = 336.0 \text{ K}$$

Temperature after isentropic expansion

$$T_5' = \frac{336}{\left(\frac{5 \times 0.504}{1.01325}\right)^{0.286}} = 258.9 \text{ K}$$

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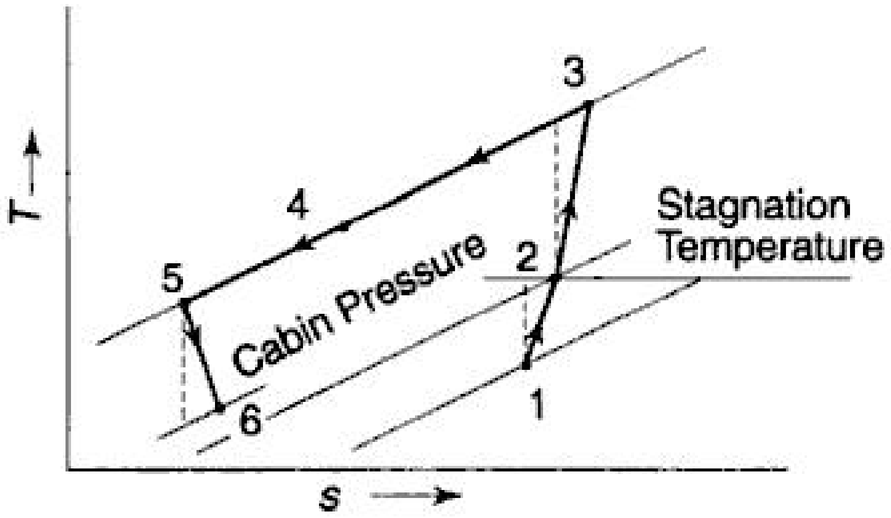

Fig. 11.10 Thermodynamic Cycle for Regenerative System

Reduced Ambient System In the reduced ambient system (Fig. 11.11) there are two expansion turbines—one in the cabin air stream and the other in the cooling air stream. Both turbines are connected to the shaft driving the fan which absorbs all the power. The turbine for the ram air operates from the pressure ratio made available by the ram air pressure. The thermodynamic cycle is shown in Fig. 11.12. The system shows promise for applications in an exceptionally high speed aircraft when the ram air temperature is too high. The cooling turbine reduces the temperature of cooling air to the level of static temperature of ambient air. Thus, primary compressed air can be cooled to, say, T_4 below the stagnation temperature T_2 and a little above the static temperature T_1.

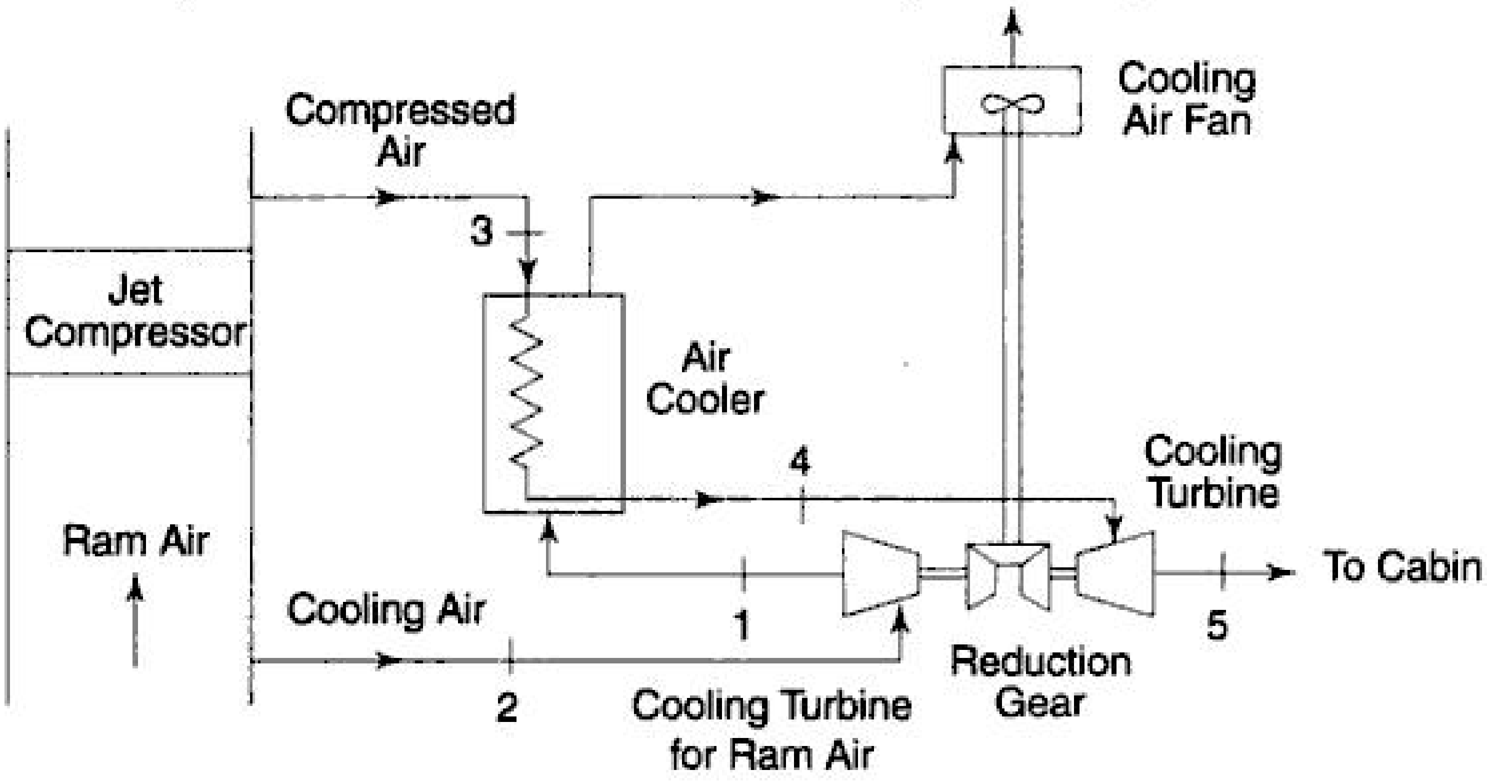

Fig. 11.11 Reduced Ambient System

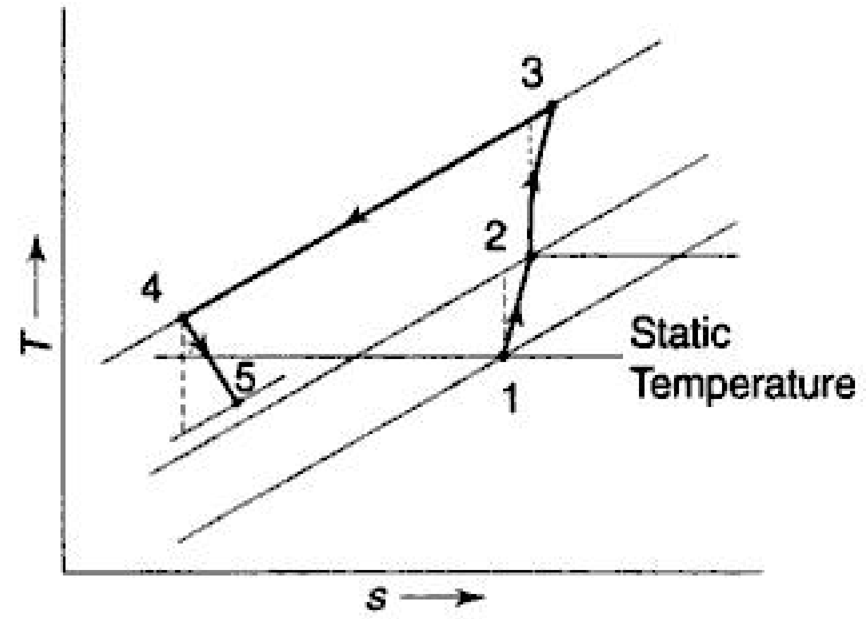

Fig. 11.12 Thermodynamic Cycle for Reduced Ambient System

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COP of the system is much lower than that of the Joule cycle. The energy balance of the vortex tube gives

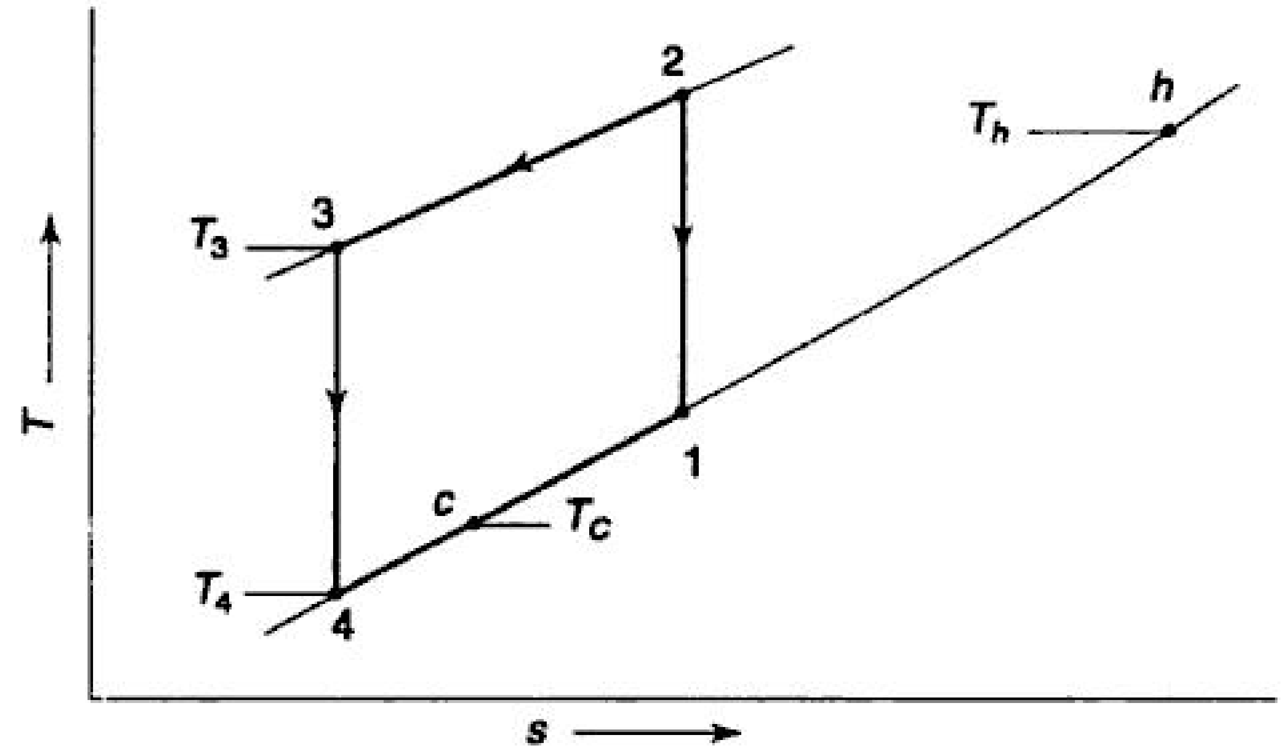

Fig. 11.17 Temperatures in Vortex Tube

$$T_3 = \mu_c T_c + \mu_h T_h \tag{11.18}$$

where $\mu_c + \mu_h = 1$ and T_c and T_h are the stagnation temperatures of the two streams.

The energy efficiency η_e of the vortex tube can now be defined as the ratio of the coefficient of performance of the vortex tube to that of an ideal refrigerating machine in which the work done by the expander is not recovered, viz.,

$$\eta_e = \frac{\mu_c (T_3 - T_c)}{T_3 - T_4} \tag{11.19}$$

It is seen that if all the air goes through the throttle valve, there will be no energy separation. Also, if all the air leaves through the orifice, there will again be no energy separation. Thus, there will be optimum values of fractions μ_c and μ_h for the maximum temperature drop, refrigerating effect and COP.

11.5 THE JOULE-THOMSON COEFFICIENT AND INVERSION CURVE

The Joule-Thomson coefficient defined in Sec. 1.10.3 by the expression

$$\mu_J = \left(\frac{\partial T}{\partial p}\right)_h$$

is a thermodynamic property. It may be readily obtained for a gas by making it undergo a throttling process through an insulated expansion valve. The pressure p_1 and temperature T_1 upstream are maintained constant and the pressure p_2 downstream is varied by operating the valve manually. For each setting of the valve, both p_2 and T_2 are measured. These states 1, 2, 2′, 2″, 2‴, etc., are then plotted on a T versus p diagram as shown in Fig. 11.18.

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Table 11.1 Saturation Properties of Air at 1 bar*

Pressure *bar*	*Temp.* *K*		*Specific* *volume* m^3/kg		*Specific* *enthalpy* kJ/kg		*Specific* *entropy* kJ/kg.K	
p	*Bubble*	*Dew*	v_f	v_g	h_f	h_g	s_f	s_g
1	78.7	81.7	0.00114	0.224	zero	205.3	zero	2.559

Table 11.2 Enthalpy of Air*

p	h							
bar	kJ/kg							
Temp., K→	120	130	140	150	160	200	250	300
100	70.1	98.1	128.0	159.5	190.3	276.7	347.6	408.5
150	70.4	95.8	121.5	147.5	172.8	257.5	335.1	400.5
200	72.1	95.8	119.4	142.7	165.5	246.0	325.9	394.3
250	75.5	98.2	120.6	142.3	163.4	239.8	319.7	389.8
300	80.0	102.2	123.6	144.3	164.1	236.8	315.9	386.8
350					166.8	236.2	314.1	385.2
400					170.5	237.1	313.7	384.9

Table 11.3 Entropy of Air*

p	s							
bar	kJ/kg.K							
Temp., K	120	130	140	150	160	200	250	300
100	0.625	0.849	1.070	1.288	1.486	1.972	2.291	2.513
150	0.572	0.775	0.966	1.145	1.308	1.782	2.130	2.369
200	0.532	0.723	0.898	1.059	1.206	1.657	2.015	2.265
250	0.507	0.689	0.855	1.005	1.142	1.569	1.926	2.182
300	0.489	0.667	0.826	0.968	1.095	1.502	1.856	2.115
350					1.064	1.451	1.800	2.060
400					1.040	1.412	1.754	2.013

* Extracted from Haywood R.W., Thermodynamic Tables in S.I. Units, Cambridge University Press, 1968.

Example 11.5 Minimum Work of Compression for Liquefaction of Air.

Determine the yield and minimum work of compression per unit mass of liquid air as a function of compressor discharge pressure. Given $T_1 = 300$ K.

Solution From Tables 11.1 to 11.3, at $p_1 = 1$ bar

$$h_f = 0,\ h_g = 205.3 \text{ kJ/kg}$$

$$h_1 = 427.8 \text{ kJ/kg},\ s_1 = 2.559$$

At $p_2 = 150$ bar, $T_2 = 300$ K, we have

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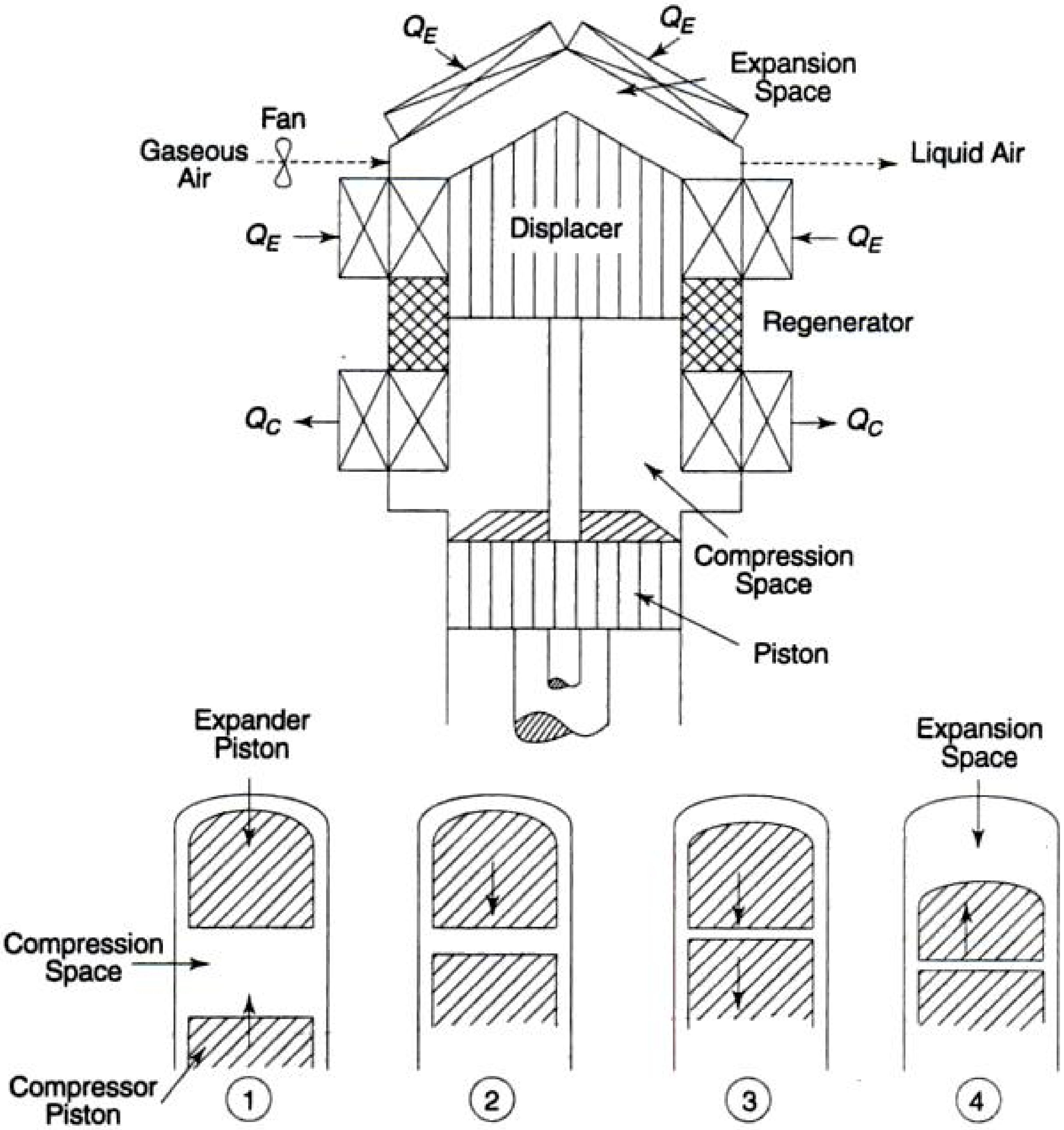

Fig. 11.23 β-Machine with Integral Cylinder for Compression and Expansion Spaces as in Philips Liquefier

Terminal point 1

Compression space is filled with gas. There is no gas in expansion space.

Process 1-2

Expander-piston is stationary. Compressor-piston moves. The gas is compressed rejecting heat Q_C.

Process 2-3

Both the compressor and the expander pistons move. The gas is transferred from compression space to expansion space via the regenerator, getting cooled in the process.

Process 3-4

Compressor-piston is stationary. Expander-piston moves. The gas is expanded absorbing heat Q_E.

Process 4-1

Both the compressor and the expander pistons move. The gas is transferred back from expansion space to compression space via the regenerator, getting heated in the process.

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$$\frac{p_e V_e}{R T_e} + \frac{p_c V_c}{R T_c} + \frac{p_d V_d}{R T_d} = \frac{1}{2} K \frac{V_E}{R T_c} = m_T \tag{11.24}$$

where K is a constant which depends on the total mass of gas charged in the system. Note, that this will also affect the system pressures.

Now, since compression and expansion are isothermal, hence $T_e = T_E$ and $T_c = T_C$. Then, substituting for volumes, putting $p_e = p_c = p_d = p$, eliminating R, and rearranging, we get

$$\frac{K}{p} = \frac{T_C}{T_E}(1 + \cos\phi) + \kappa[1 + \cos(\phi - \alpha)] + 2\frac{V_D}{V_E}\frac{T_C}{T_D}$$

$$= \tau(1 + \cos\phi) + \kappa[1 + \cos(\phi - \alpha)] + 2S \tag{11.25}$$

where $$T_D = \frac{T_E + T_C}{2} = \frac{T_C}{2}\left(1 + \frac{T_E}{T_C}\right) = \frac{T_C}{2}\left(\frac{\tau + 1}{\tau}\right)$$

Eq. (11.25) can be written as

$$\frac{K}{p} = \sqrt{(T + \kappa\cos\alpha)^2 + (\kappa\sin\alpha)^2}\,\cos(\phi - \theta) + \tau + \kappa + 2S$$

$$= \sqrt{\tau^2 + 2\tau\kappa\cos\alpha + \kappa^2}\,\cos(\phi - \theta) + \tau + \kappa + 2S$$

$$\Rightarrow \qquad p = \frac{K}{B[1 + \delta\cos(\phi - \theta)]} \tag{11.26}$$

The instantaneous pressure p is

(a) minimum when $\phi = \theta$, $\phi - \theta = 0$, and

(b) maximum when $\phi = \theta + \pi$, $(\phi - \theta) = \pi$.

Thus, we have

$$p_{\min} = \frac{K}{B(1 + \delta)},\ p_{\max} = \frac{K}{B(1 - \delta)},\ \frac{p_{\max}}{p_{\min}} = \frac{1 + \delta}{1 - \delta}$$

Accordingly, from Eq. (11.26),

$$p = p_{\max}\frac{1 - \delta}{1 + \delta\cos(\phi - \theta)}$$

$$= p_{\min}\frac{1 + \delta}{1 + \delta\cos(\phi - \theta)} \tag{11.27}$$

Mean Cycle Pressure

$$p_{\text{mean}} = \frac{\pi}{2}\int_0^{2\pi} p \, \mathrm{d}(\phi - \theta)$$

$$= \frac{\pi}{2}\int_0^{2\pi} p_{\max}\frac{1 - \delta}{1 + \delta\cos(\phi - \theta)}\,\mathrm{d}(\phi - \theta)$$

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$$\theta = \tan^{-1}\left(\frac{\kappa \sin\alpha}{\tau + \kappa\cos\alpha}\right) = 61.6^\circ$$

$$\sin\theta = 0.6581$$

$$Q_{\max} = \frac{Q_E}{p_{\max}\, V_T} = \frac{\pi}{1+\kappa}\sqrt{\frac{1-\delta}{1+\delta}}\,\frac{\delta\sin\theta}{1+\sqrt{1-\delta^2}} = 0.112$$

$$Q_E = 0.112\,(20\times 10^5)\,(2\times 6\times 10^{-5}) = 26.9\text{ J}$$

Refrigeration capacity, power consumption and pressures

$$\dot{Q}_E = Q_E\,N/60 = \frac{26.9\,(720)}{60} = 323\text{ W}$$

$$\dot{W} = \dot{Q}_E/\text{COP} = (\tau - 1)\,\dot{Q}_E = 46.5\text{ W (Actual)}$$

$$p_{\text{mean}} = p_{\max}\sqrt{\frac{1-\delta}{1+\delta}} = 14.93\text{ bar}$$

$$p_{\min} = p_{\max}\left(\frac{1-\delta}{1+\delta}\right) = 11.14\text{ bar}$$

11.6.8 Design Aspects of Regenerator

The regenerator is a critical component in Stirling machines. The regenerator efficiency should be close to 0.98 to 1. A 1% reduction in efficiency results in 21% reduction in Q_E at 75 K. At the same time, heat exchange in regenerator is very large, viz.,

$$Q_{\text{REG}} \cong 10\text{ to }50\ Q_E$$

Hence, the desirable properties of regenerator *matrix* material are:

(a) High heat capacity (high density and specific heat).
(b) Large thermal conductivity.
(c) Low pressure drop inspite of large heat-transfer surface of matrix.

Comparing a few materials, Table 11.5 clearly highlights that MS pipe can be used for the casing, while copper would be the best choice for the matrix.

Table 11.5 Comparison of Regenerator Materials

	MS	*Copper*	*Bronze*	*Brass*
C_p, kJ/kg.K	0.465	0.383	0.343	0.385
k, W/m.K	54	386	26	111

After choosing the material, it is also very important to select a *mesh number* of proper size. The mesh wire diameter d_m should be optimum.

(a) *If d_m is too small*, heat penetrates to the centre before the *blow time* expires. This means *storage volume* is insufficient.

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Note ✍: *Thus, the COP of 0.34 of the heat-operated vapour absorption system is equivalent to COP of 3.69 of vapour compression system in this case.*

12.3 COMMON REFRIGERANT-ABSORBENT SYSTEMS

Some of the desirable characteristics of a refrigerant-absorbent pair for an absorption system are low viscosity to minimize pump work, low freezing point and good chemical and thermal stability. Irreversible chemical reactions of all kinds, such as decomposition, polymerization, corrosion, etc., are to be avoided.

In addition to the above, two main thermodynamic requirements of the mixture are:

(i) *Solubility requirement*: The refrigerant should have more than Raoult's law solubility in the absorbent or adsorbent so that a strong solution, highly rich in the refrigerant, is formed in the absorber by the absorption of the refrigerant vapour.

(ii) *Boiling points requirement*: There should be a large difference in the normal boiling points of the two substances, at least 200°C, so that the absorbent exerts negligible vapour pressure at the generator temperature. Thus, almost absorbent-free refrigerant is boiled off from the generator and the absorbent alone returns to the absorber. If absorbent vapour goes with the refrigerant vapour to refrigerant circuit, the refrigeration produced will not be isothermal. In case a solid adsorbent is used, it does not exert any vapour pressure. Thus, pure refrigerant vapour only will go to the refrigerant circuit.

The two commonly used pairs are those of refrigerant NH_3 + absorbent H_2O, and refrigerant H_2O + adsorbent $LiBr_2$.

In the ammonia-water system, ammonia is the refrigerant and water is the absorbent. Ammonia forms a highly non-ideal solution in water. Hence, from the point of view of the solubility requirement, it is satisfactory. But the difference in their boiling points is only 138°C. Hence the vapour leaving the generator contains some amount of water.

Thus, the ammonia-water system is not suitable from the point of view of the boiling point requirement.

In the water-lithium bromide system, water is the refrigerant and lithium bromide is the adsorbent. Hence the mixture is used only in air-conditioning applications since water freezes at 0°C. The mixture is again non-ideal and is satisfactory from the point of view of the solubility requirement. Since lithium bromide is a salt, it exerts no vapour pressure. So the vapour leaving the generator is a pure refrigerant. The mixture, therefore satisfies the boiling point requirement also. However, it is corrosive and the plant works under high vacuum, both in condenser and evaporator. Hence, a purge unit is used.

The thermodynamic properties of the aqua-ammonia system are available in the form of Jennings and Shannon[7] tables. These tables give values of saturated

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Now, if generator temperature is $t_h = 156°\text{C}$, we have for poor solution concentration of NH_3

$$\xi_a^L = 0.2 \text{ (sat. liq. at 20.33 bar, 156°C)}$$

and if absorber temperature is $t_A = 40°\text{C}$, we have for rich solution concentration of NH_3

$$\xi_r^L = 0.34 \text{ (sat. liq. at 2.25 bar, 40°C)}$$

The concentration of NH_3 in vapour leaving generator

$$\xi_d = \xi_a^V = 0.73 \text{ (sat. vap. at 20.33 bar, 156°C)}$$

Then, from Eqs (12.7) and (12.8), we have per unit mass of vapour leaving generator, circulation rates of rich solution leaving absorber and poor solution leaving generator respectively as

$$f = \frac{\xi_d - \xi_a}{\xi_r - \xi_a} = \frac{0.73 - 0.2}{0.34 - 0.2} = \frac{0.53}{0.14} = 3.8 \text{ kg/kg vapour}$$

$$f - 1 = 2.8 \text{ kg/kg vapour}$$

But, the difference in N.B.P. of NH_3(–33°C) and that of H_2O (100°C) is small, only 133°C. The consequences of the same are the following:

(i) Considerable amount of absorbent water is present in the vapour leaving generator. This goes to the refrigerant circuit. It is equal to $(1 - \xi_d) = 1 - 0.73 = 0.27$ (27%) in this case.

(ii) As a result, temperature of – 17.5°C would not be attained after throttling to 2.25 bar.

(iii) To attain this temperature, an evaporator pressure lower than 2.25 bar would be required.

(iv) Let the throttling be done to an evaporator pressure of $p_0^* = 1$ bar. Even with this low pressure, the temperature attained after throttling would be $t_{0_1} = -19°\text{C}$ only.

(v) After complete evaporation, the temperature of this liquid-vapour NH_3/H_2O mixture would be as high as $t_9^* = 76°\text{C}$ (Fig. 12.4). As the maximum refrigeration temperature required is $t_{0_2} = -17.5°\text{C}$, it is seen that only a small fraction of the mixture could be allowed to evaporate. The rest would have to go to the absorber, along with vapour, without producing any refrigerating effect. Hence, Q_0 would be very much reduced.

(vi) Further, with the lowering of absorber pressure to 1 bar, the new rich solution concentration would be

$$\xi_r^* = 0.234 \text{ (sat. liq. at 1 bar, 40°C)}$$

Thus, $\xi_r^* - \xi_a = 0.235 - 0.2 = 0.035$ would become very small resulting in extremely large solution circulation rates f and $(f - 1)$, and hence large quantity of heat added in generator, as seen from Eq. (12.9), and very low COP.

To return the absorbent to the generator and to allow, as far as possible, only the refrigerant vapour to enter the condenser, two elements are added to the simple absorption system. These are:

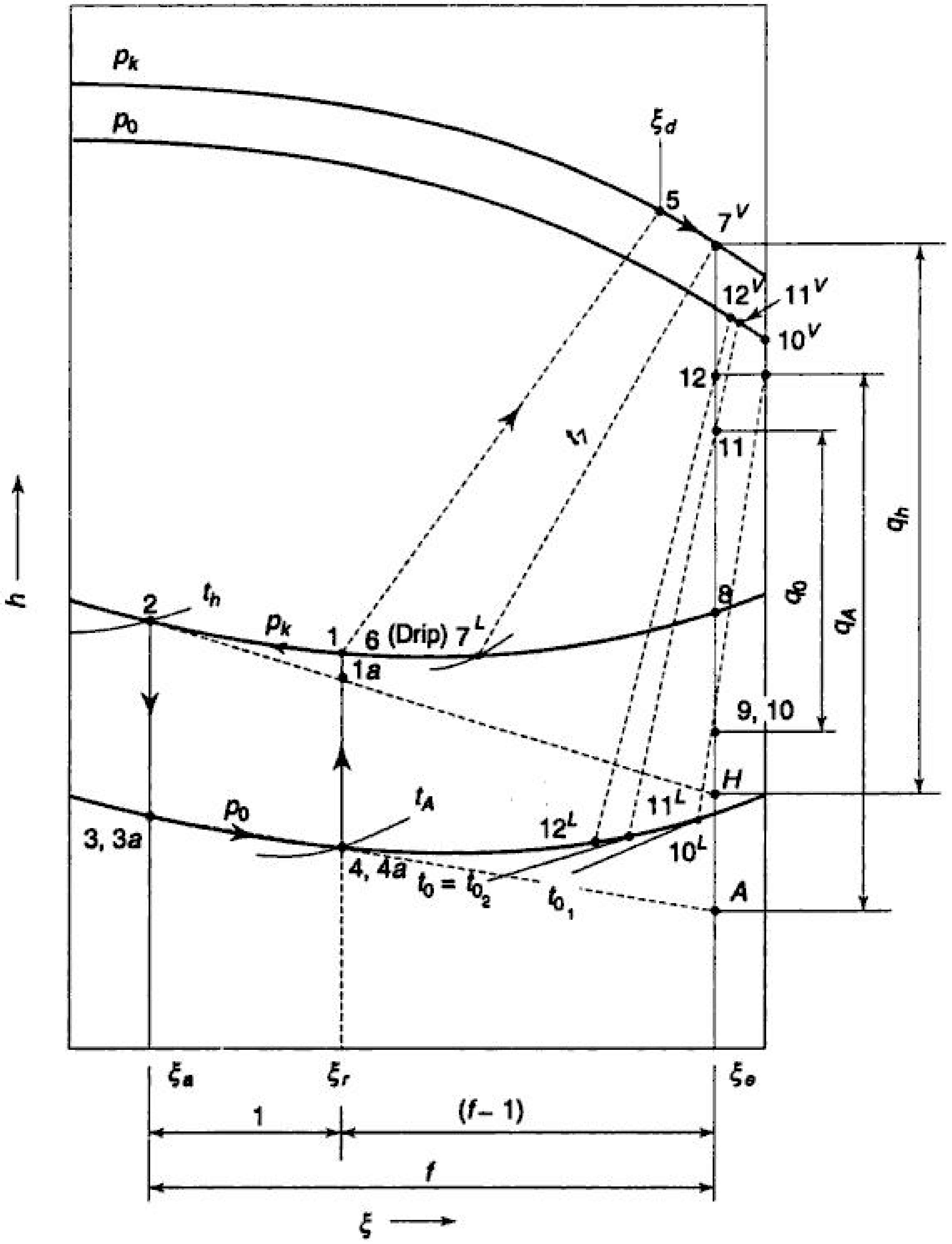

Fig. 12.9 Representation of Vapour-Absorption Cycle on h-ξ Diagram

The vapours at 5, distilled from the generator-analyser, enter the dephlegmator. The vapours from the dephlegmator with higher concentration of the refrigerant at 7 then enter the refrigerant circuit, whereas the drip at 6 returns to the generator-analyser. The vapours are condensed to 8 in the condenser, precooled to 9 in the liquid vapour regenerative heat exchanger and throttled to 10 before entering the evaporator. The state 10 is at the same point as state 9 on the h-ξ diagram as both enthalpy and composition remain the same before and after throttling. The refrigerant entering the evaporator at 10, leaving the evaporator at 11 and the liquid vapour heat exchanger at 12, comprises a liquid plus vapour mixture. The refrigerant is finally absorbed by the poor solution at 2 returning from the generator after being cooled in the liquid-liquid heat

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Evaporator

$$q_0 = h_{11} - h_{10} \tag{12.24}$$

The overall energy balance gives

$$q_h + q_0 + q_P = q_C + q_A + q_R = q_k$$

12.5.2 Auxiliary Quantities

It can be seen from Fig. 12.9 that if points 2 and 1a are joined by a line and the straight line is extended to meet the vertical line corresponding to the leaving vapour concentration ξ_e at H, then the projections of 2-1a, 1a-H and 2-H on the abscissa are proportional to 1, $(f - 1)$ and f respectively. Also, the enthalpy at H is given by

$$h_H = h_2 - f(h_2 - h_{1a}) \tag{12.25}$$

which is the same as the auxiliary quantity for the generator as defined in Eq. (12.10).

Similarly, if one joins points 3 and 4 and extends the line to meet the same vertical at A, then we get the auxiliary quantity for the absorber

$$h_A = h_3 - f(h_3 - h_4) \tag{12.26}$$

as defined by Eq. (12.17).

These auxiliary quantities make it possible to geometrically represent the generator and absorber heat quantities on the h-ξ diagram as shown in Fig. 12.9.

12.5.3 Rich and Poor Solution Concentrations

As stated earlier, the specific solution circulation f is a very significant quantity. The larger the value of f, the more is the pump work. Higher solution circulation rates of both rich and poor solutions involve larger pressure drops in the system and hence still more pump work. In addition, as seen from Eq. (12.9), a higher value of f results in a larger heat input to the generator q_h and hence a lower COP. Also, the heat rejected in the absorber q_A is more. And hence high f implies larger sizes of both generator and absorber and accompanying losses in addition to a lower COP.

It is, therefore, desirable to have f as small as possible. It is seen from Eq. (12.6) that for f to be small, the degasing range $(\xi_r - \xi_a)$ should be as large as possible. In other words, the rich-solution concentration ξ_r should be as high, and the poor-solution concentration ξ_a should be as low as possible.

The absorber pressure in the absorption system is equal to the evaporator pressure p_0 and the generator pressure is similarly equal to the condenser pressure p_k. Hence at a given generator pressure, the poor solution concentration is determined by the heating temperature t_h, and at a given absorber pressure, the rich solution concentration is determined by the cooling temperature t_A. The higher the t_h, the lower is ξ_a. And lower the t_A, the higher is ξ_r. Hence the generator temperature should be as high as possible, whereas the absorber temperature should be as low as possible.

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$$\ln p = a' - b' \frac{h_{fg} + \Delta h_m}{T} \text{ or } \ln p = A - \frac{B}{T}$$

It is to be noted that both h_{fg} and Δh_m have been assumed to be independent of temperature T. It has also been assumed that during boiling in the generator, no appreciable quantity of the absorbent is evaporated, meaning that the solvent exerts a negligible vapour pressure. This assumption is true for water-lithium bromide mixture.

Thus, $\ln p - \frac{1}{T}$ relation, to a great extent, follows a straight line for a pure substance as well as for a mixture. We have one such line for the refrigerant and another for the absorbent. We can obtain such straight lines for each composition of the mixture. The constant composition lines are parallel if $\Delta h_m = 0$. They diverge towards higher temperatures if Δh_m is positive, and converge if Δh_m is negative as shown in Fig. 12.10. The absorption system refrigerant-absorbent pairs have negative heat of mixing. Hence, these lines are convergent for absorption system.

The Appendix gives the $\ln p - 1/T$ diagram for H_2O – Li Br_2 system.

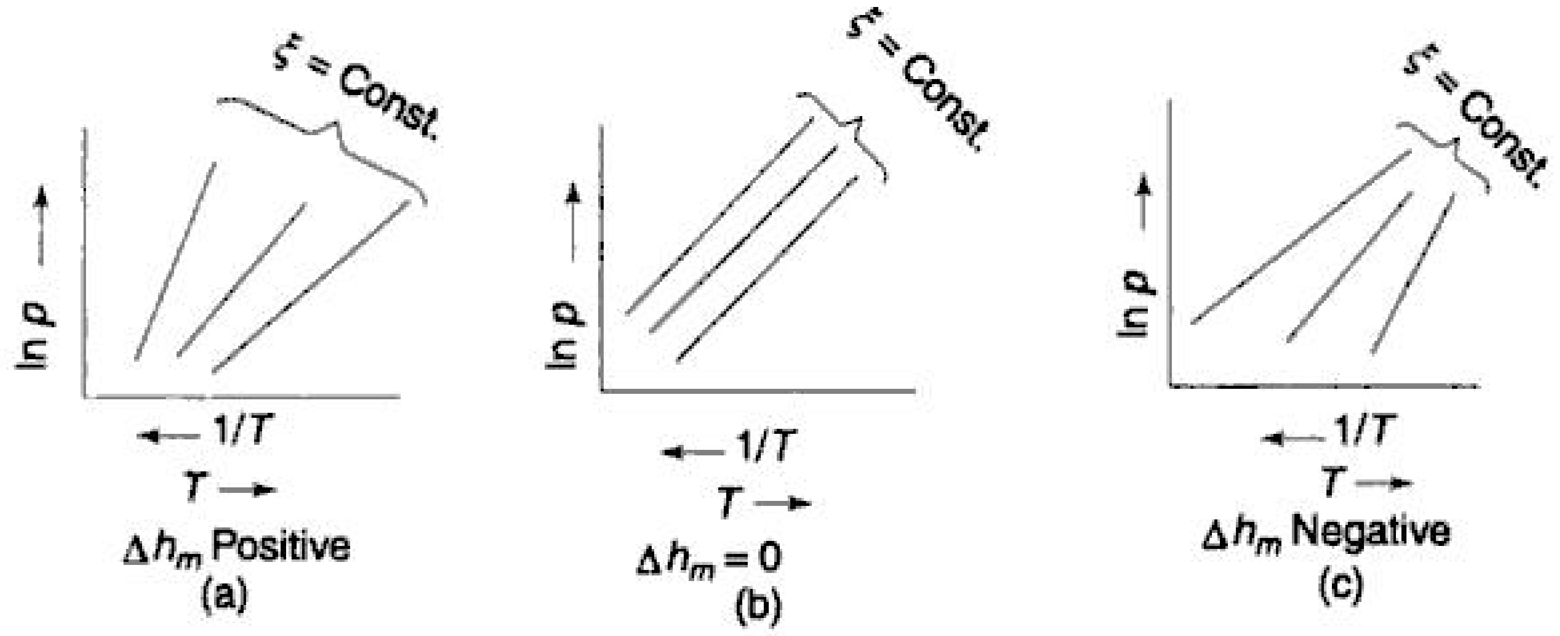

Fig. 12.10 Nature of ln $p - 1/T$ Lines for Mixtures

Figure 12.11 shows a simple absorption system of Fig. 12.1 with the liquid-liquid heat exchanger incorporated in it in addition. The ln p $-1/T$ diagram of the cycle in Fig. 12.12 represents its working. It is a *single-effect* H_2O – $LiBr_2$ vapour absorption system since it has one-stage of generation of vapour. The *double-effect* is described in Sec. 12.8.

Line R on this diagram represents ln p^{sat} versus $1/T^{sat}$ relationship of the refrigerant ($\xi = 1$). Then, we draw the two horizontal lines corresponding to the condenser pressure p_k at condenser temperature T_c, and evaporator pressure p_0 at evaporator temperature T_0. The various state points are located as follows:

State 6

Saturated liquid refrigerant at condenser temperature T_C on $\xi = 1$ line.

States 7 & 8

Liquid plus vapour state 7 after throttling, and saturated vapour state 8 at T_0 on $\xi = 1$ line.

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T_C, T_0 and T_A. It is much above the ambient temperature. On account of other irreversibilities, the required minimum temperature in generator is still higher.

***Note* ✍:** *It is thus seen that an absorption cycle cannot be devised simply from a heat source having temperature higher than the ambient temperature. To make the absorption system possible, its temperature should be above this* $T_{h_{min}}$.

12.7 PRACTICAL SINGLE-EFFECT WATER-LITHIUM BROMIDE ABSORPTION CHILLER

Several Inc., U.S.A. developed an absorption machine which uses solution of lithium bromide in water, water being the refrigerant and lithium bromide which is a highly hygroscopic salt-the adsorbent. Thus the solution leaving the absorber, being rich in *refrigerant water*, is a weak solution of salt in water. But for the sake of consistency, we shall continue to use ξ_r to denote the refrigerant concentration which, in this case, will mean the concentration of water in the solution leaving the absorber. Similarly, the solution returning from the generator is a strong solution of lithium bromide in water, but it being still poor in refrigerant water, ξ_a will, therefore, denote the concentration of water in the solution leaving the generator. The corresponding lithium bromide concentration will then be $(1 - \xi_r)$ and $(1 - \xi_a)$ respectively.

The salt does not exert any vapour pressure. So the vapour leaving the generator is a pure refrigerant (water vapour). Therefore, the analyser and dephlegmator do not form a part of the system. The equipment is normally designed for chilled-water applications with a flash system as shown in Fig. 12.14.

The generator and condenser are housed in the single-cylindrical vessel 1 and the flash evaporator and absorber in another similar vessel 6. Water is boiled off from vessel 1 by the steam coils 4. The vapour is condensed over the condenser *cooling water* coils 2 and collected in the tray 3. The condensate is flashed through expansion valve 5 into the vessel 6. The refrigerant water is collected in the tray 8. Chilled water is circulated by the pump 12 and is returned to the system at 13.

The strong brine from vessel 1 flows by gravity through the heat exchanger 7 and the pressure reducing valve 9 to the vessel 6. The flashed water vapour filling the space in 6 is absorbed by this solution. The absorber heat is removed by the cooling coils 10. Again, there is separate cooling water line for the absorber. The weak salt solution leaving the absorber is then returned to the generator by the pump 11 through the liquid-liquid regenerative heat exchanger 7.

Thus, we see that there are three kinds of water flowing in the system:

(i) *Refrigerant water*,
(ii) *Chilled water* (secondary refrigerant), and
(iii) *Cooling water.*

The chilled water, used as a secondary refrigerant, and refrigerant water are kept separate. If refrigerant water itself is used as chilled water, and it goes to the air conditioning plant and returns to the *sealed system*, it will bring in air with it. And thus it will break the vacuum.

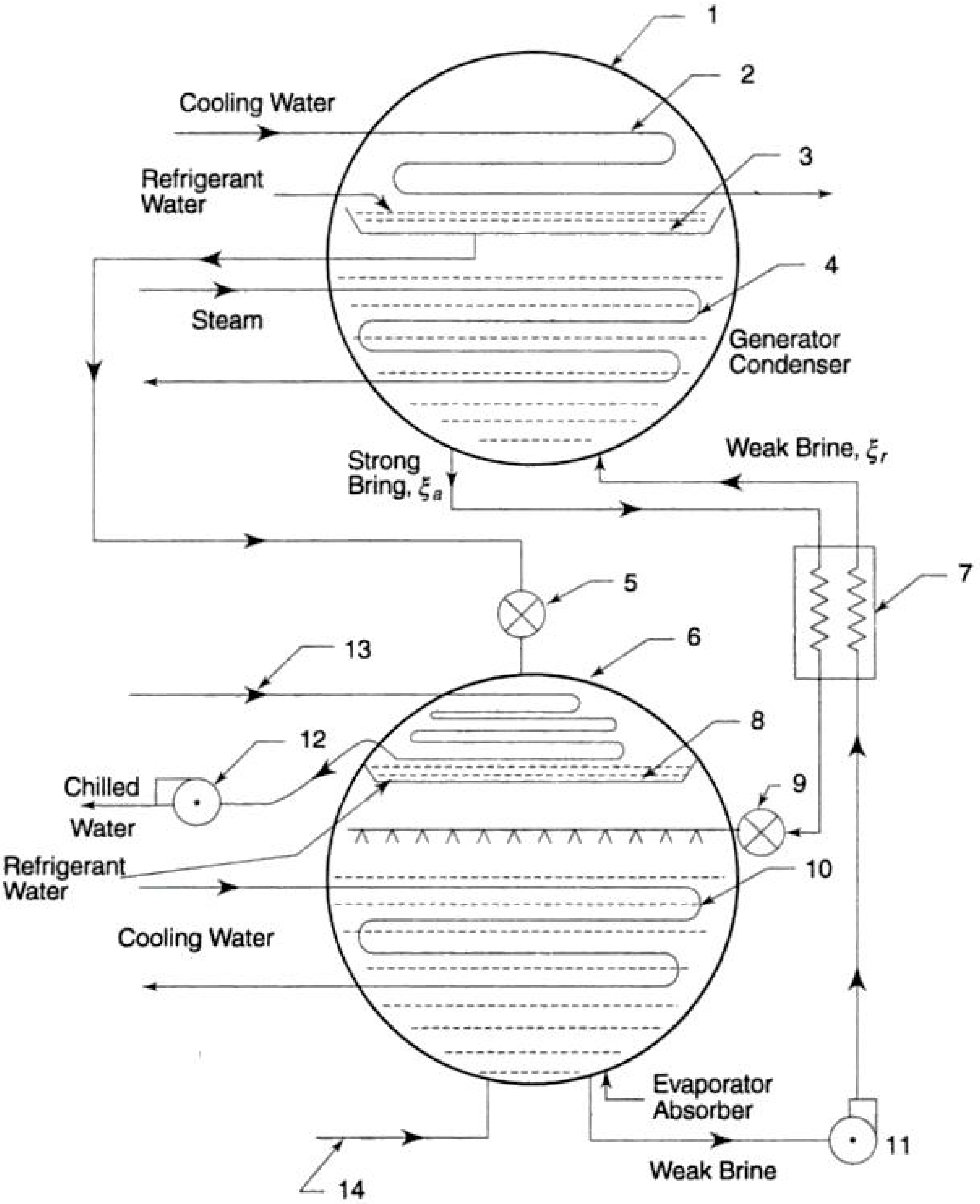

Fig. 12.14 Single-Effect Water-Lithium Bromide Absorption Chiller

Both the vessels 1 and 6 are maintained under high vacuum, vessel 1 corresponding to the condensing temperature (e.g., 55.3 mm Hg pressure at 40°C) and vessel ^{6} corresponding to the flashed refrigerant water temperature (e.g., 4.9 mm Hg pressure at 1°C). To remove air and other non-condensables that may enter the sealed system through pump glands, a two-stage *purge unit* is provided. To avoid corrosion, the temperature in the boiler should not be higher than 120°C. The overall COP of the system is reported to be approximately 0.7. The lithium bromide-water system is thus found to be more suitable in applications involving low heat-source temperatures such as are obtained with low-pressure (1 to 8 bar) or even exhaust (say 0.4 bar) steam, waste heat, solar energy, etc.

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Net heat rejected = $Q_C + Q_A = 3.77 + 4.35 = 8.12$ kW

Again, the energy balance checks very closely.

12.8 DOUBLE-EFFECT H_2O – $LiBr_2$ ABSORPTION SYSTEM

A single-stage like the single-effect absorption system is not suited to utilize a heat source at a temperature higher than a certain point unlike other heat-operated refrigerating machines that follow the Carnot trend, viz., the higher the temperature of the heat source, the higher the COP. In fact, the COP decreases as the heat source temperature increases beyond a point. This is because the absorption system is not a reversible refrigerating machine. Because of the mixing process of refrigerant and absorbent, a degree of irreversibility is involved. That is why, the COP of an absorption system levels with the increase in generator temperature, and then it starts decreasing.

It is found that the single-effect system gives best results upto a heat source temperature of 105°C. Above that temperature, it is worthwhile to switch over to double-effect system as illustrated in Figs 12.15 and 12.16.

Example 12.5 Calculations for Double-Effect H_2O-$LiBr_2$ System

The high pressure generator of a double-effect H_2O-$LiBr_2$ vapour absorption system, shown in Fig. 12.15 operates on steam supplied at 8 bar pressure. The following conditions are specified:

Pressure in high-pressure generator	100 kPa
Temperature in high-pressure generator	150°C
Condenser temperature	40°C
Absorber temperature	35°C
Evaporator temperature	10°C

The water vapour from the high-pressure generator I is condensed in the low-pressure generator II. The temperature of solution leaving generator II is 95°C.

(i) Show all the thermodynamic states on $\ln p - 1/T$ diagram.

(ii) Find enthalpies at all state points.

(iii) Determine the heat added in generator I per kg of water in refrigerant circuit.

(iv) Determine COP.

(v) What would have happened if a single-effect system were to be used?

Solution

(i) The thermodynamic states are shown on $\ln p - 1/T$ diagram in Fig. 12.16. All states are saturation states except 3a and 4a.

(ii) Condenser pressure at 40°C = 55.32 mm Hg

Evaporator pressure at 10°C = 9.27 mm Hg

Solution Circuit (Enthalpy is found from composition and temperature)

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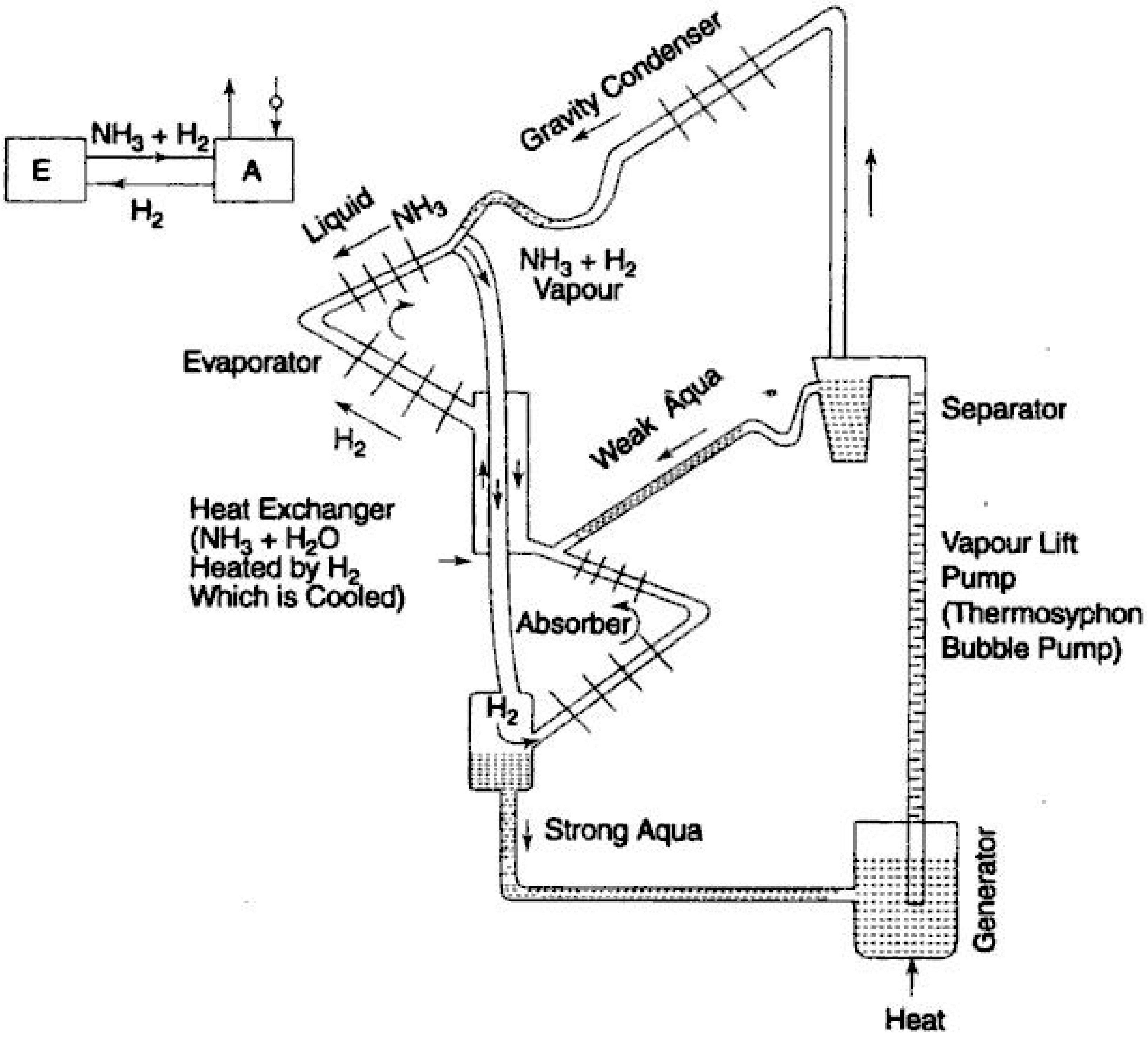

Fig. 12.17 Electrolux Refrigerator

Partial pressure of H_2 provides the pressure difference of NH_3 between the condenser and the evaporator. Accordingly, we have:

In condenser Pure NH_3 vapour pressure = Total pressure

In evaporator NH_3 vapour pressure = Total pressure – Partial pressure of H_2 For example, consider the condenser temperature as 50°C, and evaporator temperature as – 15°C. The corresponding vapour pressures of NH_3 are:

$$\text{Condenser}, p_k = 20.33 \text{ bar}$$

$$\text{Evaporator outlet}, p_{0_2} = 2.36 \text{ bar}$$

The approximate pressures in various parts of the system, then will be as given in Table 12.2.

Table 12.2 Partial Pressures in Electrolux Refrigerator in bar

Section	NH_3	H_2O	H_2	Total
Condenser	20.33	0	0	20.33
Evaporator inlet	1.516	0	18.814	20.33
Evaporator exit	2.36	0	17.97	20.33
Generator top	15.54	4.79	0	20.33

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Find the temperature and concentration of the vapour leaving the generator-analyser. Assume the specific heats of the rich and poor solutions as same. Calculate the cooling energy ratio for a system in which there is no rectifier.

(b) Calculate the same if there is a rectifier which cools the vapours to 60°C and the evaporator pressure is raised to 3.4 bar.

12.3 In a lithium bromide-water system the condenser and evaporator temperatures are 35 and 8°C respectively. The generator temperature is 85°C and the absorber temperature is 30°C. Assume a pressure drop of 2.5 mm Hg between the generator and condenser and 1 mm Hg between the evaporator and absorber. Determine the heat rejection rates in the condenser and absorber per unit of refrigerating capacity. Also find the heat input to the generator, and COP.

12.4 Let an analyser be added in the system of Prob. 12.1, and do the calculations for (a) to (e).

12.5 For an NH_3 absorption cycle, we have:

Condenser pressure	14 bar
Evaporator pressure	1.4 bar
Absorber temperature	36°C
Generator temperature	110°C

(a) Find out if such a cycle is possible or not.

(b) Change the evaporator pressure to 2.8 bar. Check if the cycle is possible now.

(c) Assume temperature leaving evaporator is $t_{0_2} = 5°C$. What is the refrigerating effect?

(d) Also find the COP.

(e) Add analyser in the system, and find COP.

(f) Add a dephlegmator which cools the vapour to 40°C. Determine the effect on COP.

12.6 (a) Find the temperature range through which liquid ammonia containing (a) 1% water, (b) 10% water will evaporate at a pressure of 2 bar.

(b) It is proposed to design a solar refrigeration NH_3-H_2O vapour absorption system operating under the following conditions:

Condenser pressure	16 bar
Evaporator pressure	2 bar
Absorber temperature	35°C

Hot water from flat plate solar collector can be obtained at a temperature of 80°C at the most. Examine the feasibility of operating the refrigeration system with this hot water.

(c) If not feasible, then suggest suitable measures to make it feasible.

12.7 (a) The operating conditions in a $H_2O - LiBr_2$ vapour absorption system are as follows:

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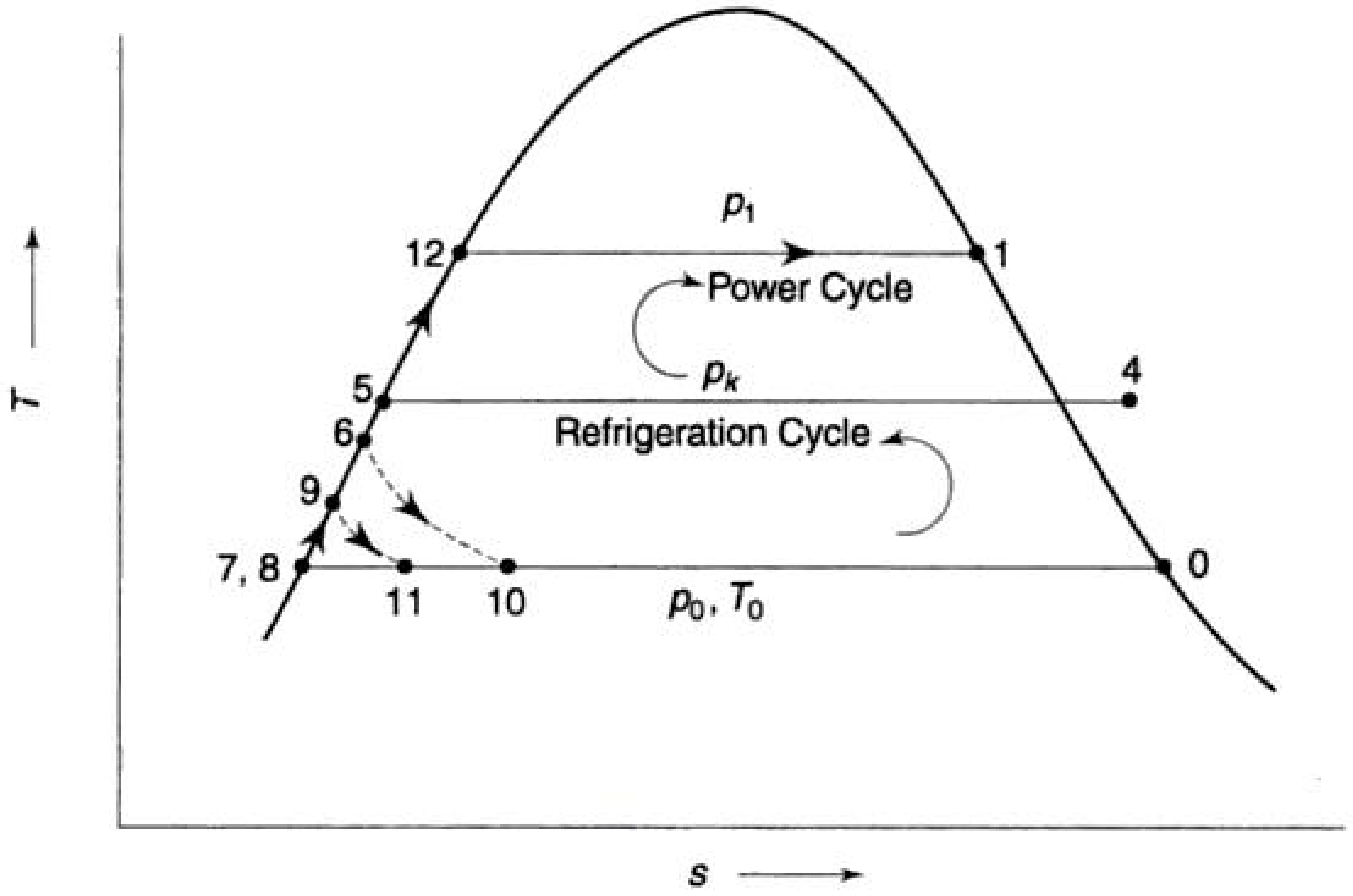

Fig. 13.2 Thermodynamic Cycle of Water-Vapour Refrigeration System

Note *✍: The control of capacity in the steam ejector system is possible either by throttling the steam inlet pressure, or by providing more than one nozzle operating in parallel, each driving a certain amount of vapour from the flash chamber.*

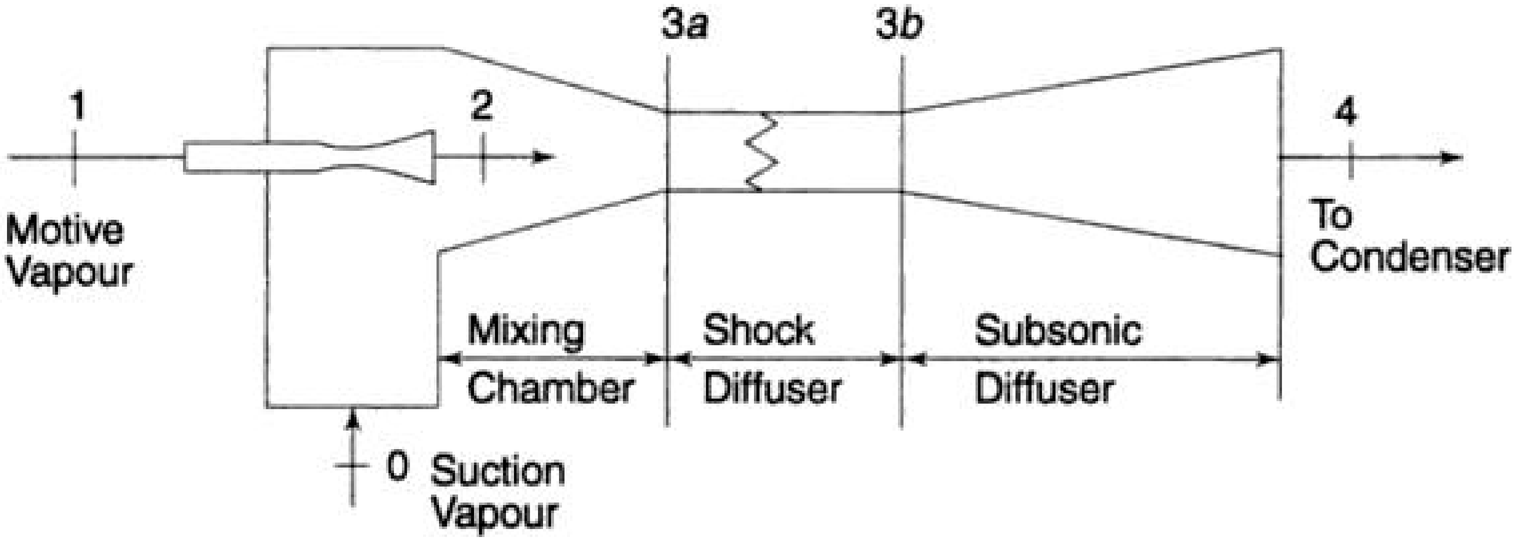

Fig. 13.3 Schematic Diagram of Steam Ejector

13.3 THEORETICAL ANALYSIS OF THE STEAM EJECTOR

It can be seen that because of the extremely low pressure in the flash chamber, the suction vapour consists of a very large volume which makes even a centrifugal compressor uneconomical to use. The suction vapour, therefore, in water vapour refrigeration is invariably compressed by an ejector using steam as the motive vapour.

A schematic steam ejector is shown in Fig. 13.3 and pressure variation along its length in Fig. 13.4. The high-pressure motive vapour at 1 expands to a pressure slightly above the pressure of the suction vapour at 0. The high velocity jet

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Thus, we find that to keep the mass ratio low, and to maintain the COP high, the enthalpy h_1 and hence the temperature and pressure of the motive vapour should be as high as possible. We note the following:

Refrigerating effect, $q_0 = h_0 - h_9$
Heat supplied, $q_h = m\,(h_1 - h_5)$

$$\text{COP} = \frac{q_0}{q_h} = \frac{h_0 - h_9}{m(h_1 - h_5)}$$

Example 13.1 In a steam-jet refrigeration system, the motive vapour is saturated at 150.3°C, and the chilled water temperature is 6°C. The mass ratio of the motive vapour to refrigerant vapour is 2.5. Find the saturated discharge temperature of the ejector. Assume the nozzle, entrainment and diffuser efficiencies as 0.85, 0.65 and 0.8 respectively.

Solution

Nozzle From the steam tables

$p_1 = 0.048$ bar at 150.3°C
$p_0 = 0.00935$ bar at 6°C
$h_1 = 2745.7$ kJ/kg
$s_1 = 6.833$ kJ/kg K

At 6°C,

$s_f = 0.091$ kJ/kg K
$s_g = 9.001$ kJ/kg K
$h_f = 25.2$ kJ/kg K
$h_g = 2512.6$ kJ/kg $= h_0$

After isentropic expansion, the dryness fraction is

$$x'_2 = \frac{6.833 - 0.091}{9.001 - 0.091} = 0.7567$$

Enthalpy after expansion

$$h'_2 = 25.2 + 0.7567\,(2512.6 - 25.2) = 1907.4 \text{ kJ/kg}$$

Velocity of motive vapour leaving nozzle

$$C_2 = \sqrt{2\eta_n(h_1 - h_2)}$$
$$= \sqrt{(0.85)(2)\,(2745.7 - 1907.4)10^3}$$
$$= 1193.8 \text{ m/s}$$

Mixing Section The momentum equation gives

$$\eta_e m C_2^2 = (m + 1)C_{3a}^2$$

$$0.65(2.5)(1193.8^2) = (2.5 + 1)C_{3a}^2$$

$$\Rightarrow \qquad C_{3a} = 813.4 \text{ m/s}$$

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6. Martinovsky V S, 'Use of waste heat for refrigeration, *Kholodilnaya Tekhnika*, Vol. 30, No. 1, Jan./March, 1953, p. 60.

PROBLEMS

13.1 (a) In a steam jet refrigeration system, the evaporator temperature is 5°C and the condensing temperature is 35°C. The motive vapour is dry saturated steam at 120°C. Assuming ideal processes, find the mass ratio of the motive vapour to the refrigerant vapour.

(b) Find the same if the nozzle, entrainment and diffuser efficiencies are 0.85, 0.65 and 0.8 respectively. Assume no shocks.

13.2 Repeat the calculations in Problem 13.1 (b) above assuming a normal shock in the constant area section.

13.3 Find the dimensions of the various sections of the ejector in Problem 13.2 for a 5,000-ton refrigerating capacity.

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If one were to measure the pressures exerted by individual gases, they would be found to be $p_1, p_2, \cdots$ etc., viz., less than the total pressure p of the mixture. These are referred to as *partial pressures*. Considering mixture and each component gas existing separately at T and V, we have for a binary gas mixture:

Mixture $\qquad pV = n\,\bar{R}\,T \qquad n = \dfrac{pV}{\bar{R}T}$

Components $\qquad p_1 V = n_1\,\bar{R}\,T \qquad n_1 = \dfrac{p_1 V}{\bar{R}T}$

$\qquad p_2 V = n_2\,\bar{R}\,T \qquad n_2 = \dfrac{p_2 V}{\bar{R}T}$

Substitution in $n_1 + n_2 = n$ gives

$$p = p_1 + p_2 \text{ or } p = \Sigma_i\, p_i \tag{14.3}$$

Thus, for a mixture of ideal gases, the total pressure p is equal to the sum of the partial pressures. This is known as the *Dalton's law* of *partial pressures*.

14.2.2 Amagat Law of Partial Volumes

In the Amagat model, each component gas is considered as existing separately at the total pressure p and temperature T of the mixture as shown in Fig. 14.1 (c).

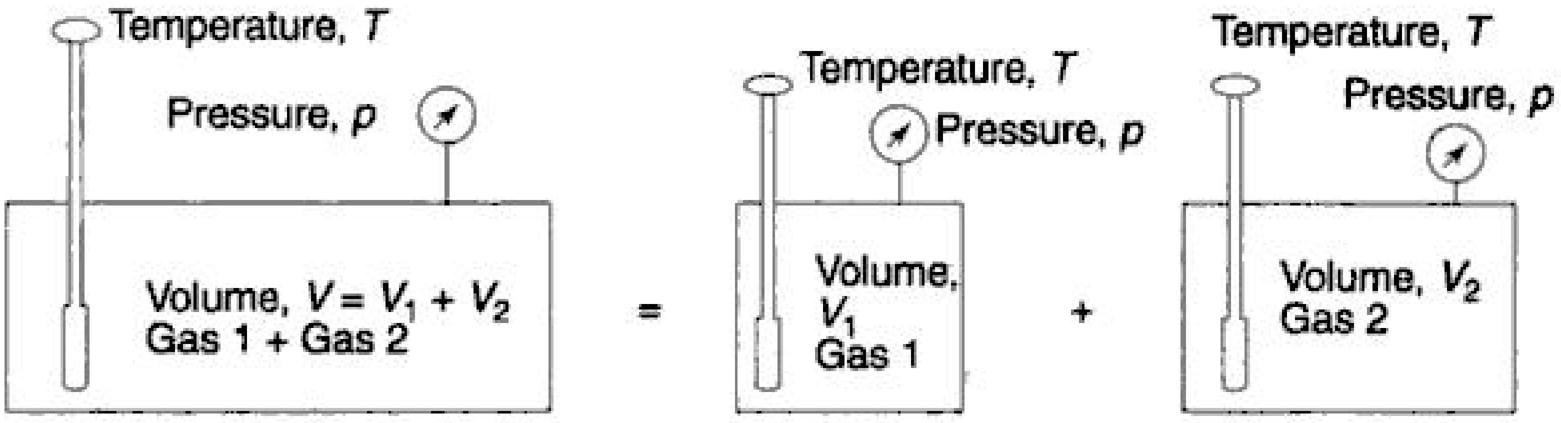

Fig. 14.1 (c) Figure Illustrating Amagat Model

Let the volumes of individual gases under these conditions be $V_1, V_2, \cdots$ etc. These are referred to as *partial volumes*. Again, applying the ideal gas equation of state to mixture and components, we have:

Mixture $\qquad pV = n\,\bar{R}\,T \qquad n = \dfrac{pV}{\bar{R}T}$

Components $\qquad p\,V_1 = n_1\,\bar{R}\,T \qquad n_1 = \dfrac{pV_1}{\bar{R}T}$

$\qquad p\,V_2 = n_2\,\bar{R}\,T \qquad n_2 = \dfrac{pV_2}{\bar{R}T}$

Substitution in $n_1 + n_2 = n$ gives

$$V = V_1 + V_2 \text{ or } V = \Sigma_i\, V_i \tag{14.4}$$

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pure substance which requires only two. One of the three properties can be the composition. The properties of moist air are called *psychrometric properties* and the subject which deals with the behaviour of moist air is known as *psychrometry*.

Water vapour is present in the atmosphere at a very low partial pressure. At this low pressure and atmospheric temperature, the water vapour behaves as a perfect gas. The partial pressure of dry air is also below one atmosphere which may also be considered to behave very much as a perfect gas. The Gibbs-Dalton laws of perfect gas mixtures can, therefore, be applied to the case of moist air.

In air-conditioning practice, all calculations are based on the dry air part since the water vapour part is continuously variable. For defining and calculating the relevant psychrometric properties, we may consider a certain volume V of moist air at pressure p and temperature T, containing m_a kg of dry air and m_v kg of water vapour as shown in Fig. 14.2. The actual temperature t of moist air is called the *dry bulb temperature* (DBT). The total pressure p which is equal to the *barometric pressure* is constant. The other relevant properties will now be discussed.

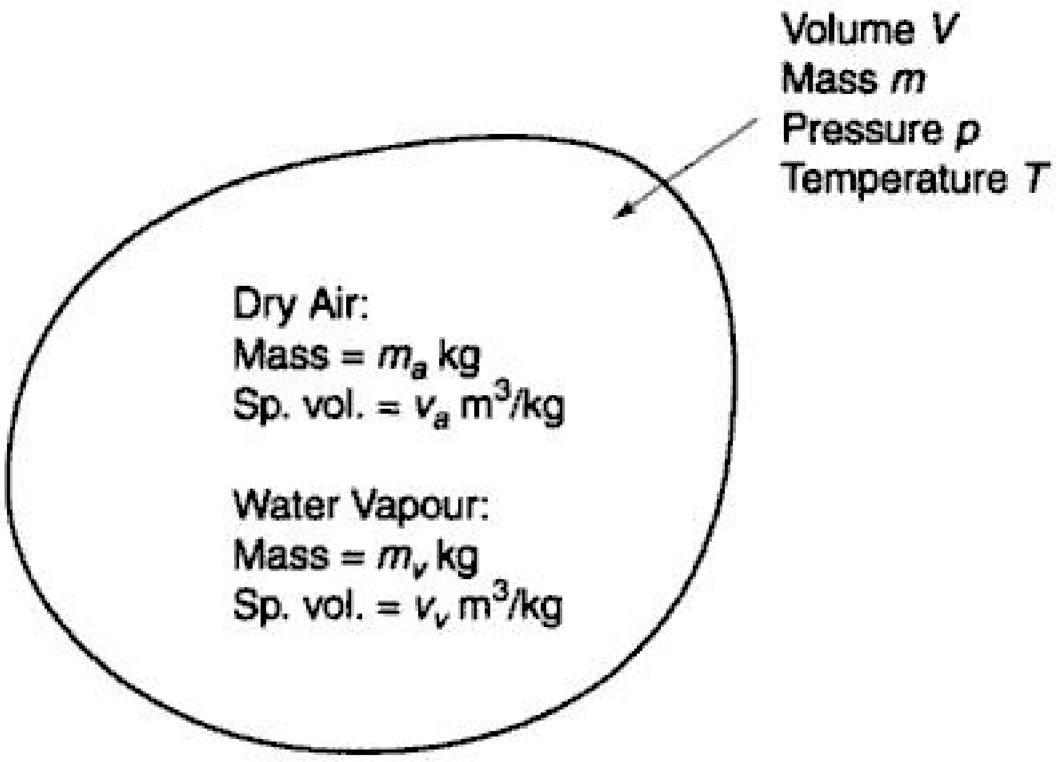

Fig. 14.2 A Mixture of Dry Air and Water Vapour (Moist Air)

14.3.1 Specific Humidity or Humidity Ratio

Specific or *absolute humidity* or *humidity ratio* or *moisture content* as it is variously called denoted by the symbol ω is defined as the ratio of the mass of water vapour (w.v.) to the mass of dry air (d.a.) in a given volume of the mixture. Thus

$$\omega = \frac{m_v}{m_a} = \frac{V/v_v}{V/v_a} = \frac{v_a}{v_v} \qquad (14.11)$$

where the subscripts a and v refer to dry air and water vapour respectively.

Now

$$p_a\, v_a = \frac{\overline{R}}{M_a}\, T \qquad p_a\, V = m_a \frac{\overline{R}}{M_a}\, T \qquad (14.12)$$

$$p_v\, v_v = \frac{\overline{R}}{M_v}\, T \qquad p_v\, V = m_v \frac{\overline{R}}{M_v}\, T \qquad (14.13)$$

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14.6.3 Constant Specific Volume Lines

Consider a line corresponding to a specific volume of, say, 0.85 m^3/kg d.a. as shown in Fig. 14.12. One point on this line is A which is on the saturation curve. To locate this point, the following trial and error procedure is to be followed:

(i) Assume t at A and find $p_s = p_v$.
(ii) Find p_a.
(iii) Calculate v and check.

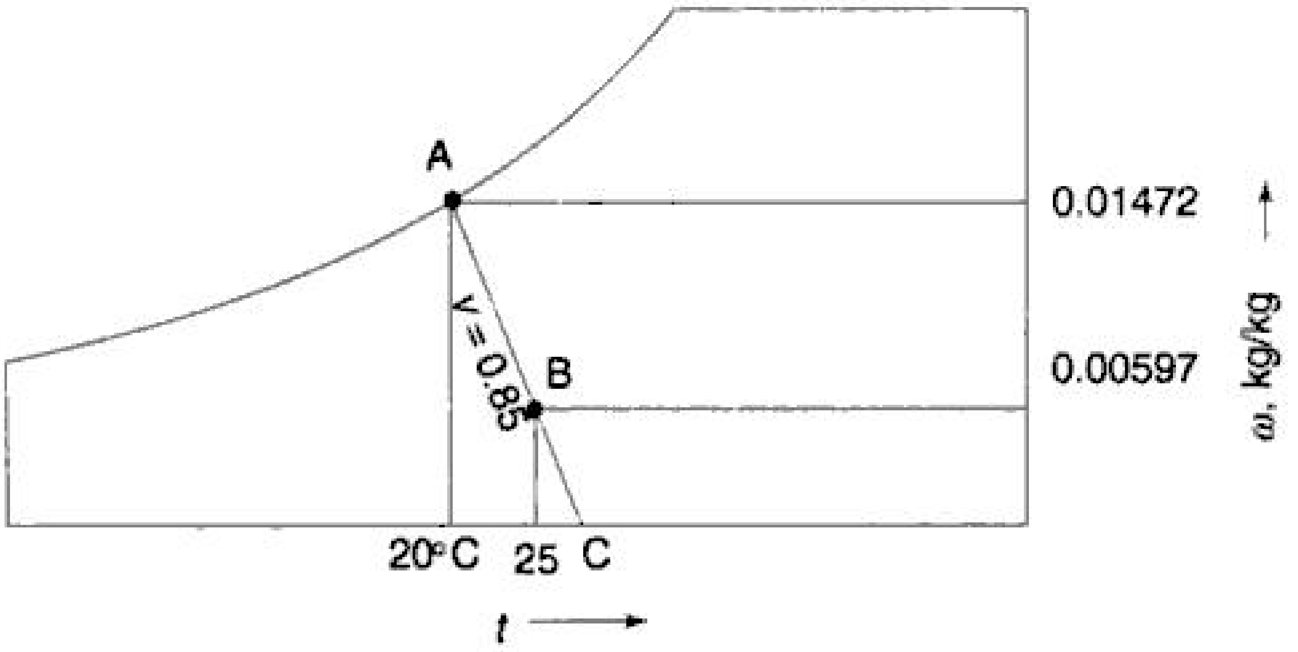

Fig. 14.12 Drawing 0.85 m^3/kg d.a. Constant-Volume Line

In this case, an assumed value of $t = 20°C$ is taken initially at which

$$p_s = p_v = 2342 \text{ N/m}^2$$
$$p_a = 101325 - 2342 = 98983 \text{ N/m}^2$$
$$v_a = R_a T/p_a$$
$$= \frac{287.3\,(273 + 20)}{98983} = 0.85 \text{ m}^3\text{/kg d.a.}$$

It checks with the required value. If it does not agree with the required value, a new value of t may be assumed. Now to plot point A, we calculate

$$\omega = \frac{0.622\,(2342)}{98983} = 0.01472 \text{ kg w.v./kg d.a.}$$

Thus point A can be plotted. Consider now another point B on this line at a DBT of 25°C. For this point with the same specific volume

$$p_a = \frac{R_a\,T}{v_a} = \frac{287.3\,(273 + 25)}{0.85} = 100724 \text{ N/m}^2$$
$$p_v = 101325 - 100724 = 601 \text{ N/m}^2$$
$$\omega = \frac{0.622\,(601)}{100724} = 0.00597 \text{ kg w.v./kg d.a.}$$

Thus point B can also be plotted. Likewise, a number of other unsaturated states can be plotted up to $\omega = 0$, to give the complete constant-volume line. It may be noted that at any point on this line

$$v = v_a = \omega v_v$$

14.6.4 Constant Thermodynamic Wet Bulb Temperature Lines

Consider the energy balance equation (14.41) for the adiabatic saturation process, viz.,

$$h + (\omega^* - \omega)h_f^* = h^*$$

Rearranging, we have

$$h - \omega h_f^* = h^* - \omega^* h_f^* = \text{const.}$$

Now in Fig. 14.13, point A at 20°C DBT also has a WBT of 20°C. For any other point on the 20°C constant WBT line, we must have

$$\Sigma = h - \omega h_f^* = h^* - \omega^* h_f^* = \Sigma^* = \text{const.} \tag{14.50}$$

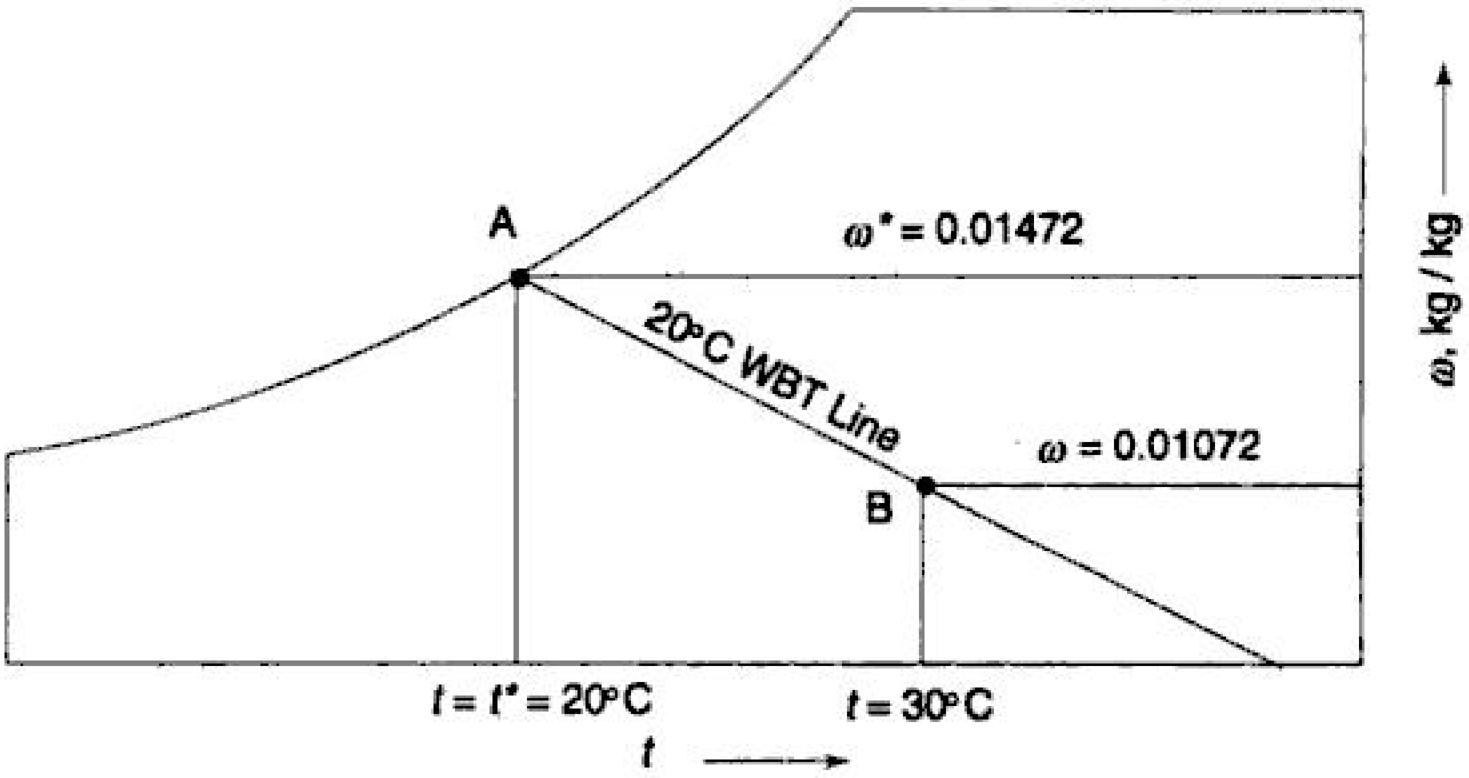

Fig. 14.13 Drawing 20°C Constant WBT Line

where Σ is called the *sigma heat function* which is constant along the constant WBT line. All states on this constant wet bulb temperature or constant sigma heat function line have a combination of h and ω which must satisfy Eq. (14.50). Now for point A

$$\omega^* = 0.01472 \text{ kg w.v./kg d.a.}$$

$$\begin{aligned} h^* &= (1.005 + 1.88\ \omega^*)\ t^* + 2501\ \omega^* \\ &= (1.005 + 1.88 \times 0.01472)20 + 2500\ (0.01472) \\ &= 57.45 \text{ kJ/kg d.a.} \end{aligned}$$

$$h_f^* = 83.9 \text{ kJ/kg (at 20°C)}$$

$$\begin{aligned} \Sigma^* &= h^* - \omega^*\ h_f^* \\ &= 57.45 - 0.01472\ (83.9) \\ &= 57.45 - 1.24 = 56.21 \text{ kJ/kg d.a.} \end{aligned}$$

Then for point B, say at 30°C

$$\Sigma = h - \omega h_f^* = \Sigma^* = 56.21$$

$$\Rightarrow \quad (1.005 + 1.88\ \omega)\ 30 + 2501\omega - 83.9\ \omega = 56.21$$

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PROBLEMS

14.1 For a dry bulb temperature of 25°C and a relative humidity of 50 per cent, calculate the following for air, when the barometric pressure is 740 mm Hg.

(a) Partial pressures of water vapour and dry air.
(b) Dew point temperature.
(c) Specific humidity.
(d) Specific volume.
(e) Enthalpy.

14.2 A sample of moist air has a dry bulb temperature of 43°C and a wet bulb temperature of 29°C. Calculate the following without making use of the psychrometric chart:

(a) Partial pressure of water vapour.
(b) Specific humidity.
(c) Relative humidity.
(d) Dew point temperature.
(e) Humid specific heat.
(f) Enthalpy.

Solution

For the first stream

$$\omega_1 = 0.0035 \text{ kg w.v./kg d.a.}$$

$$h_1 = 10.78 \text{ kJ/kg d.a.}$$

Mass of dry air per unit mass of moist air

$$m_{a_1} = \frac{1}{1+\omega_1} \text{ kg}$$

$$= \frac{1}{1.0035} = 0.9955 \text{ kg}$$

For the second stream

$$\omega_2 = 0.00765 \text{ kg w.v./kg d.a.}$$

$$h_2 = 49.67 \text{ kJ/kg d.a.}$$

Mass of dry air per two-units mass of moist air

$$m_{a_2} = \frac{2}{1+\omega_2} \text{ kg}$$

$$= \frac{2}{1.00765} = 1.9848 \text{ kg}$$

For the mixture

$$h = \frac{0.9965\,(10{\cdot}78) + 1.9848\,(49.67)}{0.9965 + 1.9848}$$

$$= 36.68 \text{ kJ/kg d.a.}$$

$$\omega = \frac{0.9965\,(0.0035) + 1.9848\,(0.00765)}{0.9965 + 1.9848}$$

$$= 0.00627 \text{ kg w.v./kg d.a.}$$

15.1.1 Mixing with Condensation

When a large quantity of cold air mixes with a quantity of warmer air at a high relative humidity, there is a possibility of condensation of water vapour, as seen in Fig. 15.2, and the mixture will then consist of saturated air and the condensate.

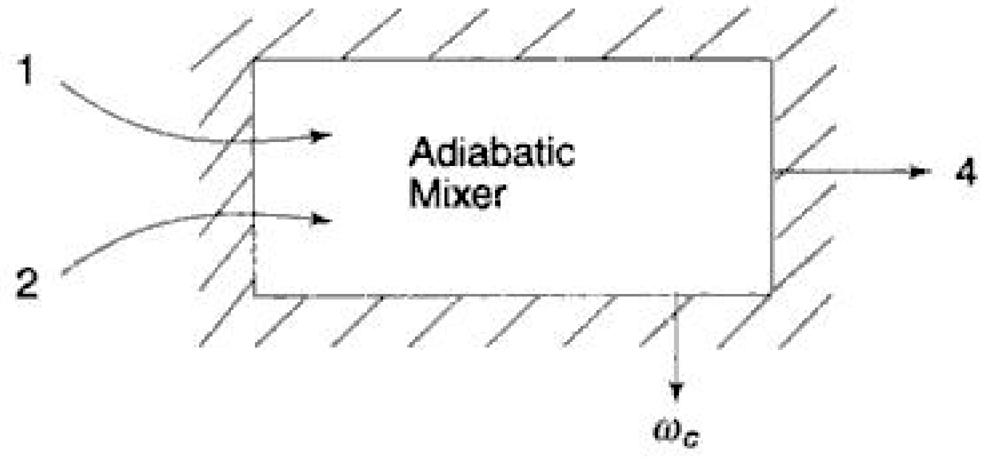

Fig. 15.2(a) Adiabatic Mixer with Condensation

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Substituting 2500 for h_{fg_0} in Eq. (15.10) and using the expression for m_a from Eq. (15.7b) we obtain

$$\dot{Q}_L = \frac{(\text{cmm})\,(1.2)\,(2501)}{60}\,\Delta\omega$$

$$= 50\,(\text{cmm})\,\Delta\omega,\ \text{kW}\ (= 0.68\,(\text{cfm})\,\Delta\omega\ \text{Btu/h}) \qquad (15.11)$$

In English units, $\Delta\omega$ is in grains/lbm d.a. Note that 7000 grains = 1 lbm.

15.2.3 Total Heat Process

Consider now a change in the state of air along the path AC as shown in Fig. 15.6. This involves both a change in temperature as well as in the humidity ratio. The change in temperature causes a sensible heat load given by

$$Q_S = m_a\,(h_B - h_A)$$

$$= m_a\,C_p\,(t_C - t_A) \qquad (15.12)$$

The change in the humidity ratio causes a moisture transfer given by

$$G = m_a\,(\omega_C - \omega_A)$$

and a latent heat load given by

$$Q_L = m_a\,(h_C - h_B)$$

$$= m_a\,h_{fg_0}\,(\omega_C - \omega_A) \qquad (15.13)$$

Adding Eqs (15.12) and (15.13) we obtain an expression for *total heat load* as

$$Q = Q_S + Q_L$$

$$= m_a\,(h_C - h_A) \qquad (15.14a)$$

$$= m_a\,[C_p\,(t_C - t_A) + h_{fg_0}\,(\omega_C - \omega_A)] \qquad (15.14b)$$

Again, expressing the mass flow rate in cmm, we get

$$\dot{Q} = \frac{(\text{cmm})\,(1.2)}{60}\,\Delta h$$

$$= 0.02\,(\text{cmm})\,\Delta h, \qquad (15.15a)$$

which is the same as

$$\dot{Q} = (\text{cmm})\,(0.0204\,\Delta t + 50\,\Delta\omega),\ \text{kW} \qquad (15.15b)$$

15.2.4 Sensible Heat Factor (SHF)

The ratio of the sensible heat transfer to the total heat transfer is termed as the *sensible heat factor*. Thus

$$\text{SHF} = \frac{Q_S}{Q_S + Q_L} = \frac{Q_S}{Q}$$

Substituting from Eqs (15.8) and (15.15), we obtain

$$\text{SHF} = \frac{h_B - h_A}{(h_B - h_A) + (h_C - h_B)} = \frac{h_B - h_A}{h_C - h_A}$$

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15.3.4 Air Washer

Figure 15.12 shows the schematic representation of an *air washer*. It involves the flow of air through a spray of water. During the course of flow, the air may be cooled or heated, humidified or dehumidified, or simply adiabatically saturated, depending on the mean surface temperature of water. The water is, accordingly, externally cooled or heated or simply recirculated by a pump. Make-up water is added for any loss in the case of humidification of air. Eliminator plates are provided to minimise the loss of water droplets.

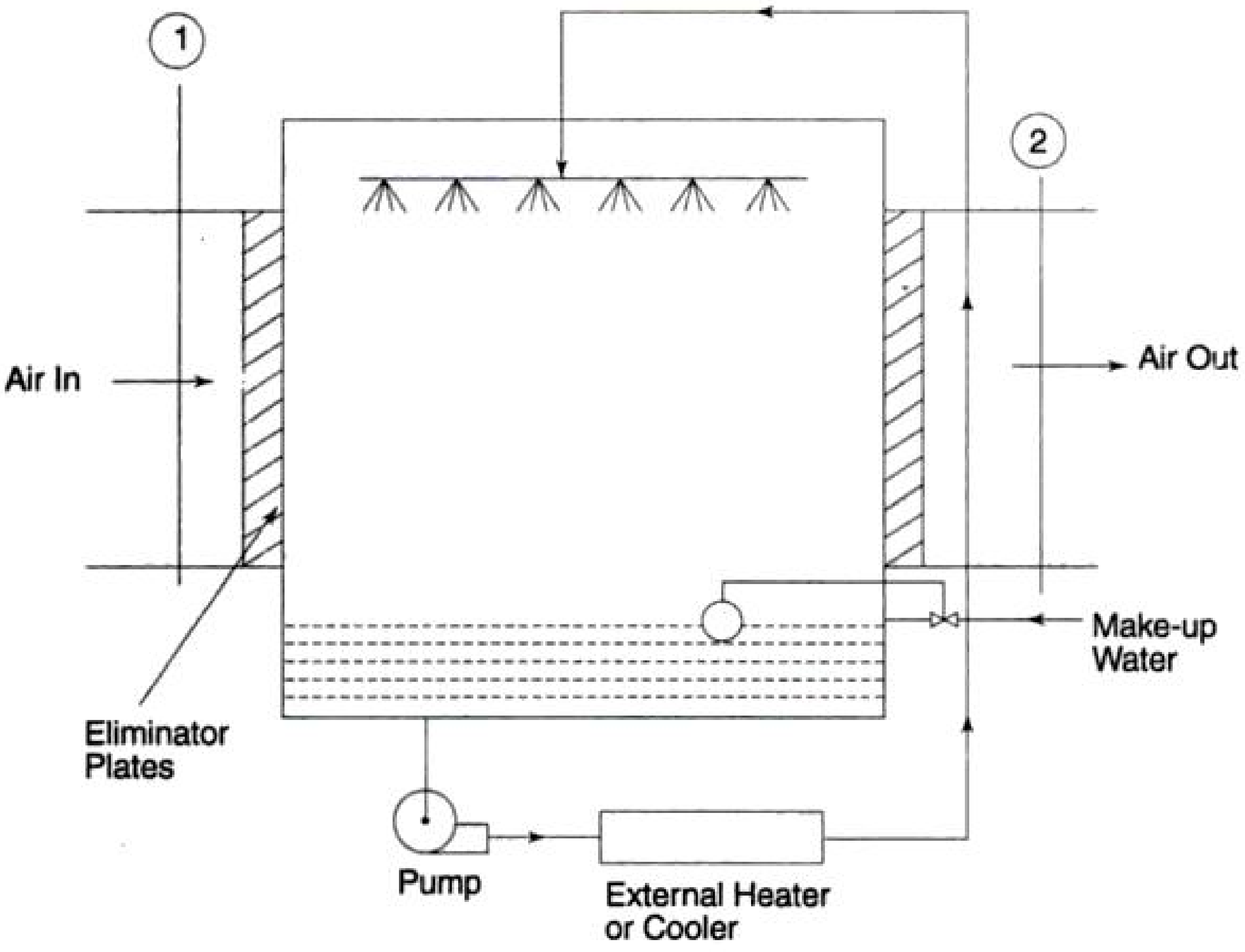

Fig. 15.12 Air Washer

Figure 15.13 shows the thermodynamic changes of state of air along paths 1-2 in an air washer, depending on the mean surface temperature of water droplets t_S which is equal to the actual temperature of water t_w.

Thus, the droplets of water act as wetted surface, and both sensible and latent heat transfers take place. Their directions depend on the temperature and vapour pressure potentials. The following processes are possible:

Process 1-2 A: *Heating and humidification* ($t_S > t_1$)

The mean surface temperature of water is greater than the dry bulb temperature of air. The water is externally heated.

Process 1-2B: *Humidification* ($t_S = t_1$)

The mean surface temperature of water is equal to the dry bulb temperature of air. The enthalpy of air increases. Hence the water is required to be externally heated.

Process 1-2C: *Cooling and humidification* ($t_1' < t_S < t_1$)

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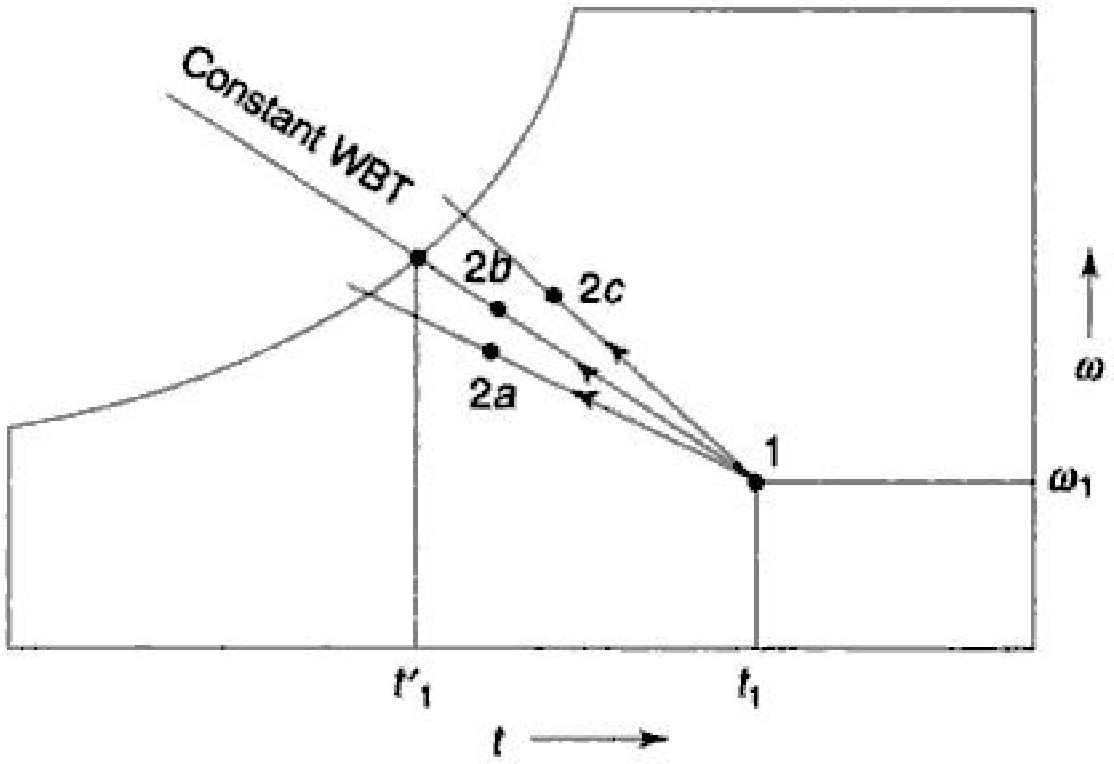

Fig. 15.17 Process with Liquid Water Injection

$$\omega_2 = \omega_1 + \frac{m_v}{m_a} \tag{15.20}$$

$$h_2 = h_1 + \frac{m_v}{m_a} h_f$$
$$= h_1 + (\omega_2 - \omega_1)h_f \tag{15.21}$$

where h_f is the enthalpy of liquid water. It is evident from Eq. (15.21) that if water is injected at the wet bulb temperature of the air, the sigma heat function is constant, and the process follows the constant WBT line 1-2b. Otherwise, the process follows line 1-2a or 1-2c, depending on whether the temperature of water is lower or higher than the WBT of air. Nevertheless since the term $(\omega_2 - \omega_1)\, h_f$ is extremely small compared to h_1 and h_2, lines 1-2a and 1-2c are very close to line 1-2b, irrespective of the temperature of the injected water.

Note ✍: *In recent years, this method of evaporative cooling has gained popularity specially in places where there is shortage of water since it eliminates at least 2-3% water loss which occurs in air washer equipment. Also, it provides for individual control of supply conditions for different spaces. And the equipment is simple and economical.*

15.3.8 Steam Injection

Steam is normally injected into fresh outdoor air which is then supplied for the conditioning of textile mills where high humidities have to be maintained. The process can be analysed by considering mass and energy balances. If m_v is the mass of steam supplied with enthalpy h_v and m_a the mass of dry air, then the leaving air state is given by

$$\omega_2 = \omega_1 + \frac{m_v}{m_a} \tag{15.22}$$

$$h_2 = h_1 + \frac{m_v}{m_a} h_v \tag{15.23}$$

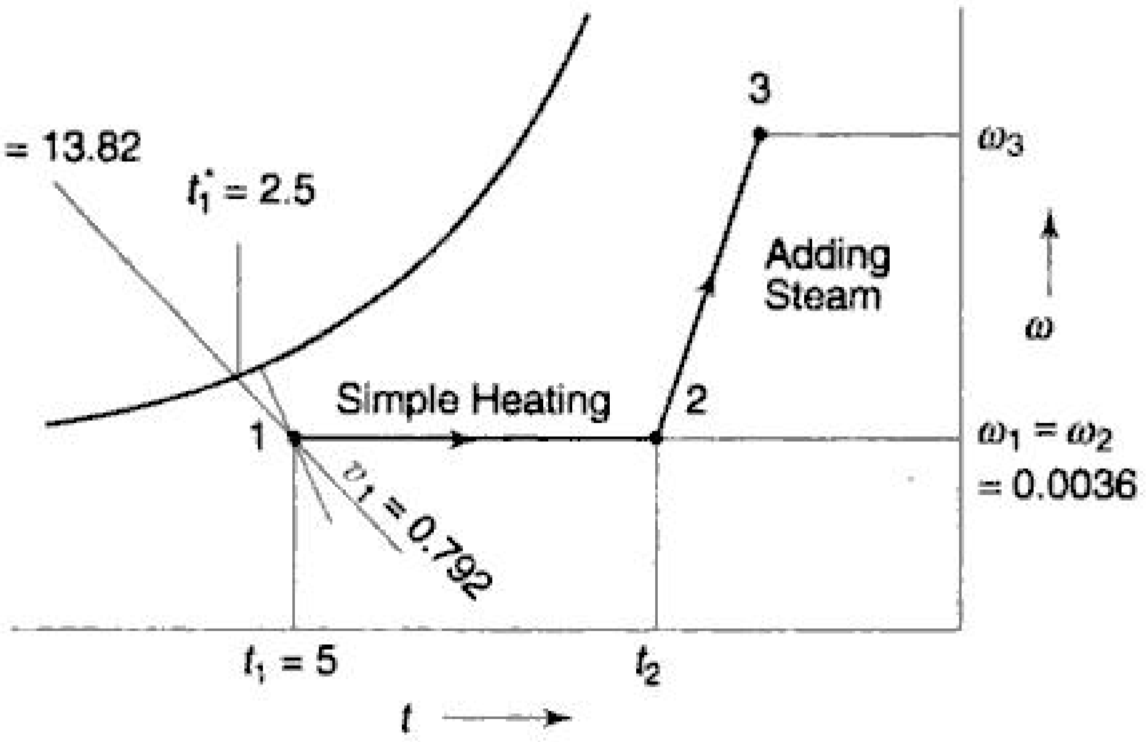

Fig. 15.19 (b) Psychrometric Processes for Example 15.4

where 0.792 m^3/kg d.a. is the specific volumes of the entering air.
By moisture balance

$$\dot{m}_a\ (\omega_3 - \omega_1) = 40$$

$$\omega_3 = \omega_1 + \frac{40}{\dot{m}_a} = 0.0036 + \frac{40}{6820}$$

$$= 0.00947 \text{ kg w.v./kg d.a.}$$

By energy balance

$$\dot{m}_a\ (h_3 - h_1) = (40.7)\ (3600) + 40\ h_v$$

where $h_v = 2691.3$ kJ/kg is the enthalpy of saturated steam at 110°C.
Thus

$$h_3 = 13.82 + \frac{1}{6820}\ [146{,}540 + 40\ (2691.3)]$$

$$= 51.1 \text{ kJ/kg d.a.}$$

From psychrometric chart, at 3

DBT = 26.5°C
WBT = 18.1°C

15.4 SIMPLE AIR-CONDITIONING SYSTEM AND STATE AND MASS RATE OF SUPPLY AIR

The problem of air-conditioning a space essentially reduces to the calculation of the state and mass rate of air to be supplied to the space-necessary to pick up its sensible and latent heat loads. For the simplest air-conditioning system, consider a space which is to be maintained at the room or inside conditions of, say, dry bulb temperature t_i and humidity ratio ω_i.

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$$\Rightarrow \qquad (m_{a_i})_1 = \frac{0.22\,\omega_1 - 3.41}{3.65 - \omega_1} \qquad \text{(I)}$$

Mixing after the dehumidifier

$$\left(\dot{m}_{a_1}\right)\,\omega_i + \dot{m}_{a_2}\,\omega_2 = \dot{m}_{a_s}\,\omega_s$$

$$[0.759 - (\dot{m}_{a_i})_1]\,(3.65) + [0.22 + (\dot{m}_{a_i})_1]\,\omega_2 = 0.759\,(3.3)$$

or

$$(\dot{m}_{a_i})_1 = \left(\frac{0.22\,\omega_2 + 0.195}{3.65 - \omega_2}\right) \qquad \text{(II)}$$

Equating (I) and (II), we have

$$\frac{0.22\,\omega_1 - 3.41}{3.65 - \omega_1} = \frac{0.22\,\omega_2 + 0.195}{3.65 - \omega_2} \qquad \text{(III)}$$

The relationship between the entering moisture content ω_1 and the leaving moisture content ω_2 is given in terms of the performance data. Solving Eq. (III) simultaneously or graphically with the tabulated data, we get

$$\omega_1 = 8.1 \text{ g w.v./kg d.a.}$$
$$\omega_2 = 1.96 \text{ g w.v./kg d.a.}$$

From Eq. (I)

$$(\dot{m}_{a_i})_1 = \frac{0.22(8.1) - 3.41}{3.65 - 8.1}$$
$$= 0.366 \text{ kg d.a./s}$$

and

$$(\dot{m}_{a_i})_2 = m_{a_i} - (m_{a_i})_1$$
$$= 0.759 - 0.366 = 0.393 \text{ kg d.a./s}$$

Dry air flow through the dehumidifer

$$\dot{m}_{a_1} = m_{a_s} - (m_{a_1})_2$$
$$= 0.979 - 0.393 = 0.586 \text{ kg d.a./s}$$

Heat liberated due to the condensation of moisture

$$\dot{Q}_{\text{cond}} = \dot{m}_{a_1}\,(\omega_1 - \omega_2)h_{fg}$$
$$= 0.586\left(\frac{8.1 - 1.96}{1000}\right)(2500) = 9.0 \text{ kW}$$

Heat of adsorption

$$\dot{Q}_{\text{ads}} = \dot{m}_{a_1}\,(\omega_1 - \omega_2)\,(390)$$
$$= 0.586\left(\frac{8.1 - 1.96}{1000}\right)(390) = 1.5 \text{ kW}$$

Temperature rise of air in the dehumidifier

$$t_2 - t_1 = \frac{\dot{Q}}{\dot{m}_a\,C_p}$$

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$$\text{RSHF} = \frac{14.7}{14.7+3} = \frac{14.7}{17.7} = 0.831$$

This is the slope of line *i–s–*2. However, it is not possible to fix points s and 2 at this stage. But, we know that

$$\text{BPF} = \frac{\text{Line } 2-\text{S}}{\text{Line } 0-\text{S}} = 0.1$$

Accordingly, point 2 can be fixed by trial and error on the RSHF line. However, the following construction is simpler and more accurate. Draw line *o–i* and extend it to *A* such that:

$$\frac{\text{Line } i-A}{\text{Line } o-A} = 0.1$$

And then draw AS parallel to RSHF line *i – s*. Intersection with saturation curve gives coil ADP of 11.1°C at *S*. Join 0 – *S*. It cuts *i–s* extended at 2. Thus, we get condition of air leaving coil as:

$$t_2 = 12.2°\text{C},\ \omega_2 = 8.5 \text{ g/kg d.a.},\ h_2 = 33.8 \text{ kJ/kg d.a.}$$

(i) DBT and moisture content of supply air

$$\dot{m}_{a_s} = \frac{\dot{Q}_{v_s}}{v_s} = \frac{191/60}{v_s}$$

$$\text{RTH} = \dot{m}_{a_s}(h_i - h_s) = \frac{191/60}{v_s}(48.8 - h_s) = 17.7$$

This equation can be solved by trial and error. Thus, point *s* can be located on RSHF line. It is found that:

$$v_s = 0.856 \text{ m}^3/\text{kg d.a.},\ t_s = 20°\text{C},$$
$$h_s = 43.8 \text{ kJ/kg d.a.},\ \omega_s = 9.1 \text{ g/kg d.a.}$$

(ii) Mass flow rate of moist air supplied to room

$$\dot{m}_s = \frac{191/60}{0.856}(1 + 0.0091) = 3.753 \text{ kg/s}$$

(iv) Fresh air through coil

$$\dot{m}_{a_2} t_2 + (\dot{m}_{a_s} - \dot{m}_{a_2}) t_i = \dot{m}_{a_s} t_1$$

$$\dot{m}_{a_2}(12.2) + \left(\frac{191}{0.856} - \dot{m}_{a_2}\right)(24) = \frac{191}{0.856}(20)$$

$$\Rightarrow \qquad \dot{m}_{a_2} = 76.1 \text{ kg/min} = 1.268 \text{ kg/s} = \dot{m}_{a_0}$$

Load on cooling coil

$$\text{GTH} = \dot{m}_{a_2}(h_0 - h_2) = 1.268\,(69.8 - 33.8) = 45.7 \text{ kW}$$

Example 15.10 Given for a conditioned space:

Room sensible heat gain = 20 kW

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(c) How much latent heat is picked up in the auditorium?
(d) What is the sensible heat factor?
(e) What is the ADP of the coil, and what is its BPF?

15.6 An air conditioned space is maintained at 25°C DBT and 50% RH. The outside conditions are 40°C DB and 25°C WB. The space has a sensible heat gain of 24.5 kW. Conditioned air is supplied to the space as saturated air at 10°C. The equipment consists of an air washer. The air entering the air washer comprises 25% outside air, the remainder being recirculated room air.

Calculate:
(i) Volume flow rate of air supplied to space.
(ii) Latent heat gain of space.
(iii) Cooling load of air washer.

15.7 Given for the air conditioning of a room
Room conditions: 26.5°C DBT and 50 per cent RH
Room sensible heat gain = 26.3 kW
Room sensible heat factor = 0.82

Find:
(i) The room latent heat gain.
(ii) The apparatus dew point.
(iii) The cmm of air if it is supplied to the room at the apparatus dew point.
(iv) The cmm and specific humidity of air if it is supplied to the room at 17°C.

15.8 (a) 28.5 cmm of room air at 25.5°C DBT and 50 per cent RH is mixed with 28.5 cmm of outside air at 38°C DBT and 27°C WBT. Find the ventilation load and the condition of air after mixing.
(b) The above mixture of air is passed through an air conditioning equipment. If the wet bulb temperature of air after the equipment is 14.5°C, determine the heat removed by the equipment.

15.9 An air washer cools and dehumidifies 18,200 kg of dry air per hour from 41°C DBT and 24°C WBT. Chilled water enters the washer at 7°C with a flow of 18,500 kg per hour. The washer is 88 per cent effective. What is the heat removed from the air in kW?

15.10 Saturated steam at standard atmospheric pressure is injected into a passing air stream in an amount sufficient to raise the absolute humidity from 0.0057 to 0.0143 kg w.v./kg d.a. If the air enters the humidifier at 21°C DBT, determine its leaving state.

15.11 300 cmm of outside air at 5°C and 60 per cent RH are heated and humidified to maintain room conditions at 21°C and 40 per cent RH. Devise a suitable system for the air conditioning and find the kW of various heaters and the moisture added. The supply air temperature is 45°C and the room sensible heat factor is 0.75.

According to Malhotra[5], the effective temperatures and also the range of DBT at 50 per cent RH for comfort are as follows:

Climate	*ET,* °C	*Corresponding DBT at 50% RH,* °C
Hot and dry	21.1 to 26.7	23.9 to
Hot and humid	22 to 25.6	26.7

The general practice is to recommend the following optimum inside design conditions for comfort for summer air conditioning:

ET	21.7°C
DBT	25 ± 1°C
RH	50 ± 5%

The corresponding room air velocity is 0.4 m/s. The points of this equal comfort are shown in Fig. 16.1.

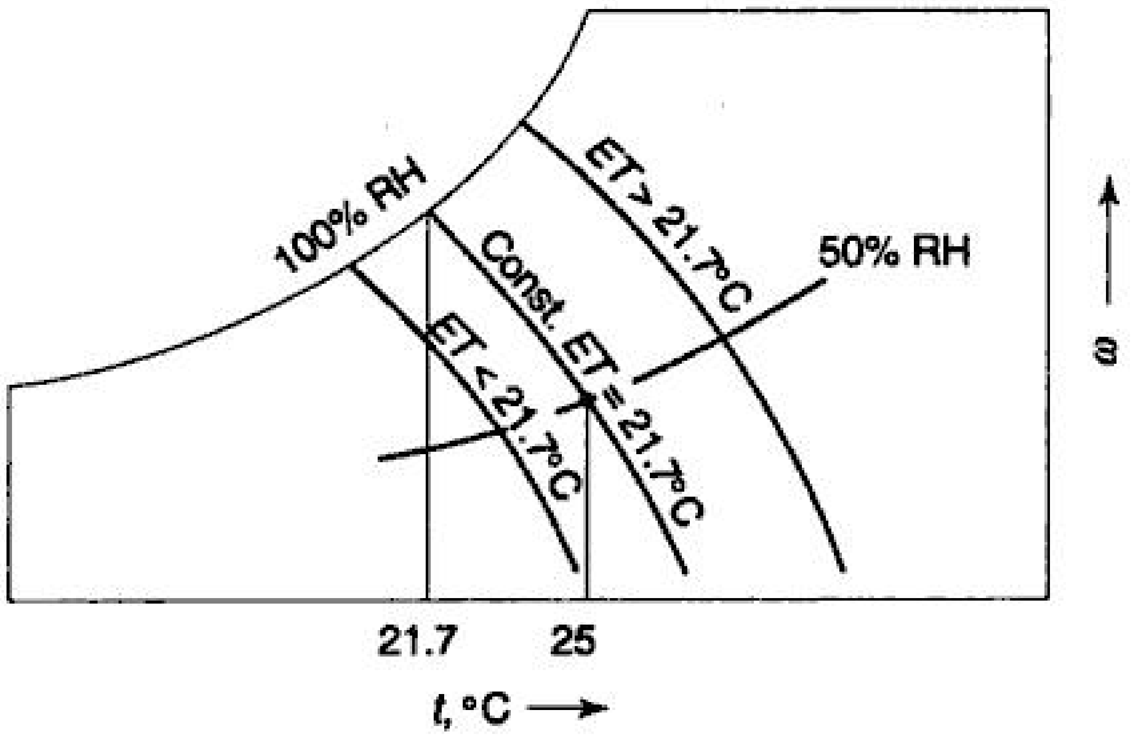

Fig. 16.1 Effective Temperature Lines

During winter, the body gets acclimatized to withstand lower temperatures. Consequently, a DBT of 21°C at 50 per cent RH and 0.15–0.2 m/s air velocity is quite comfortable.

In addition to the maintenance of temperature, humidity and air velocity, it is also important to maintain the purity of room air. Even if there are no sources of production of pollutants within the conditioned space, the carbon dioxide content of air increases because of the occupants. It is, therefore, necessary to introduce fresh air or *ventilation air* into the space. The requirement of *ventilation air* is much more when some occupants are smoking. In the case of auditoriums, because of very large occupancy, the ventilation air requirement is very large. Hence the need to prohibit smoking in auditoriums and assembly halls. Table 16.2 gives the ventilation air requirements for some applications.

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the room sensible heat factor line. Thus, thermodynamically, there is a wide choice available for the supply air condition. For each supply air condition, there is a supply air rate. Corresponding to the minimum weight of the supply air, the supply condition is at the apparatus dew point for summer, or slightly away from it in the case of a finite bypass factor of the conditioning apparatus. This minimum weight of the supply air may not be adequate from the point of view of good room air distribution aiming at equalizing both the temperature and humidity throughout the conditioned space, in addition to providing a certain air movement at the occupancy level. For good room air distribution, it is found necessary to supply the conditioned air equivalent to 8–12 air changes per hour. The normally acceptable value is 10 air changes per hour. A lower value will cause *stagnancy* whereas a higher value will cause *draft*, both leading to discomfort. Some consultants in India take supply air upto 15 air changes per hour as an insurance against dirty filters. This comes to about 2.5 cfm/ft^2 of floor area with a height of about 10 feet. In case of evaporative cooling, this value is close to 20 air changes per hour.

Corresponding to the specified amount of the quantity of supply air, the supply air temperature for summer air conditioning would be 12.5 to 7.5°C below the room temperature. For a room temperature of 25°C, the supply air temperature is close to 13 to 15°C.

In case, the minimum weight of the supply air does not meet the air change requirement, it is possible to increase the quantity of the supply air by mixing it with an additional amount of recirculated room air after the A/C apparatus as shown in Fig. 16.2.

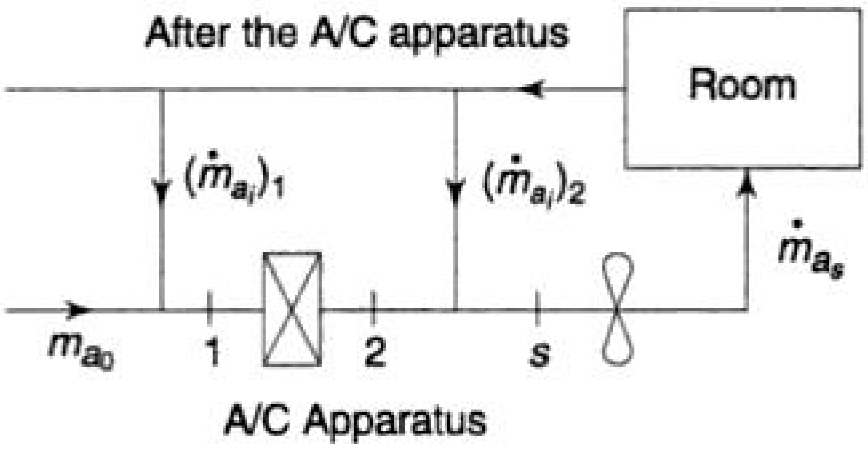

Fig. 16.2 Recirculated Room Air added after the A/C Apparatus

The effect of such mixing is merely to increase the bulk of the supply air, and correspondingly change its temperature and moisture content, but in no way does it affect the capacity of this air to meet the room sensible and latent heat loads. In such a case, the conditioned air quantity is $[(\dot{m}_{a_i})_1 + \dot{m}_{a_0}]$ but the supply air quantity becomes $[(\dot{m}_{a_i})_1 + \dot{m}_{a_0} + (\dot{m}_{a_i})_2]$. Accordingly, the supply air condition moves to *s* as shown in Fig. 16.3. The dehumidified air rise available is $t_i - t_s$.

Note ✍: *The supply air temperature for winter heating is kept at about 45-55°C.*

Example 16.1 An air-conditioned space is maintained at 26°C DBT and 50 per cent RH when the outside conditions are 35°C DBT and 28°C WBT.

(a) If the space has a sensible heat gain of 17.6 kW and air is supplied to the room at a condition of 8°C saturated, calculate:

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Chapter 17

Solar Radiation

17.1 DISTRIBUTION OF SOLAR RADIATION

Solar radiation forms the greatest single factor of cooling load in buildings. It is, therefore, necessary to study the subject not only for the purpose of load calculation, but also from the point of view of load reduction. Further, the subject has acquired new dimensions in the present-day world in the context of solar energy utilization for heating as well as cooling.

For all practical purposes, the sun is the source of energy for the continuation of life on earth. It is a sphere of intensely hot gaseous matter. It is a fusion reactor—the most important of its reactions is the combination of hydrogen to form helium, the difference in mass being converted to energy. This fusion energy is produced in the interior of the solar sphere at a temperature of many millions of degrees. The energy is transferred to the surface of the sun by radiation and convection. The surface is opaque. For all practical purposes, it may be considered to be radiating energy as a black body at an effective temperature of 6000 K. The spectrum of the wavelength of radiation stretches from 0.29 to 4.75 μ m. As a consequence of high temperature, the maximum radiation intensity is found to be at a wavelength of 0.5 μ m.

The mass of the sun is about 332,000 times that of the earth and its diameter is about 1,392,400 km. The earth is about 12,710 km in diameter. It makes one rotation about its axis in 24 hours, and a revolution around the sun in a period of approximately $365\frac{1}{4}$ days.

The earth revolves round the sun in an elliptical orbit. The earth is closest to the sun on January 1, and remotest from it on July 1 (about 3.3 per cent farther away). The mean distance of the earth from the sun is 149,500,000 km. The intensity of solar radiation outside the earth's atmosphere varies inversely with the square of the distance between the centre of the earth and the centre of the sun. Accordingly, the earth receives 7 per cent more radiation in January than in July.

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the angle between the sun's rays and the equatorial plane. It will be seen that the sun's declination is maximum at the solstice positions and is equal to 23.5° south ($d = -23.5°$) of the equatorial plane at the time of the winter solstice on December 21 and 23.5° north ($d = +23.5°$) of the plane at the time of the summer solstice on June 21. At the autumnal and vernal equinoxes it is equal to zero. The weekly variation of d over the year is given in Table 17.5. During the year d can be approximated by a sinusoidal variation

$$d = 23.47 \sin \frac{360\,(284 + N)}{365} \tag{17.4}$$

where N is the day of the year counted from January 1.

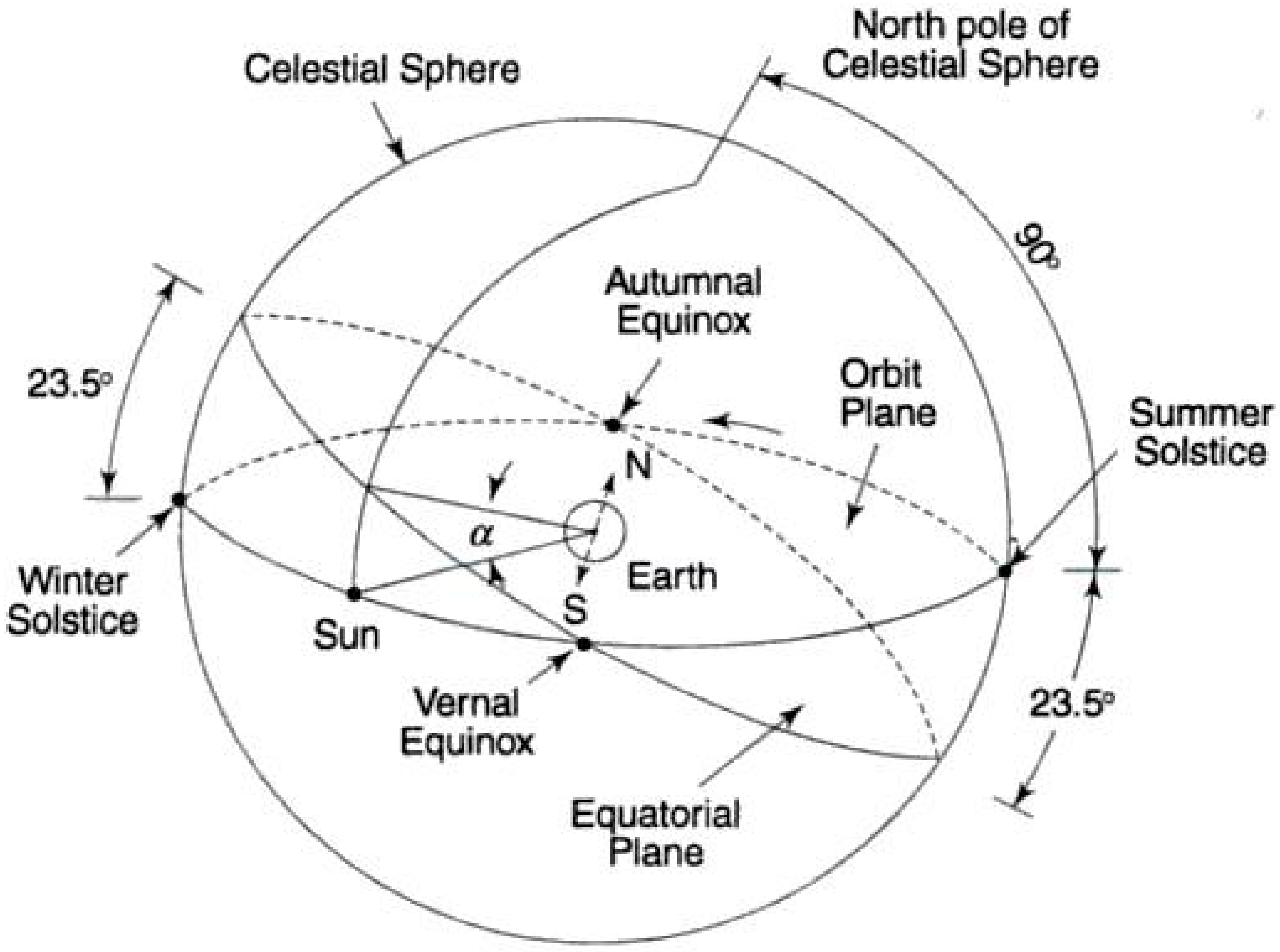

Fig. 17.5 Relative Motion of the Sun with Respect to the Earth

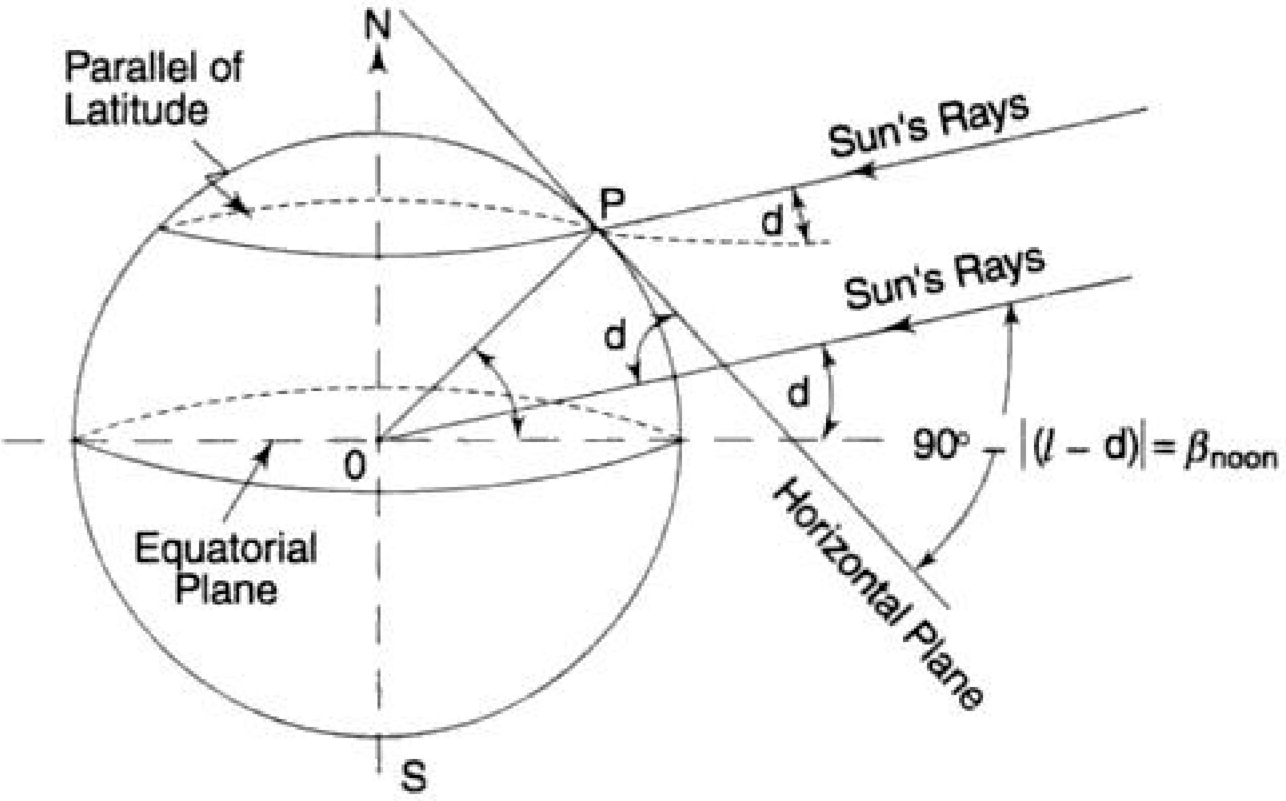

Fig. 17.6 Another Representation of Sun's Declination as Angle between the Sun's Rays and Equatorial Plane

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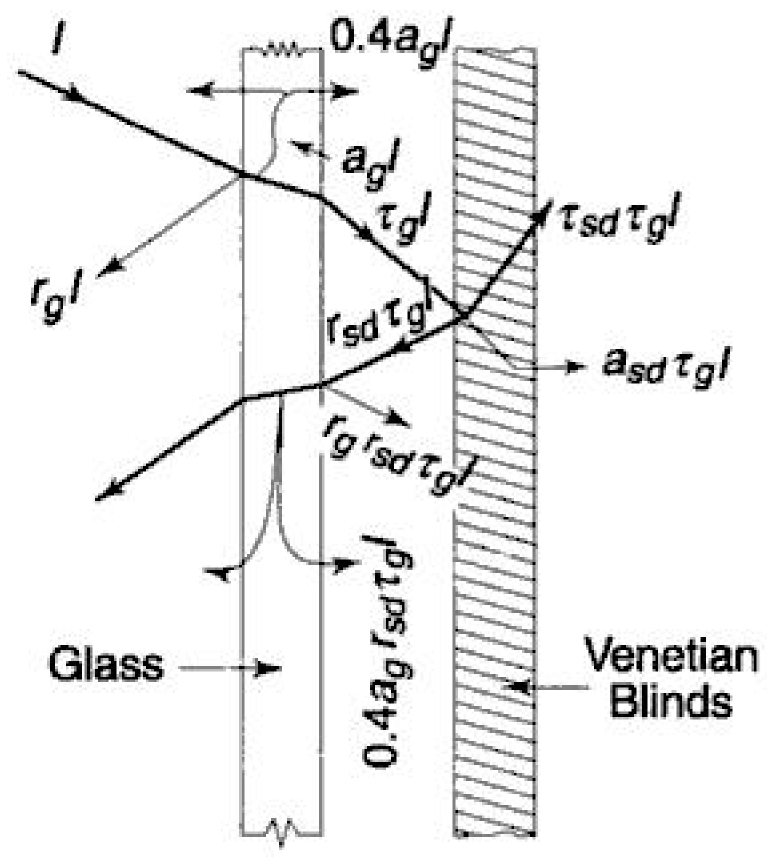

Fig. 17.18 Heat Gain of Space through Glass with Venetian Blinds Shading

Consider a radiation I incident on an ordinary glass surface at an angle of incidence of θ, say equal to 30° for which transmissivity is 0.86. A part of this radiation will be reflected, a part will be transmitted and a small part, of the order of 5 to 6 per cent, will be absorbed.

The heat absorbed by glass increases its temperature above that of the outside and inside air. A part of this heat, therefore, travels to the outside and a part to the inside. It was seen in Examples 17.4 and 17.5 that about 40 per cent of the absorbed radiation enters the space. The total heat gain of the space from the direct solar radiation is then equal to the sum of the transmitted radiation and about 40 per cent of the absorbed radiation. Thus, the radiation heat gain through a 6.35 mm thick, regular plate glass is

$$\begin{aligned} Q_g &= \tau_g I + 0.4 a_g I \\ &= [0.86 + 0.4(0.06)]I = 0.884I \end{aligned}$$

where $\tau_g = 0.86$ and $a_g = 0.06$ are the transmissivity and absorptivity of the glass.

Now, if an inside-shading device is used for which the transmissivity, reflectivity and absorptivity are τ_{sd}, r_{sd} and a_{sd} respectively, then from Fig. 17.18, the heat gain of the space to first approximation is

$$Q_{sd} = [0.4a_g + \tau_g(a_{sd} + \tau_{sd} + r_{sd}r_g + 0.4a_g r_{sd})]I \tag{17.29}$$

The values of τ, r and a for an angle of incidence of 30° for various types of glass and shading devices are given in Table 17.8. The values for venetian blinds are

$$\tau_{sd} = 0.12,\ r_{sd} = 0.51,\ a_{sd} = 0.37$$

Accordingly, the radiation heat gain of space through the glass with venetian blinds is

$$\begin{aligned} Q_{sd} = [&0.4(0.15) + 0.86(0.37 + 0.12 + 0.51(1 - 0.86 - 0.15) \\ &+ 0.4\,(0.15)\,(0.15))]I = 0.492I \end{aligned}$$

Table 17.9 (a) Solar Heat Gain through Ordinary Glass (W/m²)

0° North Latitude		a.m.												p.m.	0° South Latitude	
Time of yr.	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6	Exposure	Time of yr.
	North	0	142	205	233	246	252	259	252	246	233	205	142	0	South	
	North east	0	375	492	486	426	300	167	63	44	41	35	19	0	South east	
	East	0	365	464	426	293	136	44	44	44	41	35	19	0	East	
	South east	0	117	132	85	47	44	44	44	44	41	35	19	0	North east	
June 21	South	0	19	35	41	44	44	44	44	44	41	35	19	0	North	December 22
	South west	0	19	35	41	44	44	44	44	47	85	132	117	0	North west	
	West	0	19	35	41	44	44	44	136	293	426	464	366	0	West	
	North west	0	19	35	41	44	63	167	300	420	486	492	375	0	South west	
	Horizontal	0	88	274	464	603	685	713	685	603	464	274	89	0	Horizontal	
	North	0	117	170	192	205	208	211	208	205	192	170	117	0	South	
	North east	0	372	483	473	391	271	136	50	44	41	35	35	0	South-East	
	East	0	382	479	438	303	136	44	44	44	41	35	35	0	East	
July 23	South east	0	145	164	114	57	44	44	44	44	41	35	19	0	North east	Jan. 21
&	South	0	19	35	41	44	44	44	44	44	41	35	19	0	North	&
May 21	South west	0	19	35	41	44	44	44	44	57	114	164	145	0	North west	Nov. 21
	West	0	19	35	41	44	44	44	136	303	438	479	382	0	West	
	North west	0	19	35	41	44	50	136	271	391	473	482	372	0	South west	
	Horizontal	0	91	287	476	615	703	735	703	615	476	287	91	0	Horizontal	
	North	0	54	88	98	104	107	107	107	104	98	88	54	0	South	
	North east	0	347	445	420	322	192	76	44	44	41	38	19	0	South east	
	East	0	407	514	467	325	145	44	44	44	41	38	19	0	East	
Aug. 24	South east	0	211	249	205	110	47	44	44	44	41	38	19	0	North east	Feb. 20
&	South	0	19	35	41	44	44	44	44	44	41	38	19	0	North	&
April 20	South west	0	19	35	41	44	44	44	47	110	205	249	211	0	North west	Oct. 23
	West	0	19	35	41	44	44	44	145	325	467	514	407	0	West	
	North west	0	19	35	41	44	44	16	192	322	420	444	347	0	South west	
	Horizontal	0	98	965	473	650	738	773	738	650	473	306	98	0	Horizontal	
	North	0	19	38	41	44	44	44	44	44	41	38	19	0	South	
	North east	0	300	372	319	215	97	44	44	44	41	38	19	0	South east	

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Considering an elemental plate surface area dA over which the temperature of the fluid rises by dt, we have:

$$\dot{m}\, C\,dt = U_0 \frac{A_0}{A}\, dA\,(t_s - t) - U_B\, dA\,(t - t_0) \tag{17.31}$$

where $\dot{m}$ is the mass flow rate of the carrier fluid, and C is its specific heat

Now, eliminating t_s between Eqs (17.30) and (17.31), and integrating, we get

$$\int_{t_1}^{t_2} \frac{-K_2 \mathrm{d}t}{K_1 - K_2 (t - t_0)} = -K_3 \tag{17.32}$$

where
$$K_1 = \frac{\left(\frac{A_{\text{sun}}}{A} I_D \tau_D + I_d \tau_d\right)\alpha_s}{1 + \frac{U_F}{U_0}\frac{A}{A_0}} = \frac{\dot{Q}\, A}{1 + \frac{U_F}{U_0}\frac{A}{A_0}}$$

$$K_2 = \frac{U_F}{1 + \frac{U_F}{U_0}\frac{A}{A_0}}$$

$$K_3 = \frac{A}{\dot{m}C} K_2$$

Eq. (17.32) gives, for the rise in temperature of the carrier fluid

$$\Delta t = t_2 - t_1 = \left[\frac{K_1}{K_2} - (t_1 - t_0)\right](1 - e^{-K_3}) \tag{17.33}$$

The rate of heat collection is given by

$$\dot{Q} = \dot{m}\; C\,\Delta t$$

and the collector efficiency by

$$\eta_c = \frac{\text{Heat collected per unit area}}{\text{Solar intensity}}$$

$$= \frac{\dot{Q}/A}{I_D + I_d}$$

It can be seen that the heat collection rate depends on a number of factors including the orientation of the collector. In general, the collector should face south in the northern hemisphere, and its surface should be normal to the sun's rays as far as possible. The *optimum angle* of tilt from the horizontal position for winter operation is 10 to 20 degrees greater than the latitude angle of the place. The optimum angle of tilt for summer operation is taken as equal to the latitude angle.

Eq. (17.33) shows that for maximum t_2, K_3 should be ∞, so that $\dot{m} = 0$. Under these conditions, $\dot{Q} = 0$ and $t_2 = t_1 = t_s$. Then, if A_0 and A are equal, we have

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Table 18.1 Thermophysical Properties of Selected Building and Insulating Materials

Material	*Description*	*Specific Heat*	*Density*	*Thermal Conductivity*	*Conductance*
		kJ/kg.K	kg/m^3	W/m.K	W/m^2K
Asphalt				0.74-0.76	
Bricks	Common	0.84	1600	0.77	
	Face brick	0.84	2000	1.32	
	Diatomaceous (fired)			0.24	
	Firebrick				
	(500 to 1100)	0.96	2000	1.04 to 1.09	
	Magnesite				
	(200 to 1200)	1.13			
Woods	Ply	—	544	0.1	
	Hard	2.39	720	0.158	
	Soft	2.72	512	0.1	
Masonary	Concrete	0.88	1920	1.73	
Materials	Plaster, cement	0.796	1885	8.65	
	Hollow clay				
	tiles, 10 cm	—	—	—	5.23
	20 cm	—	—	—	3.14
	30 cm	—	—	—	2.33
	Hollow concrete				
	blocks,	10 cm	—	—	8.14
		20 cm	—	—	5.23
		30 cm	—	—	4.54
	Foam concrete		210–704	0.043–0.128	
	(Precast slabs for roof)				
Glass	Window	0.84	2700	0.78	
	Corosilicate		2200	1.09	
Insulating	Mineral or				
Materials	glass wool	0.67	24–64	0.038	
	Rockwool	—	64	0.067	
	Fibreglass board	0.7	64-144	0.038	
	Cork board	1.884	104–128	0.038	
	Cork granulated	1.88	45–120	0.045	
	Expanded polysterene				
	(Thermocole)	—	30	0.037	
	PUF			0.0137	
	Diatomaceous				
	earth	—	320	0.061	
	Felt	—	330	0.052	
	Insulex, dry	—	—	0.064	
	Kapok	—	—	0.035	
	Magnesia	—	270	0.067	
	Asbestos	0.816	470–570	0.154	

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Table 18.14 Infiltration through Swinging Doors on Opposite Walls (Wind Velocity 12 kmph)[3]

% Time 2nd Door is Open	*cmm/Pair of Doors % Time 1st Door is Open*				
	10	25	50	75	100
10	2.8	7	14	21	28
25	7.1	18	35	53	71
50	14	35	71	106	142
75	21	53	106	159	210
100	28	71	142	210	280

Table 18.15 Infiltration through Doors based on Usage (Wind Velocity 12 kmph[3])

Application	*cmm/Person in Room/Door*		
	182 cm	*91 cm Swinging Door*	
	Revolving Door	*No Vestibule*	*Vestibule*
Bank	0.184	0.227	0.17
Barber Shop	0.113	0.142	0.108
Soda Shop	0.156	0.198	0.15
Cigar Store	0.566	0.85	0.637
Department Store	0.182	0.227	0.17
Dress Shop	0.057	0.071	0.108
Drug Store	0.156	0.198	0.15
Hospital	—	0.099	0.637
Lunch Room	0.113	0.142	0.17
Men's Shop	0.076	0.105	0.054
Restaurant	0.057	0.071	0.15
Shoe Store	0.076	0.099	0.074

Example 18.7 A room is 5.5 m × 3.5 m × 3 m high. The total perimeter of an average fit non-weatherstripped window crack in the room is 10 m. The wind velocity is 9 m/s. Calculate the air-change rate of infiltration:

(i) Using Table 18.11.
(ii) Using the empirical relation for the frictional resistance to air flow

$$\dot{Q}_v = 0.125\,(0.64\,\Delta p)^{0.63}\ \text{L/m}$$

where Δp is in N/m^2

Solution

Volume of room

$$= (5.5)\,(3.5)\,(3) = 57.75\ \text{m}^3$$

Wind pressure

$$\Delta p = \frac{\rho C^2}{2} = \frac{(1.2)(9)^2}{2} = 48.6\ \text{N/m}^2$$

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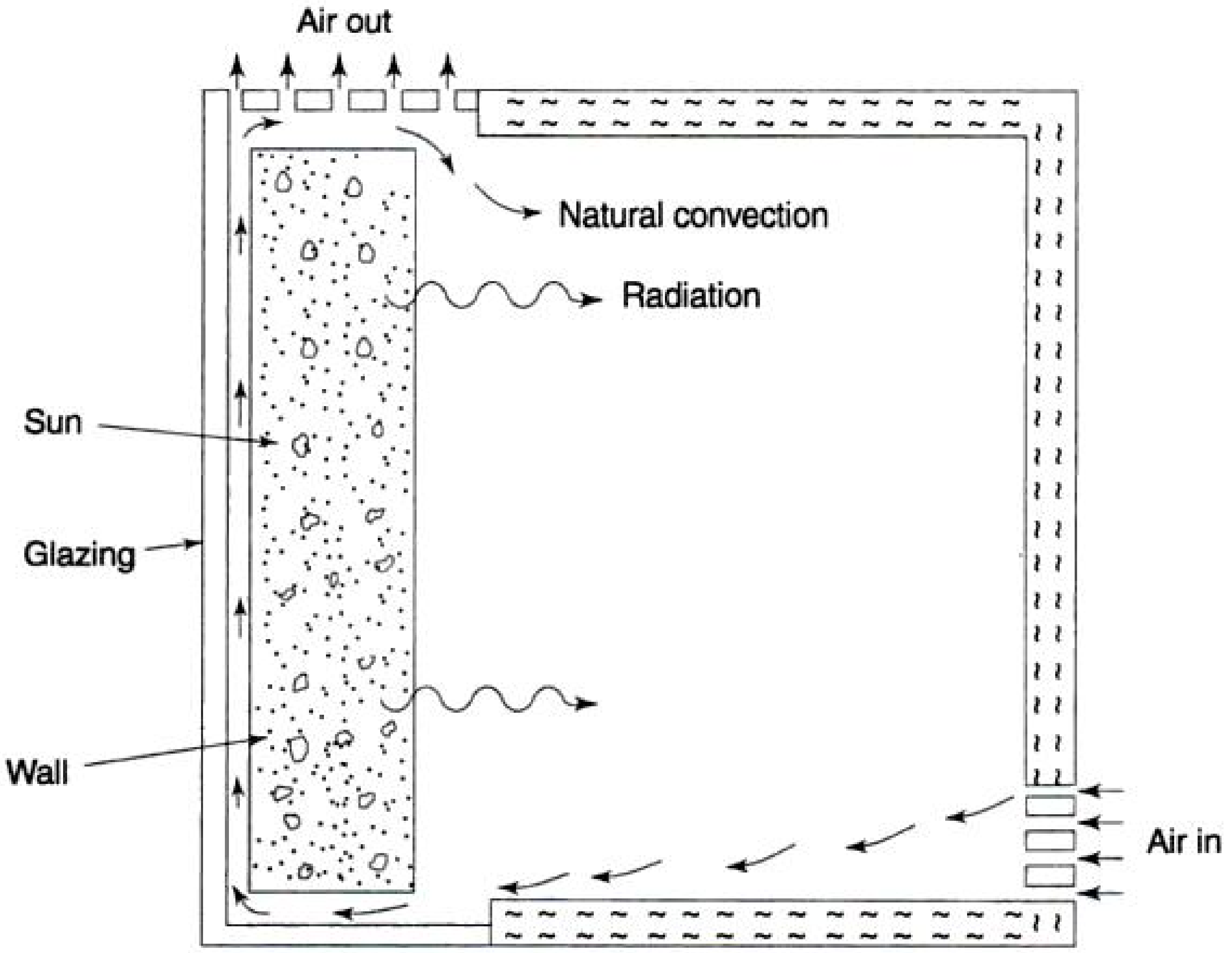

Fig. 18.19 Mass Trombe Wall with Induced Ventilation

The following controls are possible for the optimum operation of a mass Trombe wall building.

(i) In winter, when the sky is overcast and during the night, external insulation must be provided to prevent undesirable heat loss.

(ii) Similarly in summer, during the day, overhangs or external insulation will serve the purpose of preventing overheating of the living space.

(iii) The mass Trombe wall has the potential of providing induced ventilation during summer if exhaust vents are provided at the top. For this purpose, inlet openings for fresh air must be provided from a shaded or cooler area, as shown in Fig. 18.19. Thus sun's heat can be used to induce air movement to augment natural ventilation.

The roof pond is another indirect gain-type construction in which water is the storage mass, and it is located at the roof instead of the floor or wall. The body of the water is directly exposed to the sun. The thermal storage is also provided by the ceiling mass. External insulation is necessary for control, as in the case of the mass Trombe wall, to cut off the day heat gain in summer and the night loss in winter.

A clear night sky acts as a large heat sink. Thus, the heat collected by water during the day and stored in its mass is dissipated to the sky during the night. During the day, therefore, the water mass acts as a *cold storage*, drawing the heat away from the living space. It is obvious that such a system will be effective in a climate with a large swing in day and night temperatures.

Direct evaporative cooling has already been discussed. There are other methods which use *Indirect* evaporative cooling. For example, a shaded roof pond

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Partition wall construction : 33 cm brick
Roof construction : 20 cm RCC slab with 4 cm asbestos cement board
Floor construction : 20 cm concrete
Densities, brick : 2000 kg/m^3
concrete : 1900 kg/m^3
plaster : 1885 kg/m^3
asbestos board : 520 kg/m^3
Fenestration (Weather-stripped, Loose fit) : 2 m × $1\frac{1}{2}$ m glass $U = 5.9$ Wm^{-2} K^{-1}
Doors : $1\frac{1}{2}$ m × 2 m wood panels $U = 0.63$ Wm^{-2} K^{-1}
Outdoor-design conditions : 43°C DBT, 27°C WBT
Indoor-design conditions : 25°C DBT, 50% RH
Daily range : 31°C to 43°C = 12°C
Occupancy : 100
Lights : 15,000 W fluorescent

Assumed bypass factor of cooling coil 0.15

Find the room sensible and latent heat loads, and also the grand total heat load.

Solution

Thermal conductivities from Table 18.1

$$k_{\text{glass}} = 0.78 \text{ W m}^{-1} \text{ K}^{-1}$$
$$k_{\text{concrete}} = 1.73 \text{ W m}^{-1} \text{ K}^{-1}$$
$$k_{\text{brick}} = 1.32 \text{ W m}^{-1} \text{ K}^{-1}$$
$$k_{\text{plaster}} = 8.65 \text{ W m}^{-1} \text{ K}^{-1}$$
$$k_{\text{asbestos}} = 0.154 \text{ W m}^{-1} \text{ K}^{-1}$$

Assumed film coefficients

$$f_o = 23 \text{ W m}^{-2} \text{ K}^{-1}$$
$$f_i = 7 \text{ W m}^{-2} \text{ K}^{-1}$$

Outside wall

$$\frac{1}{U} = \frac{1}{23} + \frac{0.1}{1.32} + \frac{0.2}{1.73} + \frac{1}{7} + \frac{0.0125}{8.65}$$

$$U = 3.5 \text{ W m}^{-2} \text{ K}^{-1}$$

Partition wall

$$\frac{1}{U} = \frac{1}{7} + \frac{0.33}{1.32} + \frac{1}{7} + \frac{2(0.0125)}{8.65}$$

$$U = 1.86 \text{ W m}^{-2} \text{ K}^{-1}$$

WEST GLASS	12 m^2	492	—	5,900
NORTH GLASS	3 m^2	91	—	270
SOUTH GLASS	6 m^2	32	—	190
SKY LIGHT	– m^2	—	—	—
SOLAR TRANSMISSION GAIN-WALLS AND ROOF				
EAST WALL	– m^2	—	—	—
WEST WALL	96 m^2	16.5	3.5	5,540
NORTH WALL	34 m^2	11.3	3.5	1,345
SOUTH WALL	59 m^2	17.4	3.5	3,590
ROOF-SUN	459 m^2	29.7	2.13	29,035
ROOF-SHADED	– m^2	—	—	—
TRANSMISSION GAIN-OTHERS				
DOORS	9 m^2	18	0.63	100
ALL GLASS	(12+3+6) m^2	18	5.9	2,230
PARTITION	(108+28) m^2	15.5	1.86	3,930
CEILING	– m^2	—	—	—
FLOOR	459 m^2	2.5	6.05	6,940
INFILTRATION	19.8 cmm	18	20.4	7,270
INTERNAL HEAT GAIN				
PEOPLE	100	—	75	7,500
POWER	—	—	—	—
LIGHTS	15,000	—	1.25	18,750
APPLIANCES	—	—	—	—
ADDITIONAL	—	—	—	—
			SUB TOTAL	92,690
STORAGE (Neglected)	—	—	—	—
SAFETY FACTOR			5%	4,635
		ROOM SENSIBLE HEAT		103,090
SUPPLY DUCT				
SUPPLY DUCT HEAT GAIN	0.5% + LEAKAGE	0.5% + Fan HP	5%	5,560
OUTDOOR AIR BYPASSED	28 cmm	18°C	20.4 × 0.15	1,540
		EFFECTIVE ROOM SENSIBLE HEAT		104,425
LATENT HEAT				
INFILTRATION	19.8 cmm	0.006	50,000	5,940
PEOPLE	100	—	55	5,500
STEAM	—	—	—	—
APPLIANCES	—	—	—	—
ADDITIONAL	—	—	—	—
VAPOUR TRANS	—	—	—	—
			SUB TOTAL	11,440

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Evaporative cooling is a process in which heat is neither added to nor it is removed from the water outside the air washer. Water is simply recirculated by a pump.

Evaporative cooling is commonly used when the outdoor conditions are very dry. This means that the wet-bulb depression of air is very large. In a dry climate, evaporative cooling can give some relief by removing the sensible heat from the room. But the humidity cannot be controlled.

Another defect of the evaporative cooling system is the large quantity of air that must be supplied to meet the room sensible heat load as the temperature difference between the room and supply air is generally small. Thus, whereas in air conditioning, the supply air quantity may be of the order of 8–10 air changes per hour, in the case of evaporative cooling, the same may be of the order of 20 air changes. This quantity increases rapidly as the humidifying efficiency of the air washer decreases as illustrated in Example 19.5.

Example 19.5 In an industrial evaporative cooling application with outside conditions as in Example 19.2, the inside is to be maintained at a maximum relative humidity of 55 per cent. The room sensible heat is 581.5 kW. All outdoor air must be used. Find the room dry-bulb temperature and supply air quantity as a function of humidifying efficiencies of 80, 85, 90, 95 and 100 per cent.

Solution (i) Room dry-bulb temperature:

To maintain 55 per cent RH in the room, it can be seen from Fig. 19.12 that the room DBT varies along 55 per cent RH line, depending on the humidifying efficiency η_H.

Also, as the room DBT is lowered with decreasing saturation efficiency, the supply air quantity increases.

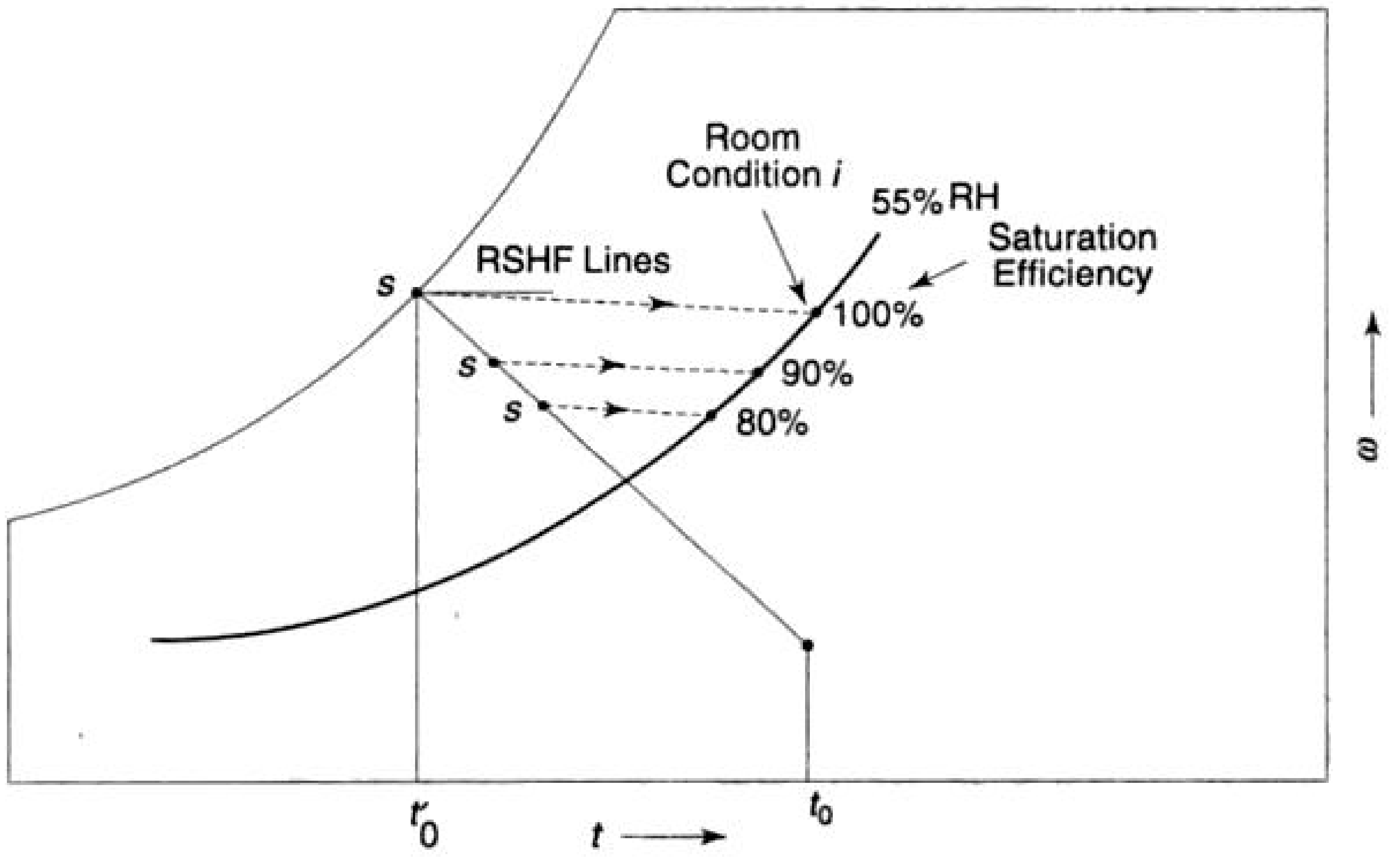

Fig. 19.12 Evaporative Cooling—Processes for Example 19.5

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illustrated in Figs 23.16 and 23.17 respectively. In this, the entering recirculated room air and outdoor air mixture is purposely diverted around the cooling coil. As against the inherent *coil bypass*, this is referred to as *system bypass*.

This method lacks humidity control as described in Sec. 23.6.2 In addition, therefore, one can provide controls allowing for the space temperature swing to 26°C in summer and 16°C in winter at the time of *peak cooling and heating loads*, thus reducing the size and energy consumption of the A/C system for the building. It has now been found that small drifts in temperature and humidity from the recommended steady-state conditions do not result in decline in comfort level. An acceptable level of change in temperature is ≤ 0.6°C per hour from a 25°C base, and in humidity is ≤ 16 torr with normal summer clothing and sedentary activity. The controls can be devised accordingly.

Note ✍: *Other energy conservation measures include the following:*

(i) Winter heating by heat pump. It will provide 3.4 kW of heating for 1 kW of power consumption.
(ii) Providing air locks in all major entrances to minimise infiltration.
(iii) Shading of the building by trees blooming in summer. The trees provide shade, and also act as evaporative coolers.
(iv) Replacing all resistance type regulators, dimmers, etc., with electronic ones.
(v) Installing timers, controls, etc., to automatically switch off lights, and air handling units (AHUs) when the space is not in use.
(vi) Preferring central A/C plants which employ water-cooled condensers in place of a number of window-type A/Cs which employ air-cooled condensers.

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3. Carrier W H, R E, Cherne, W A Grant and W H Roberts, *Modern Air Conditioning, Heating and Ventilating*, Pitman, New York, 1959.

PROBLEMS

19.1 Calculate the total heat gain of a restaurant at its peak occupancy load at about 1 p.m. when 100 diners and 15 employees are present. Given:

Heat gain through walls and roof : 2500 kJ/h
Heat gain through glass areas : 500 kJ/h
Number of fluorescent tube lights : 60
Rating of each tube light : 40 W
Rating of toasters inside space : 2650 W

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$$\cong \frac{f_g}{C_p} \quad \text{for Le} = 1$$

Combining Eqs (20.1) and (20.2) we have for total heat transfer

$$\mathrm{d}\dot{Q}_T = f_g\,(t - t_S)\,\mathrm{d}A + \frac{f_g}{\mathrm{Le}\,C_p}\,h_{fg_S}\,(\omega - \omega_S)\,\mathrm{d}A$$

$$= \frac{f_g}{C_p}\left[C_p\,(t - t_S) + \frac{h_{fg_S}}{\mathrm{Le}}\,(\omega - \omega_S)\right]\mathrm{d}A \qquad (20.5)$$

where $h_{fg_S} = h_v - h_{f_S}$.

Rearranging

$$\mathrm{d}\dot{Q}_T = \frac{f_g}{C_p}\left[\left(C_p t + \frac{h_{fg_S}}{\mathrm{Le}}\,\omega\right) - \left(C_p t_S + \frac{h_{fg_S}}{\mathrm{Le}}\,\omega_S\right)\right]\mathrm{d}A \qquad (20.6)$$

Taking average values of the humid specific heat C_p and latent heat as h_{fg_o}, we may write

$$\mathrm{d}\dot{Q}_T = \frac{f_g}{C_p}\,(h - h_S)\,\mathrm{d}A + \frac{f_g}{C_p}\left(\frac{1}{\mathrm{Le}} - 1\right) h_{fg_S}\,(\omega - \omega_S)\,\mathrm{d}A$$

$$= \frac{f_g}{C_p}\,(h - h_S)\,\mathrm{d}A, \quad \text{for} \quad \mathrm{Le} = 1 \qquad (20.7)$$

The quantity $(h - h_S)$ is the *enthalpy potential*. Thus, the driving force for the total heat transfer is the difference in the enthalpy h of unsaturated air and the enthalpy h_S of saturated air at the temperature of the wetted surface. And the coefficient determining this heat transfer is equal to $f_g/C_p = k_\omega$.

20.2.2 Surface Temperature and Direction of Process^6

The following examples illustrate the usefulness of the concept of enthalpy potential. Depending on the wetted surface temperature, and the state of the unsaturated air in contact with the wetted surface, the direction of the process can be explained. Let the air being conditioned be at state 1 as shown in Fig. 20.2, with the dry bulb, wet bulb and dew point temperatures of t, t' and t_d respectively. Then the cases arise as follows:

Case I Cooling and Dehumidification (Process 1 – S_1)

The surface temperature is lower than dew point temperature of air.

$t_{S_1} < t$	$\mathrm{d}Q_S$ is from air to surface
$\omega_{S_1} < \omega$	$\mathrm{d}Q_L$ is from air to surface
$h_{S_1} < h$	$\mathrm{d}Q_T$ is from air to surface

Case II Simple Cooling (Process 1 – S_2)

The surface temperature is equal to the dew point temperature.

$t_{S_2} = t_d < t$	$\mathrm{d}Q_S$ is from air to surface

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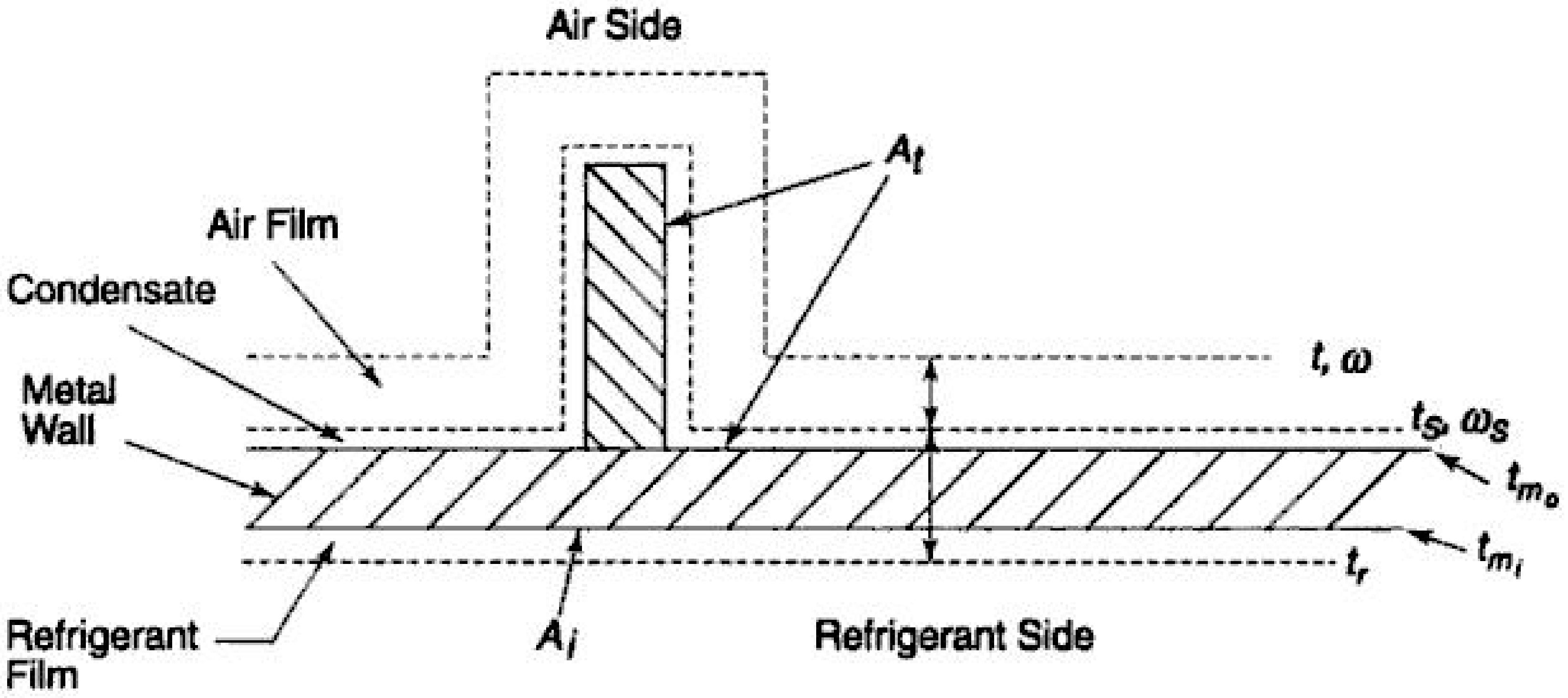

Fig. 20.4 Section of Finned Cooling and Dehumidifying Coil: Thermal Resistances

Let us consider that the coil is a finned-type. The total heat transfer through section I with the inside tube surface area $\mathrm{d}A_i$ is

$$\mathrm{d}\dot{Q}_T = \mathrm{d}\dot{Q}_S + \mathrm{d}\dot{Q}_L = \mathrm{d}\dot{Q}_T$$

$$= \frac{f_g}{C_p}(h - h_s)\,\eta_f R\,\mathrm{d}A_i \tag{20.15}$$

where η_f is the fin efficiency and

$$R = \frac{A_t}{A_i} = \text{Ratio of the total fin-side surface area } A_t \text{ to the inside surface area } A_i$$

Section II: This section comprises the three thermal resistances due to the condensate film, metal wall and refrigerant film. The heat is transferred as sensible heat only by virtue of the temperature difference $(t_S - t_r)$. Then, if U_i represents the overall heat-transfer coefficient for the three resistances based on the inside tube surface area, the heat transfer through the section is given by

$$\mathrm{d}\dot{Q}_{II} = U_i(t_S - t_r)\,\mathrm{d}A_i \tag{20.16}$$

By energy balance

$$\mathrm{d}\dot{Q}_I = \mathrm{d}\dot{Q}_{II}$$

so that we have from Eqs (20.15) and (20.16)

$$U_i\,(t_S - t_r)\,\mathrm{d}A_i = \frac{f_g}{C_p}(h - h_S)\,\eta_f R\,\mathrm{d}A_i$$

or

$$\frac{t_S - t_r}{h - h_s} = \frac{\eta_f f_g R}{U_i C_p} = \frac{k_\omega}{U_i}\,\eta_f R \tag{20.17}$$

The right-hand side is constant in case f_g and U_i are assumed to be constant. Equation (20.17) can be used for design by dividing the coil into a number of segments since t_S is varying.

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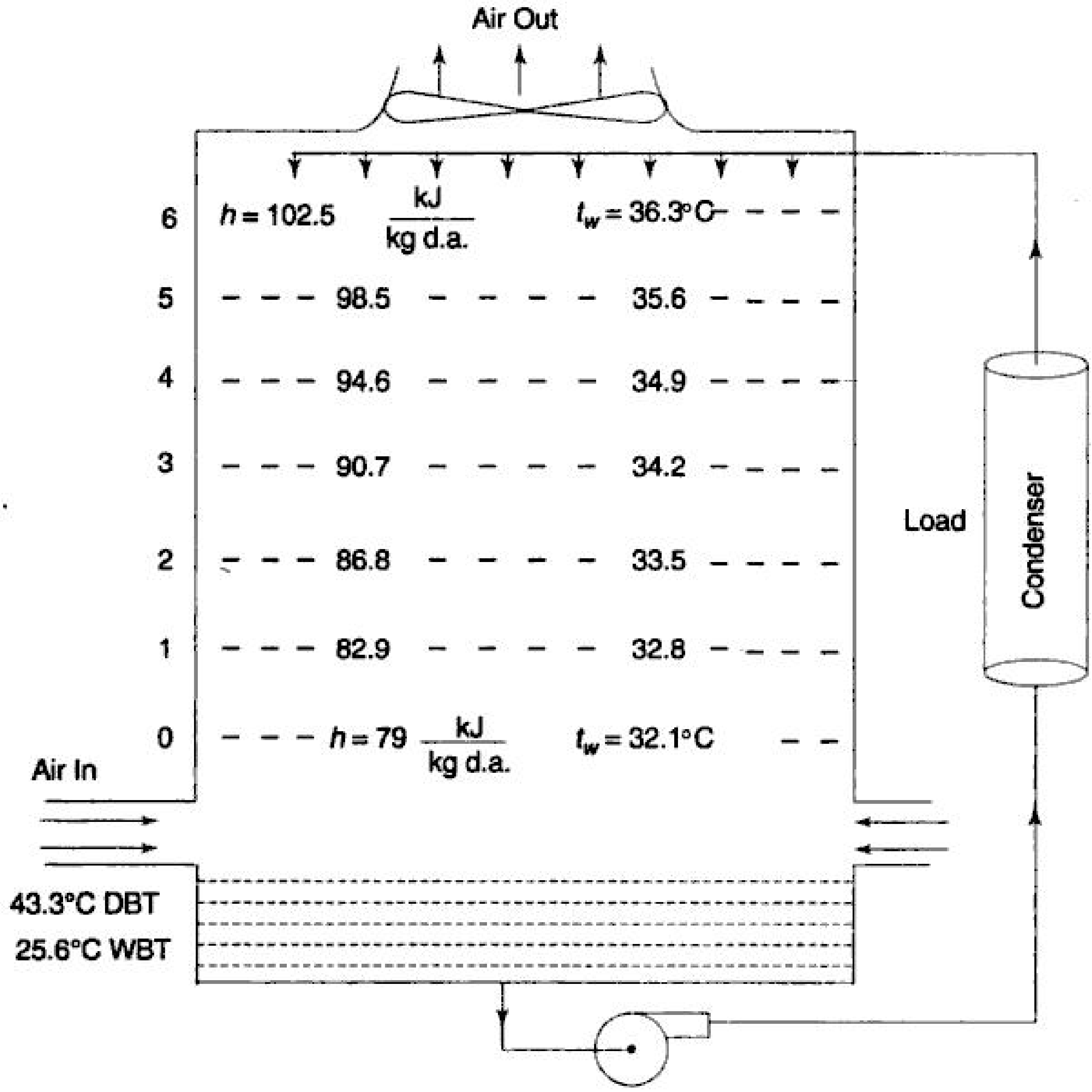

Fig. 20.13 Induced Draft Cooling Tower: Figure Showing Sections for Example 20.5

Note ✍: *(i) This value of performance coefficient is much below the value given by line B in Fig. 20.12.*

(ii) More the number of sections taken, better is the accuracy.

(iii) Calculations can also be done by using Eq. (20.43) for summation, instead of Simpson's rule, taking mean values for sections 0-1, 1-2, etc.

20.5.2 Crossflow Cooling Tower

Cooling towers without fans are known as *atmospheric cooling towers* in which the air velocity depends on *wind velocity*. The flow of air and water in them is *crossflow*. Some forced or induced draft cooling lowers are also designed as crossflow towers. This reduces the height of the tower. Such towers are convenient for installation on the roofs of high-rise buildings.

Calculations in crossflow towers can be done by dividing the tower in N number of horizontal and vertical sections. The volume of each section is thus reduced to V/N. Similarly, the area $A = aV$ as well as the performance coefficient

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in the larger branch is 9 m/s, find the mean velocities in the main duct and smaller branch.

Solution Refer to Fig. 21.14.

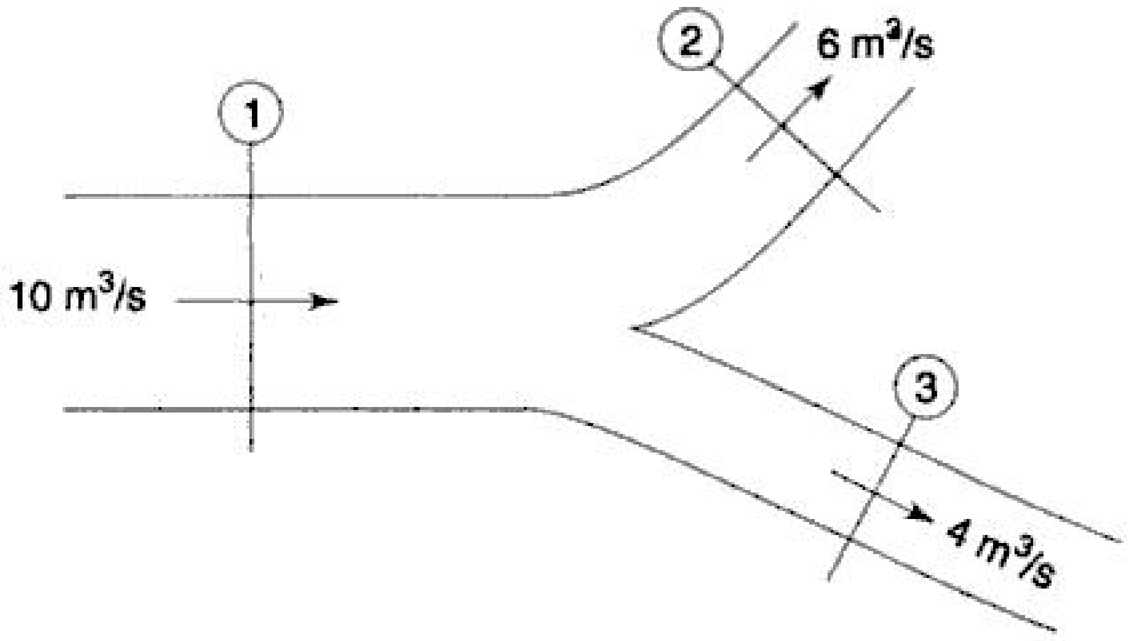

Fig. 21.14 Figure for Example 21.2

Duct areas

$$A_1 = (1.0)\,(0.9) = 0.9\ \text{m}^2$$
$$A_2 = (0.8)\,(0.8) = 0.64\ \text{m}^2$$
$$A_3 = (0.8)\,(0.6) = 0.48\ \text{m}^2$$

Flow rates

$$\dot{Q}_{v_1} = 10\ \text{m}^3/\text{s}$$
$$\dot{Q}_{v_2} = A_2 C_2$$
$$= (0.64)\,(9) = 5.76\ \text{m}^3/\text{s}$$
$$\dot{Q}_{v_3} = \dot{Q}_{v_1} - \dot{Q}_{v_2}$$
$$= 10 - 5.76 = 4.24\ \text{m}^3/\text{s}$$

Air velocities

$$C_1 = \frac{\dot{Q}_{v_1}}{A_1} = \frac{10}{0.9} = 11.1\ \text{m/s}$$

$$C_3 = \frac{\dot{Q}_{v_2}}{A_2} = \frac{4.24}{0.48} = 8.83\ \text{m/s}$$

21.2.2 Pressure Drop in Ducts

As flow continues in a duct, the static pressure of air drops. This drop in pressure takes place due to two factors:

(i) Duct friction, and
(ii) Change of direction and/or velocity.

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Table 21.4 Pressure Losses due to Elbows, Bends and Tees[1]

Reference	*Type*	*Conditions*			*Pressure loss*		
in Fig. 21.17					*K*	L_e/D	L_e/b
(a)	θ-deg	Rectangular or round; with or without vanes			$\frac{\theta}{90}$ times value for similar 90-deg elbow		
(b)		Miter			1.3	65	
	90-deg round section	$R/D = 0.5$			0.9		
		1.0			0.33	17	
		1.5			0.24	12	
		2.0			0.19	10	
(c)	90-deg rectangular section	a/b	R/b				
		0.25	Miter		1.25		25
			0.5		1.25		25
			0.75		0.6		12
			1.0		0.37		7
			1.5		0.19		4
			Miter			1.47	49
		0.5	0.5		1.1		40
			0.75		0.5		16
			1.0		0.28		9
			1.5		0.13		4
		1.0	Miter		1.5		75
			0.5		1.0		50
			0.75		0.41		21
			1.0		0.22		11
			1.5		0.09		45
		4.0	Miter		1.38		110
			0.5		0.96		65
			0.75		0.37		43
			1.0		0.19		17
			1.51		0.07		6
(d)	Miter with turning vanes	$K = 0.1$ to 0.35 depending on manufacture					
(e)	Miter tee with vanes	Consider equal to a similar elbow, base loss on upstream velocity					
(f)	Radius tee	As above					
(g)	90-deg section with splitter vanes	R/b	R_1/b	R_2/b	K		L_e/b
		Miter	0.5				28
		0.5	0.4		0.7		19
		0.7	0.6				
		1.0	1.0		0.13		7.2
		1.5			0.12		
		Miter	0.3	0.5			22
		0.5	0.2	0.4	0.45		16
		0.75	0.4	0.7	0.12		
		1.0	0.7	1.0	0.1		
		1.5	1.3	1.6	0.15		

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(i) If it is a 20°-angle gradual enlargement, the pressure loss is given by

$$\Delta p_L = 0.45\ (p_{V_1} - p_{V_2})$$

Calculate the static regain.

(ii) What will be the static regain in an abrupt enlargement?

(b) A 10 cm diameter duct converges gradually to 7.5 cm diameter. The static pressure and velocity just upstream of the reducer are 3 cm H_2O and 7.6 m/s respectively. The dynamic loss coefficient of the reducer is 0.1.

Calculate:

(i) The flow rate.

(ii) The pressure loss in the reducer.

(iii) The pressure indicated by a U-tube water manometer connected to pressure tappings at the upstream and downstream of the reducer.

Solution (a) (i) *Gradual enlargement*:
Pressure drop $(1 - R = 0.45)$

$$\begin{aligned}\Delta p_L &= 0.45\ (p_{V_1} - p_{V_2})\\ &= 0.45\ (0.6 \times 20^2 - 0.6 \times 5^2)\\ &= 0.45\ (225) = 101.3 \text{ N/m}^2\end{aligned}$$

Static regain

$$\begin{aligned}\text{SR} &= (p_{V_1} - p_{V_2}) - \Delta p_L\\ &= 225 - 101.3 = 123.7 \text{ N/m}^2\end{aligned}$$

(ii) *Abrupt enlargement*:
Dynamic loss coefficient

$$(K_2)_{\max} = \left(\frac{A_2}{A_1} - 1\right)^2 = \left(\frac{C_1}{C_2} - 1\right)^2 = \left(\frac{20}{5} - 1\right)^2 = 9$$

Pressure drop

$$\begin{aligned}(\Delta p_L)_{\max} &= (K_2)_{\max}\, p_{V_2}\\ &= 9\ (0.6 \times 5^2) = 135 \text{ N/m}^2\end{aligned}$$

Static regain

$$\text{SR} = (p_{V_1} - p_{V_2}) - \Delta p_L = 225 - 135 = 90 \text{ N/m}^2$$

(b) *Gradual reducer*:
(i) Flow rate

$$\dot{Q}_v = A_1 C_1 = \frac{\pi\ (0.1)^2}{4}\ (7.6) = 0.0597 \text{ m}^3\text{/s}$$

(ii) Downstream velocity

$$C_2 = \frac{\dot{Q}_v}{A_2} = \frac{(0.0597)^4}{\pi (0.075)^2} = 13.5 \text{ m/s}$$

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(iv) Diverging sections should be made gradual. The angle of divergence should not exceed 20°.

(v) Rectangular ducts should be made as nearly square as possible. This will ensure minimum duct surface, and hence cost, for the same air-carrying capacity. An aspect ratio of less than 4 : 1 should be maintained.

(vi) Ducts should be made of smooth materials such as galvanized iron (GI) or aluminium sheet metals. Whenever other materials are used, allowance should be made for the roughness of the material.

(vii) Dampers should be provided in each branch outlet for balancing the system.

(viii) Avoid duct obstructions.

21.6.1 Duct Construction

The most commonly used duct material is GI sheet metal. Aluminium sheet metal may be used where there are benefits to be derived due to its lighter weight and resistance to moisture. Heavy gauge black-steel may be used for a kitchen exhaust. Cement asbestos may be used for underground air distribution and for exhausting corrosive materials. Fibre glass ducts are in use in low velocity applications where thermal insulation or accoustic treatment is required.

Duct reinforcing is done by sheet metal itself or by steel or extruded aluminium angles.

21.6.2 Pressure Classifications

Air imposes two types of loads on a duct structure:

(i) The static load due to the mean static pressure differential across the duct wall. This is the dominating load.

(ii) The pulsating load due to turbulent air flow. This load is comparatively small.

Accordingly, the ducts are classified as follows:

(i) *Low-pressure system*
Velocities < 600 mpm and static pressure ≤ 5 cm H_2O gauge.

(ii) *Medium-pressure system*
Velocities < 600 mpm or static pressure up to 15 cm H_2O gauge.

(iii) *High-pressure system*
Velocities > 600 mpm or static pressure over 15 cm and up to 25 cm H_2O gauge.

The low-pressure system permits the use of simple forming methods. A majority of duct construction belongs to the low-pressure type. The gauge of the sheet metal depends on the dimension of the longest side as given in Table 21.7.

The recommended maximum duct velocities for the low-velocity system are given in Table 21.8.

Medium and high pressure rectangular ducts may be used to meet the space limitations. It is generally a more expensive construction. Tie rods as well as

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$$= \frac{1}{3}(20.4 - 13.14) = 2.42 \text{ mm H}_2\text{O}$$

Outlet duct friction loss

$$\Delta p_{f_{3-4}} = \frac{4 f_3 L_{3-4}}{D_3} \; p_{V_3}$$

$$= \frac{4(0.0035)(60)(13.14)}{1} = 11.04 \text{ mm H}_2\text{O}$$

By adding all the pressure drops we get the total duct loss as 43.87 mm H_2O. Note that Δp_O has not been included.

Fan total pressure

$$\text{FTP} = \text{Pressure loss in duct system} + \text{outlet velocity pressure}$$

$$= 43.87 + 13.14 = 57.01 \text{ mm H}_2\text{O}$$

(ii) Equating total pressures in the atmosphere and fan inlet

$$0 = \Delta p_{0-2} + p_{V_2} + p_{S_2}$$

whence the static pressure at the fan inlet is

$$p_{S_2} = -\,\Delta p_{0-2} - p_{V_2}$$

$$= -\,(10.05 + 5.36 + 15) - 20.1$$

$$= -\,30.41 - 20.1 = -\,50.51 \text{ mm H}_2\text{O} \quad \text{(Negative)}$$

Energy balance of fan from Eq. (21.52)

$$p_{S_3} + p_{V_3} = p_{S_2} + p_{V_2} + \text{FTP}$$

whence the static pressure at the fan outlet is

$$p_{S_3} = p_{S_2} + p_{V_2} + \text{FTP} - p_{V_3}$$

$$= -\,50.51 + 20.1 + 57.01 - 20.4 = 6.2 \text{ mm H}_2\text{O (Positive)}$$

(iii) The total pressures and static pressures at the other sections are as follows:

Section 1:

$$p_{T_1} = p_{T_0} - \text{Inlet loss}$$

$$= 0 - 10.05 = -\,10.05 \text{ mm H}_2\text{O}$$

$$p_{S_1} = p_{T_1} - p_{V_1}$$

$$= -\,10.05 - 20.1 = -\,30.15 \text{ mm H}_2\text{O}$$

Section 2:

$$p'_{T_2} = p_{T_1} - \text{Inlet duct friction loss}$$

$$= -\,10.05 - 5.36 = -\,15.41 \text{ mm H}_2\text{O}$$

$$p'_{S_2} = p'_{T_2} - p'_{V_2}$$

$$= -\,15.41 - 20.1 = -\,35.51 \text{ m H}_2\text{O}$$

Section 2:

$$p_{T_2} = p'_{T_2} - \text{Loss in A/C apparatus}$$

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drop of 25 mm H_2O, and are normally operated to a final pressure drop of 75 mm H_2O *before being discarded.* The life of HEPA filters can be enhanced by using roughing filters, and microfine pre-filters, particularly in the *make-up/ventilation* air circuit, prior to them.

21.7.2 Conventional Flow Clean Rooms

Conditioned air in the clean rooms should be directed to obtain the *cleanest air at the most critical work areas.* Thereafter, as the particles are entrained, they are led to the less critical areas. These criteria require introduction of large quantities of air at low velocities.

Further, all rooms should be maintained at positive pressure, i.e.—at a static pressure sufficiently above atmospheric pressure—so as to prevent infiltration of contaminated outside air and permit only exfiltration.

Conventional flow patterns of air are described in Sec. 21.1.4. A conventional flow clean room A/C system (cooling and dehumidification) is illustrated in Fig. 21.28.

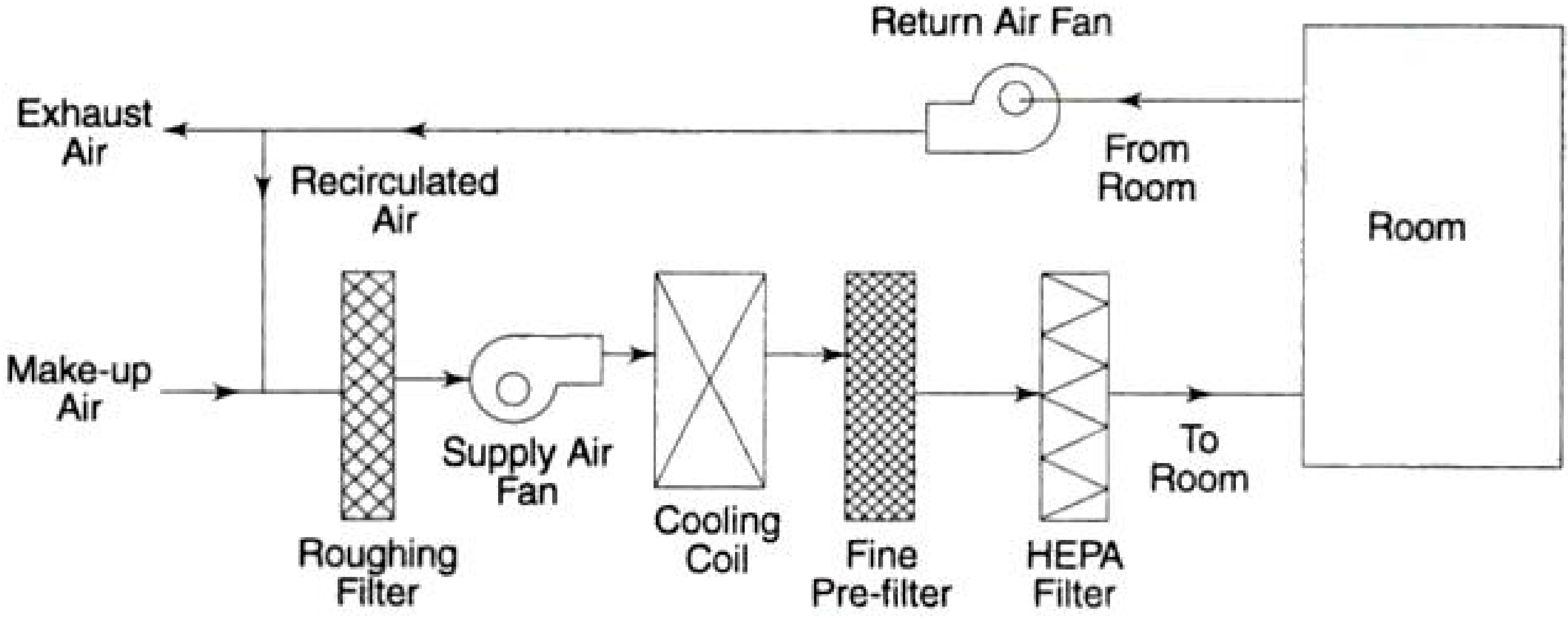

Fig. 21.28 Conventional Flow Clean Room A/C Systems

Major design features of the conventional flow clean rooms are 20 air changes per hour, positive pressure, final filters to be HEPA filters, installed only on discharge side of the fan, and to be preferably *terminally mounted*, people/particle generating activities near air exit, etc. The building materials used should not chip off or flake. All surfaces should be flush finished. Surface finish is done by apoxy paint, etc. Special lint free garments and clothing are to be used. Further, support areas are provided for *air showers* and *air locks*, restricting entry of personnel and introducing clean room discipline.

21.7.3 Laminar Flow Clean Rooms

It was found that even after taking all the measures described above, cleanliness levels could not be improved any further in conventional flow rooms. Major drawback is the build-up of smaller size particles. This method also does not prevent *cross-contamination* from one work area to another.

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Chapter 23

Refrigeration and Air-Conditioning Control

If the refrigeration load and capacity of a plant remained constant, there would not be any need to have any control. However, in practice, both do fluctuate. The cooling load changes due to input conditions and changes in the outside conditions. The plant capacity also changes due to the changes in the outside conditions. Also, since the plant is designed for peak load, its capacity is usually greater than the immediate demand, resulting in a trend to establish a suction pressure and temperature that are lower than those necessary to fulfil the peak demand.

Automatic control makes the plant independent of skilled supervision. It also achieves a higher degree of accuracy in maintaining the required temperatures, pressures and humidities. Further, it offers flexibility in the operation of complicated plants on various impulses from different parts of the system ensuring overall control and protection.

23.1 BASIC ELEMENTS OF CONTROL

In every control device, there is a *controlled variable* or controlled condition such as the temperature of the conditioned space. Figure 23.1 shows the basic elements of a control system in the form of a block diagram. The *detecting element* feeds a signal to the controlling unit, when any deviation from the desired value of the controlled condition, i.e., from the *set value*, occurs. The measuring element of the control unit then measures the extent of this deviation and triggers the *actuating element*. The *positioner* is responsible for setting the value of the controlled variable. In case, the force from the *measuring element* is not strong enough to actuate the regulating unit, servomotors or relays can be used. The *transmitting element* is at times necessary between the actuating and regulating elements. It is a part of the control unit proper. Often, the functions of the various

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C. Defective Compressor or Wrong Application of Compressor			
Mechanical defects			×
Wrong application range (evaporating temp.)	×	×	
Wrong application (ambient temp).	×		
Wrong voltage range	×	×	×
D. Defects in Refrigeration System			
Dirty condenser	×		
Capillary tube partly or completely blocked			×
System overcharged	×	×	
System or Cabinet			
E. Wrong Design of Refrigeration System or Cabinet			
Condenser too small	×		
Wrong condenser design			×
Wrong evaporator design (too large volume)	×		
Too high equalizing pressure		×	
Wrong capillary tube			×
Bad mounting conditions for compressor or condenser	×		
Wrong thermostat differential			×
F. Other Conditions			
Very low compressor temperature		×	

23.6 CONTROLLING ROOM CONDITIONS AT PARTIAL LOAD

An air-conditioning plant is selected for design peak load conditions. But most of the time the plant operates under partial load which is either due to change in the outside conditions or due to withdrawal of load. The control of room conditions under part load is, therefore, very important. For this purpose, one or more of the following methods may be employed:

(i) On-off control of the air-handling unit or refrigeration unit.
(ii) Controlled bypass of the air-entering apparatus.
(iii) Varying water flow in the chilled-water coil.
(iv) Reheating the supply air.
(v) Controlling the volume of the supply air.

23.6.1 On-off Control

In the on-off control of the air-handling unit or fan-coil unit, the fan motor is operated intermittently by the room thermostat, the bulb of which is installed in the return air. In this, the refrigerant or chilled water continues to flow through the cooling coil. This results in fluctuating room temperature. In the on-off con-

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Secondary room-air quantity

$$(\text{cmm})_2 = 4\,(\text{cmm})_1$$
$$= 4(8) = 32 \text{ cmm}$$

Supply air quantity

$$(\text{cmm})_s = 8 + 32 = 40 \text{ cmm}$$

Supply air temperature

$$t_s = t_i - \frac{\text{RSH}}{0.0204\,(\text{cmm})_s}$$
$$= 24 - \frac{5}{0.0204\,(40)} = 17.9°\text{C}$$

(a) Room latent-heat balance:

$$\text{RLH} = 50(\text{cmm})_s\,(\omega_i - \omega_s)$$
$$= 50\,(\text{cmm})_s\left(\omega_i - \frac{4\omega_i + \omega_1}{5}\right)$$
$$1 = 50(40)\left(\omega_i - \frac{4\omega_i + 0.0077}{5}\right)$$

whence $\omega_i = 0.0102$ kg/kg

The room relative humidity from the psychrometric chart at $t_i = 24°\text{C}$ and $\omega_i = 0.0102$ kg/kg is found to be 53.5%.

(b) Specific humidity of air after cooling in the induction unit:

$$\omega_2 = \omega_i = 0.0102 \text{ kg/kg}$$

Dry bulb temperature at 2

$$t_2 = \frac{(\text{cmm})_s\,t_s - (\text{cmm})_1\,t_1}{(\text{cmm})_2}$$
$$= \frac{40(17.9) - 8(12)}{32} = 19.4°\text{C}$$

Enthalpies at, o, i, 1 and 2 from the psychrometric chart

$$h_o = 68.0 \text{ kJ/kg}$$
$$h_i = 50.0 \text{ kJ/kg}$$
$$h_2 = 45.0 \text{ kJ/kg}$$
$$h_1 = 31.5 \text{ kJ/kg}$$

Cooling load on the room-induction unit

$$\dot{Q} = 0.02\,(\text{cmm})_2\,(h_i - h_2)$$
$$= 0.02(32)\,(50 - 40) = 6.4 \text{ kW}$$

Cooling load on the central plant

$$\text{OATH} = 0.02\,(\text{cmm})_1\,(h_o - h_1)$$
$$= 0.02\,(8)\,(68 - 31.5) = 5.84 \text{ kW}$$

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Chapter 24

Applications in Food Processing and Preservation

Air conditioning for comfort is one of the most common application of refrigeration, and it has been dealt with at length in the preceding chapters. However, the most significant application of refrigeration is in food preservation, whether it is by way of processing or for storage. Processing is done by heating, heat drying, etc., and by refrigeration such as in chilling, freezing or freeze-drying. Storage may be of either chilled or frozen product. Some of the important products involved in processing are candy, beverages, meat, poultry, fish, bakery and dairy products, fruits and vegetables, fruit-juice concentrates, precooked foods, etc. The common products preserved by storage after chilling are fruits such as apples, pears, grapes, citrus fruits, etc., vegetables such as onions, potatoes, tomatoes, etc., and dry fruits, candies, milk, eggs and their products. Storage under frozen conditions is resorted to for preserving the food value as well as to store perishable products over a long period. The common items of frozen food are fish, meat, poultry, and some vegetables such as peas, beans, carrots, cauliflower, etc. It may be noted that all products are not amenable to freezing.

An interesting feature of the chilled and frozen food industry is the cold chain that must be maintained from the farm to the consumer. An important link in this chain is that of transport refrigeration.

In this chapter, a few problems in food refrigeration and dehydration have been studied from the point of view of thermodynamic and heat transfer principles. It must be stated here that processing by simple heat drying is a more economical way of food preservation. However, with many products, it may involve loss of quality in which case it is not considered suitable. The study of the drying process involves psychrometric principles.

24.1 TYPICAL EXAMPLES OF FOOD PROCESSING BY REFRIGERATION[1] AND STORAGE

A few typical examples of food processing by refrigeration and the operating conditions involved may be of interest to the readers.

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multilayered plates in contact with primary or secondary refrigerant. For obtaining individual freezing of small-sized fish, such as shrimp, two methods are employed. *Air-blast freezers* use air at – 29 to – 40°C flowing at 2.5 m/s or more. *Immersion freezers* make use of liquid freon or even liquid nitrogen at atmospheric pressure. The freezing time depends on the water content and thermophysical properties of fish. It also depends on the refrigeration temperature. Frozen fish is normally stored at a temperature of –23°C or lower.

24.1.7 Fruits and Vegetables

Storage requirements of most fruits and vegetables are given in Chapter 16. It may be stated here that rapid pre-cooling after harvesting increases the storage life considerably. Further, *controlled atmosphere* (CA) storage greatly reduces the respiration rate and retards the deterioration of the product, thus increasing its storage life. It may otherwise raise the requirement of the storage temperature. A controlled atmosphere aims at maintaining the oxygen level at a low value of 3-5 per cent and the carbon dioxide level at a high concentration of 3-10 per cent.

A few vegetables, such as peas, beans, carrots and cauliflower are also found suitable for freezing. These are partially cooked before freezing. Quick freezing is not so critical in the case of vegetables. Nevertheless, airblast freezers are commonly used for individual quick freezing of these products. Indirect immersion freezing is also being resorted to on a small scale for packaged products.

Dehydro-freezing is another process that has been found successful in the preservation of some products, such as apples, apricots, peas, etc. This involves partial dehydration, and then freezing. The process retains the advantages of both, without the disadvantages of either. One clear advantage is the reduced refrigeration load during freezing. Another is the reduced weight of the product for transport. In simple dehydration, there is a marked deterioration in quality during the last stages of drying. This is eliminated in dehydro-freezing.

24.1.8 Mushroom Cultivation and Storage: A Case Study

Mushroom cultivation, processing, transportation and storage has increased manifold in the last few years. It provides an interesting case study. The stages involved are as follows:

Stage	*Area of Treatment*	*Inside Conditions*	*Method*	*Remarks*
I	Growth period (1 Month)	15.5°C DBT, 90% RH	A/C by chilled-water plant and steam injection	
II	Pre-cooling (45 min)	3°C DBT, 90% RH	D-X plant with Sprayed-coil	Rapid pre-cooling after harvesting
	(5 min)	–1°C DBT, 90% RH		Slight Cool-down from 3°C to –1°C

(Contd)

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Outside:	43°C DBT,	27°C WBT
Inside:	2°C DBT,	90% RH

The transmission load is estimated as 1.5 kW. The fan motor and lighting load is assumed as 0.3 kW. The product loading is 2000 kg of fruits/vegetables every day. Assume 6 air changes per day equivalent of infiltration due to door openings. Determine:

(i) Refrigerating capacity required.
(ii) Approximate saturated suction temperature of refrigerant.
(iii) Number of air changes of supply air.

Solution

Refer to Fig. 24.1.

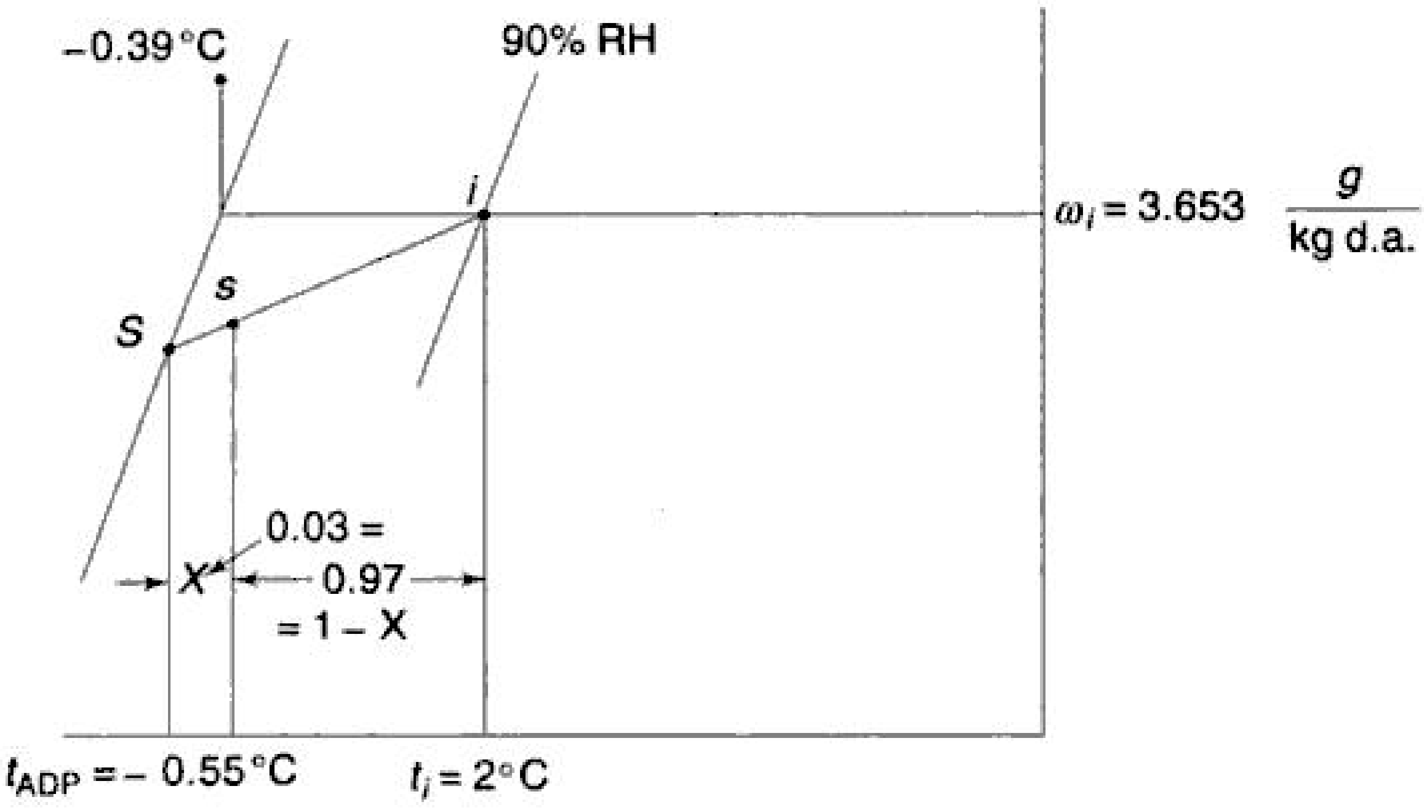

Fig. 24.1 Psychrometric Points for Example 24.1

As the inside condition is very close to saturation curve, all psychrometric properties are found by calculation rather than by using psychrometric chart. The same are as follows:

	DBT,°C	WBT,°C	RH,%	p'_v, mmHg	p_s, mmHg	p_v, mmHg	DPT, °C	ω, g/kg	h, kJ/kg
Outside	43	27	29	26.74	64.8	18.79		15.77	83.91
Inside	2		90		4.93	4.437	–0.39	3.653	11.16

Volume of space

$$V = 8.5 \times 5 \times 2.5 = 106.25 \text{ m}^3$$

(i) Product load (specific heat of fruits/vegetables = 3.77 kJ/kg.K)

$$\dot{Q} = \frac{2000\,(3.77)\,(43 - 2)}{24 \times 3600} = 3.58 \text{ kW}$$

Infiltration air

$$\dot{Q}_v = \frac{106.25 \times 6}{24 \times 60} = 0.44 \text{ cmm}$$

perature is at 0°. This involves subtracting or adding a constant to each temperature involved, in order to obtain the excess temperature to be used in the foregoing equations.

Example 24.6 A package of meat 5.1 cm thick is being frozen in a plate or contact freezer. The meat is initially at 4.4°C. It has a moisture content of 75 per cent and its freezing-point average is – 5°C. The meat is being frozen from both sides and the refrigerant is at – 29°C. Determine the time for the meat to pass through the freezing zone. The thermophysical properties of thawed (2) and frozen (1) materials are as follows:

$$k_2 = 0.571 \text{ W m}^{-1}\,°\text{C}^{-1}$$
$$k_1 = 1.04 \text{ W m}^{-1}\,°\text{C}^{-1}$$
$$\rho_2 = 1057 \text{ kg m}^{-3}$$
$$\rho_1 = 961 \text{ kg m}^{-3}$$

C_2 = specific heat of thawed meat

$$= 0.75\,(4.18) + 0.25\,(1.38) = 3.488 \text{ kJ kg}^{-1}\,°\text{C}^{-1}$$

C_1 = specific heat of frozen meat

$$= 0.75\,(2.09) + 0.25\,(1.38) = 1.918 \text{ kJ kg}^{-1}\,°\text{C}^{-1}$$

Also calculate the time for the meat centre to reach – 17.8°C.

Note: *The specific heat of dried material is 1.38. The specific heats of water and ice are taken as 4.18 and 2.08 respectively.*

Solution

In this case, the surface temperature is at – 29°C. In order to make the surface-excess temperature zero, we have a temperature scale

$$t = t' - (-29) = t' + 29$$

where t' is the actual temperature and t is the excess temperature. Thus, on the excess-temperature scale

$$t_i = 4.4 + 29 = 33.4$$
$$t_s = -5 + 29 = 24$$

Also, taking the latent heat of fusion of water as 335.5 kJ/kg, we have for the latent heat of fusion of meat (with 75% water content)

$$\Delta h_{\text{meat}} = 0.75\,(335.5) = 252 \text{ kJ/kg}$$

Thermal diffusivities

$$\alpha_2 = \frac{0.571}{(1057)\,(3.488 \times 10^3)} = 1.549 \times 10^{-7} \text{ m}^2\,\text{s}^{-1}$$

$$\alpha_1 = \frac{1.04}{(961)\,(1.918 \times 10^3)} = 5.64 \times 10^{-7} \text{ m}^2\,\text{s}^{-1}$$

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(b) Let T_R be the absolute temperature of the radiating surface. Then by energy balance between heat transfer to the surface by radiation and heat transfer by conduction between the surface and the ice front, we have

$$\frac{k}{\Delta x}(t_s - t_f) = 5.669\left[\left(\frac{T_R}{100}\right)^4 - \left(\frac{T_S}{100}\right)^4\right]$$

$$\frac{(0.015\,(65+26)}{(0.01125/2)} = 5.669\left[\left(\frac{T_R}{100}\right)^4 - \left(\frac{273+65}{100}\right)^4\right]$$

whence

$$T_R = 363 \text{ K}$$

$$t_R = 90°\text{C}$$

24.5.2 Freeze Drying of Yoghurt: Influence of Concentration of Milk Solids

Yoghurt is a common ingredient of diet because of its nutritional and therapeutic value. Accordingly, efforts should be made to incorporate it in the manufacturing of products, such as yoghurt-flavoured baby foods, by freeze drying process.

The high cost of freeze drying can be reduced, particularly in the case of yoghurt, by reducing the initial moisture content of the product. Sharma, Arora and Mital[11] conducted a study to determine the effect of *milk solids* in yoghurt on its freeze drying rate, and the quality of the reconstituted product.

For yoghurt preparation, the milk was heated to 85°C for 30 min. The total solids in milk were increased by adding non-fat dry milk (NFDM) (moisture 5%, fat 2.7%, protein 35%) at 2, 5 and 8% (w/v) levels. The resultant solids in the yoghurt were 14.7% without addition of NFDM, and 16.3%, 18.8% and 20.4% with the addition of NFDM. The milk was then cooled and inoculated with the *cultures Streptococcus thermophilous*-YHS and *Lactobacillus delbrueckii* subsp. *bulgraicus*-YHL in the ratio of 1:1 at 3% *v/v*.

The inoculated milk was incubated at 42°C for 3.5 hours. The yoghurt thus prepared was kept at 4 ± 1°C until transferred to a *product tray* for freezing at – 24°C.

The experimental set-up is shown in Fig. 24.11. The compressor maintained a temperature of – 54 ± 3°C in the freeze condenser. A temperature controller automatically adjusted the heater-platen temperature to 45 ± 3°C. The schematic of the product tray is shown in Fig. 24.12. The pressure in the vacuum chamber ranged from 1.3 m bar at the beginning to 0.52 m bar at the end of drying run. After drying, the samples were stored in glass-stoppered containers at 4°C.

There are two methods of heating:

(i) Contact-plate heating.

(ii) Radiant heating.

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15. Tressler D K and C F, Evers, *The Freezing Preservation of Foods*, Vols. I and II, AVI, Westport, Connecticut, 1957.
16. Trifonova L I, '*Theoretical and experimental investigation of freeze dehydration process*, Ph D. thesis, IIT, Delhi, 1975.
17. Trifonova L I and C.P. Arora , 'Symmetrical steady state freeze drying and optimization of radiator temperature, *XIV Int. Cong. Refrigeration, Moscow*, 1975.

PROBLEMS

24.1 (a) 10 cm thick meat slabs, initially at 35°C, are kept in a cold room at 1°C. Using numerical method, find the time required for the centre of the slab to reach a temperature of 4°C. The heat-transfer coefficient of air may be taken as 10 $\text{Wm}^{-2}\ \text{K}^{-1}$. The thermophysical properties of meat are as follows:

Thermal conductivity = 0.65 $\text{W m}^{-1}\ \text{K}^{-1}$

Specific heat = 3.5 $\text{kJ kg}^{-1}\ \text{K}^{-1}$

Density = 1020 kg m^{-3}

(b) Compare your result with that obtained by using Gurnie-Lurie charts.

24.2 (a) A 10 cm thick meat slab is being frozen from both sides in a contact-plate freezer. The meat is initially at 5°C and has a moisture content of 75 per cent. The average freezing point is – 5°C. The plates are maintained at – 18°C. Employing Neuman solution, determine the time for the meat centre to pass through the freezing zone.

(b) Determine the time required if the freezing is carried out in an air-blast freezer with air supplied at – 18°C and the heat-transfer coefficient of air is 20 $\text{Wm}^{-2}\ \text{K}^{-1}$.

All other data remain the same.

The thermophysical properties of thawed and frozen meat are given below.

Property	*Thawed Meat*	*Frozen Meat*	*Units*
Thermal conductivity	0.57	1.04	$\text{W m}^{-1}\ \text{K}^{-1}$
Specific heat	3.48	1.91	$\text{kJ kg}^{-1}\ \text{K}^{-1}$
Density	1057	961	kg m^{-3}

24.3 Solve Problem 24.2 using modified Plank's equation.

24.4 Determine the freeze-drying time for a tray of 1.2 cm thick, frozen lean steaks in contact with heating platens on both sides at 90°C. The drying chamber is maintained at 1 mm Hg pressure. The thermophysical properties of dried and frozen meat are as follows:

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$$
\begin{aligned}
&+ \rho R T \left(\left[A_1 - A_3 \left(\frac{385.15}{T} \right)^2 - A_4 \left(\frac{385.15}{T} \right)^3 \right] - E_2 \right) \\
&+ \frac{\rho^2 R T}{2} \left(\left[A_5 - A_7 \left(\frac{385.15}{T} \right)^2 \right] - E_3 \right) \\
&+ \frac{\rho^3 R T}{3} \left(\left[A_8 - A_{10} \left(\frac{385.15}{T} \right)^2 \right] - E_4 \right) \\
&+ \frac{\rho^4 R T}{4} [A_{11} - E_5] + \frac{\rho^6 R T}{6} [A_{13} - E_7] \\
&+ \frac{\rho^8 R T}{8} [A_{15} - E_6] \qquad \text{(A.1.5.)}
\end{aligned}
$$

$$
\begin{aligned}
s = s_o &+ \int_{T_o}^{T} (C_p^{\rho} - R) \frac{dT}{T} - R \ln \left[\frac{RT\rho}{P_o} \right] \\
&- \rho R \left[A_1 - A_3 \left[\frac{385.15}{T} \right]^2 - A_4 \left[\frac{385.15}{T} \right]^3 \right] \\
&- \frac{\rho^2 R}{2} \left[A_5 - A_7 \left[\frac{385.15}{T} \right]^2 \right] \\
&- \frac{\rho^3 R}{3} \left[A_8 - A_{10} \left[\frac{385.15}{T} \right]^2 \right] - \frac{\rho^4 A_{11} R}{4} \\
&- \frac{\rho^6 R A_{13}}{6} - \frac{\rho^8 R A_{15}}{8} \qquad \text{(A.1.6)}
\end{aligned}
$$

Correlation for Saturated Liquid Density

$$
\rho_L - \rho_c = \sum_{i=1}^{6} D_i (T_c - T)^{i/3} \qquad \text{(A.1.7)}
$$

where ρ_L is in kg/dm^3, and the constants are as follows:

$D_1 = 0.2477199$ $D_2 = -0.1480948$ $D_3 = 0.008001550$

$D_4 = -0.01962269$ $D_5 = 0.0023223$ $D_6 = -0.0001057677$

Zero-Pressure Constant Volume Specific Heat

$$
C_v^o = C_{v1} + C_{v2} T + C_{v3} T^2 + C_{v4} T^3 \qquad \text{(A.1.8)}
$$

where the units of specific heat are in kJ/kg.K, and the constants are

$C_{v1} = 0.0479836$ $C_{v3} = -2.94985 \times 10^{-6}$

$C_{v2} = 0.00238154$ $C_{v4} = 1.37374 \times 10^{-9}$

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Vapour Pressure Correlation

$$\ln P_s = P_1 + \frac{P_2}{T_s} + P_3 \ln T_s + [P_4 \times T_s] + P_5 \frac{(\gamma - T_s)}{T_s} \ln(\gamma - T_s) \qquad \text{(A.4.1)}$$

where P_s is in Pa, and the constants are

$P_1 = 7.1554148092$
$P_2 = -4.8189575050 \times 10^3$
$P_3 = -7.8610312200$
$P_4 = 9.0806824483 \times 10^{-3}$
$P_5 = 4.4574670300 \times 10^{-1}$
$\gamma = 381.17$

From Eq. (A.4.1) we obtain

$$\left[\frac{\mathrm{d}P_s}{\mathrm{d}T_s}\right] = P_s\left[-\frac{P_2}{T_s^2} + \frac{P_3}{T_s} + P_4 - \frac{P_5\,\gamma \ln(\gamma - T_s)}{T_s^2} - \frac{P_5\,\gamma}{T_s(\gamma - T_s)} + \frac{P_5}{(\gamma - T_s)}\right] \qquad \text{(A.4.2)}$$

Equation of State and Vapour Phase Enthalpy and Entropy

Modified Martin-Hou equation is used.

$$P = \frac{RT}{v-b} + \sum_{i=2}^{5} \frac{(A_i + B_i\,T + C_i\,e^{-kT/T_c})}{(v-b)^i} + \frac{(A_6 + B_6\,T + C_6\,e^{-kT/T_c})}{e^{\alpha v}(1 + ce^{\alpha v})} \qquad \text{(A.4.3)}$$

where P is in Pa, v is in m^3/kg, and the constants are

$R = 96.1467$ $\qquad$ $k = 4.2$
$b = 1.24855636 \times 10^{-4}$ $\qquad$ $\alpha = 8781.3417$
$A_2 = -1.16981908 \times 10^2$ $\qquad$ $A_3 = -2.92952588 \times 10^{-2}$
$A_4 = 2.41919261 \times 10^{-4}$ $\qquad$ $A_5 = -2.43458381 \times 10^{-7}$
$A_6 = 9.40022615 \times 10^{11}$ $\qquad$ $B_2 = 1.16431240 \times 10^{-1}$
$B_3 = 2.30319412 \times 10^{-4}$ $\qquad$ $B_4 = -6.79667708 \times 10^{-7}$
$B_5 = 6.30201766 \times 10^{-10}$ $\qquad$ $B_6 = -2.07580650 \times 10^9$
$C_2 = -1.18409710 \times 10^3$ $\qquad$ $C_3 = 2.48896136$
$C_4 = 0.0$ $\qquad$ $C_5 = -1.20619716 \times 10^{-6}$
$C_6 = 0.0$

From Eq. (A.4.3), we obtain

$$\left[\frac{\mathrm{d}P}{\mathrm{d}T}\right]_\rho = \frac{R}{(v-b)} + \sum_{i=2}^{5} \frac{1}{(v-b)^i}\left[B_i - \frac{C_i\,k\,e^{-kT/T_c}}{T_c}\right]$$

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Appendix B

Tables

B.1 THERMOPHYSICAL PROPERTIES OF AIR AT ATMOSPHERIC PRESSURE*

The values of μ, k, C_p and Pr are not strongly pressure-dependent and may be used over a fairly wide range of pressures.

T (K)	ρ (kg/m^3)	C_p (kJ/kg K)	μ (kg/m s $\times 10^5$)	ν (m^2/s $\times 10^6$)	k (W/mk)	α (m^2/s $\times 10^4$)	Pr
100	3.9010	1.0266	0.6924	1.923	0.009246	0.02501	0.770
150	2.3675	1.0099	1.0283	4.343	0.013735	0.05745	0.753
200	1.7687	1.0061	1.3289	7.49	0.01809	0.10165	0.739
250	1.4128	1.0053	1.488	9.49	0.02227	0.13161	0.722
300	1.1774	1.0057	1.983	15.68	0.02624	0.2216	0.708
350	0.9980	1.0090	2.075	20.76	0.03003	0.2983	0.697
400	0.8826	1.0140	2.286	25.90	0.03365	0.3760	0.689
450	0.7833	1.0207	2.284	28.86	0.03707	0.4222	0.683
500	0.7048	1.0295	2.671	37.90	0.04038	0.5564	0.680
550	0.6423	1.0392	2.848	44.34	0.04360	0.6532	0.680
600	0.5879	1.0551	3.018	51.34	0.04649	0.7512	0.680
650	0.5430	1.0635	3.177	58.51	0.04953	0.8578	0.682
700	0.5030	1.0752	3.322	66.25	0.05230	0.9672	0.684
750	0.4709	1.0856	3.481	73.91	0.05509	1.0774	0.686
800	0.4405	1.0978	3.625	82.29	0.05779	1.1951	0.689
850	0.4149	1.1095	3.765	90.75	0.06028	1.3097	0.692
900	0.3925	1.1212	3.899	99.3	0.06269	1.4271	0.696
950	0.3716	1.1321	4.023	108.2	0.06525	1.5610	0.699
1000	0.3524	1.1417	4.152	117.8	0.06752	1.6779	0.702
1100	0.3204	1.160	4.44	136.6	0.0732	1.969	0.704
1200	0.2947	1.179	4.92	159.1	0.0782	2.251	0.707
1300	0.2707	1.197	4.93	182.1	0.0837	2.583	0.705
1400	0.2515	1.214	5.17	205.5	0.0891	2.920	0.705
1500	0.2355	1.230	5.40	229.1	0.0946	3.262	0.705
1600	0.2211	1.248	5.63	254.5	0.100	3.609	0.705

* From *National Bureau of Standards* (USA), Circ, 564, 1955.

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Table B.4.2 Thermodynamic Properties of Gaseous R 744 (Carbon Dioxde)

p, MPa			*Temperature of Vapour/Gas,* K			
(T_{sat}, K)		Sat.	300	400	500	600
1.0	*v*	.0384	.0538	.0742	.0938	.1130
(233)	*h*	321.1	384.3	477.9	577.5	683.2
	s	1.377	1.617	1.886	2.108	2.300
2.0	*v*	.0190	.0254	.0364	.0465	.0564
(253.6)	*h*	322.7	373.7	472.8	574.3	681.0
	s	1.275	1.460	1.746	1.972	2.166
5.0	*v*	.0064	.0078	.0137	.0182	.0224
(287.5)	*h*	304.1	331.3	456.6	564.7	674.7
	s	1.086	1.178	1.542	1.784	1.984
10.0	*v*			.0062	.0089	.0111
	h			427.5	549.0	664.5
	s			1.356	1.628	1.838
20.0	*v*			.0026	.0043	.0055
	h			367.3	519.6	646.2
	s			1.106	1.448	1.680

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t(°C)	v,(m^3/kg)	h,(kJ/kg)	s, (kJ/kg. K)	t(°C)	v,(m^3/kg)	h,(kJ/kg)	s, (kJ/kg. K)
Saturation temperature, 30°				Saturation temperature, 32°C			
30	.0197	414.0	1.7120	35	.0191	417.1	1.7182
35	.0204	418.3	1.7262	40	.0197	421.4	1.7322
40	.0210	422.4	1.7400	45	.0203	425.5	1.7458
45	.0216	426.6	1.7534	50	.0209	429.7	1.7591
50	.0222	430.5	1.7664	55	.0214	433.8	1.7719
55	.0228	434.6	1.7791	60	.0220	437.9	1.7845
60	.0234	438.7	1.7915	65	.0225	442.0	1.7968
65	.0239	442.6	1.8036	70	.0231	446.0	1.8089
				75	.0236	450.0	1.8207
				80	.0241	454.0	1.8323
Saturation temperature, 34°C				Saturation temperature, 36°C			
35	.0179	415.7	1.7099				
40	.0185	420.0	1.7243	40	.0173	418.7	1.7162
45	.0191	424.4	1.7382	45	.0179	423.1	1.7304
50	.0196	428.5	1.7517	50	.0185	427.4	1.7442
55	.0202	432.8	1.7647	55	.0190	431.7	1.7575
60	.0207	436.9	1.7775	60	.0195	436.0	1.7704
65	.0212	441.0	1.7899	65	.0200	440.0	1.7830
70	.0217	445.0	1.8021	70	.0205	444.3	1.7954
75	.0222	449.0	1.8141	75	.0210	448.2	1.8074
80	.0227	453.0	1.8258	80	.0214	452.1	1.8193
Saturation temperature, 38°C				Saturation temperature, 40°C			
40	.0162	417.3	1.7080	40	.0151	415.9	1.6995
45	.0168	421.9	1.7225	45	.0157	420.4	1.7144
50	.0173	426.2	1.7365	50	.0162	424.9	1.7287
55	.0178	430.6	1.7501	55	.0168	429.3	1.7426
60	.0183	434.8	1.7632	60	.0172	433.6	1.7560
65	.0188	439.0	1.7760	65	.0177	438.0	1.7690
70	.0193	443.4	1.7885	70	.0182	442.1	1.7817
75	.0198	447.3	1.8008	75	.0187	446.2	1.7940
80	.0202	451.2	1.8127	80	.0191	450.5	1.8061
				85	.0195	454.8	1.8180
Saturation temperature, 42°C				Saturation temperature, 45°C			
45	.0147	419.0	1.7061	45	.0133	416.5	1.6931
50	.0152	423.5	1.7208	50	.0138	421.3	1.7084
55	.0157	428.0	1.7349	55	.0143	426.0	1.7231
60	.0162	432.4	1.7486	60	.0148	430.5	1.7372
65	.0167	436.8	1.7618	65	.0153	435.1	1.7509
70	.0172	441.2	1.7747	70	.0157	439.4	1.7641
75	.0176	445.4	1.7872	75	.0161	443.6	1.7769
80	.0180	449.5	1.7995	80	.0165	448.0	1.7895
85	.0185	451.7	1.8115	85	.0170	452.4	1.8017
				90	.0174	456.6	1.8137

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* **Ashok Babu** T P, A *Theoretical and Experimental Investigation of Alternatives to CFC 12 in Refrigerators,* Ph. D. Thesis, IIT Delhi, 1997.

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Temp.	p_{sat}	Specific Volume Liquid $v_f \times 10^3$	Specific Volume Vapour v_g	Enthalpy h_f	Enthalpy h_{fg}	Enthalpy h_g	Entropy s_f	Entropy s_{fg}	Entropy s_g
(°C)	(bar)	(m^3/kg)	(m^3/kg)	(kJ/kg)			(kJ/kg.K)		
32	7.300	1.136	0.0443	256.57	271.05	527.61	1.1996	0.8882	2.0879
34	7.720	1.143	0.0419	260.34	268.38	528.72	1.2123	0.8737	2.0860
36	8.159	1.150	0.0396	264.19	265.61	529.80	1.2251	0.8591	2.0842
38	8.616	1.157	0.0374	267.99	262.87	530.86	1.2377	0.8448	2.0825
40	9.092	1.165	0.0354	271.86	260.03	531.89	1.2505	0.8303	2.0808
42	9.587	1.172	0.0335	275.68	257.22	532.91	1.2630	0.8162	2.0792
44	10.103	1.180	0.0317	279.66	254.22	533.88	1.2760	0.8016	2.0776
46	10.638	1.188	0.0300	283.57	251.26	534.83	1.2888	0.7873	2.0760
48	11.195	1.197	0.0285	287.49	248.27	535.76	1.3015	0.7730	2.0745
50	11.774	1.206	0.0270	291.50	245.14	536.64	1.3145	0.7586	2.0731
52	12.374	1.214	0.0256	295.52	241.97	537.49	1.3275	0.7442	2.0717
54	12.997	1.224	0.0242	299.55	238.76	538.32	1.3405	0.7298	2.0703
56	13.643	1.233	0.0230	303.59	235.51	539.10	1.3536	0.7155	2.0690
58	14.313	1.243	0.0218	307.74	232.10	539.84	1.3669	0.7009	2.0678
60	15.008	1.253	0.0207	311.90	228.63	540.53	1.3803	0.6863	2.0666
62	15.727	1.264	0.0196	316.07	225.12	541.18	1.3937	0.6717	2.0654
64	16.471	1.275	0.0186	320.25	221.54	541.79	1.4073	0.6571	2.0644
66	17.242	1.286	0.0177	324.44	217.92	542.36	1.4208	0.6425	2.0634

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Table B.11.2 Superheat Table: R600a (Isobutane) Vapour

p, bar							t, °C				
(t_{sat}, °C)		sat.	– 5	0	30	45	70	95	120	145	170
0.25	v	1.3024	1.5216	1.5506	1.7242	1.8108	1.9552	2.0992	2.2432	2.3870	2.5307
(– 42.68)	h	499.59	554.13	561.85	610.61	636.59	682.24	730.82	782.28	836.55	893.56
	s	2.3365	2.5556	2.5841	2.7535	2.8372	2.9754	3.1121	3.2473	3.3812	3.5137
0.50	v	0.6804	0.7543	0.7690	0.8568	0.9006	0.9733	1.0459	1.1183	1.1906	1.2628
(–30.04)	h	515.93	553.14	560.90	609.86	635.92	681.67	730.34	781.86	836.19	893.25
	s	2.3078	2.4536	2.4823	2.6524	2.7364	2.8749	3.0113	3.1473	3.2813	3.4138
0.75	v	0.4865	0.4984	0.5084	0.5677	0.5971	0.6460	0.6943	0.7433	0.7918	0.8401
(–19.00)	h	542.90	552.12	559.93	609.10	635.24	681.10	729.85	781.45	835.83	892.93
	s	2.3578	2.3926	2.4215	2.5924	2.6767	2.8155	2.9527	3.0884	3.2226	3.3553
1.01	v	0.3551	0.3654	0.3729	0.4173	0.4394	0.4759	0.5123	0.5485	0.5845	0.6205
(–11.91)	h	540.36	551.03	558.80	608.29	634.51	680.50	729.34	781.00	835.44	892.59
	s	2.3060	2.3464	2.3755	2.5473	2.6318	2.7711	2.9086	3.0444	3.1788	3.3116
1.50	v	0.2461		0.2476	0.2784	0.2936	0.3187	0.3436	0.3684	0.3930	0.4175
(–1.40)	h	554.68		556.92	606.77	633.16	679.36	728.37	780.17	934.72	891.96
	s	2.3054		2.3136	2.4871	2.5723	2.7123	2.8503	2.9866	3.1212	3.2542
2.50	v	0.1520			0.1625	0.1720	0.1877	0.2031	0.2183	0.2334	0.2484
(13.74)	h	575.52			603.51	630.27	676.97	726.35	778.45	833.23	890.66
	s	2.3100			2.4053	2.4918	2.6334	2.7727	2.9098	3.0451	3.1786
3.50	v	0.1101			0.1127	0.1199	0.1315	0.1428	0.1540	0.1650	0.1759
(24.50)	h	590.30			600.06	627.25	674.49	724.28	776.68	831.71	889.34
	s	2.3149			2.3476	2.4357	2.5792	2.7193	2.8578	2.9938	3.1279
4.50	v	0.0863				0.0908	0.1002	0.1093	0.1183	0.1270	0.1357
(33.41)	h	602.61				624.06	671.91	722.14	744.88	830.16	887.99
	s	2.3218				2.3911	2.5368	2.6783	2.8179	2.9546	3.0893

(*Contd*)

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