## *DETAILS EXPLANATIONS*
[PART : A]

1. For defining any system certain parameters are needed. 'Properties' are those observable characteristics of the system which can be used for defining it. Thermodynamic properties are observable characteristics of the thermodynamic system. Pressure, temperature, volume, viscosity, modulus of elasticity etc. are the examples of property

State : 'State' of a system indicates the specific condition of the system. To know the characteristics of the system quantitatively refers to knowing the state of system. Thus, when the properties of system are quantitatively defined then it refers to the 'state'.

2. Thermodynamic equilibrium is a situation in which thermodynamic system does not undergo any change in its state. Generally, thermodynamic equilibrium of a system may be ensured by ensuring the mechanical, thermal, chemical and electrical equilibriums of the system.

3. A continuous and homogenous medium is called continuum . It is kind of idealization of the properties of the matter for thermodynamic analysis. Any matter is composed of several molecules which may be widely spaced apart, especially in the gas phase. continumm concept assumes a continuous distribution of mass within the matter with no empty space or voids.

4. Zeroth law of thermodynamics states that if the bodies A and B are in thermal equilibrium with a third body C separately then the two bodies A and B shall also be in thermal equilibrium with each other. This is the principle of temperature measurement.

Block diagram shown in Fig. a show the zeroth law of thermodynamics and its application for temperature measurement respectively.
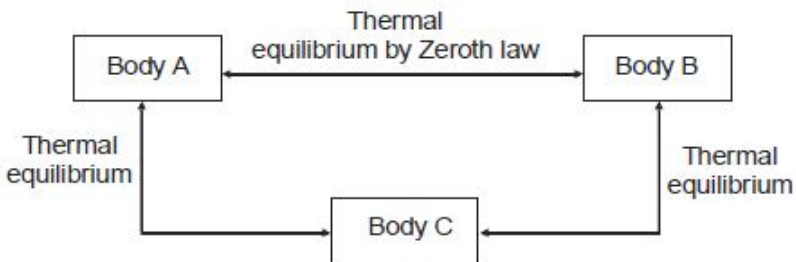
5. Degree Celsius and Fahrenheit are related as below,

$$T\ (°C) = \frac{T°(F) - 32}{1.8}$$

Substituting values.

$$T\ (°C) = \frac{98.6 - 32}{1.8} = 37°C$$

Temperature in degree celsius shall be **37°C. Ans.**

6. Let the conversion relation be $X = aC + b$

where C is temperature in degree celsius, a & B are constant and X is temperature in °X.

At freezing point, temperature = 0°C, 0°X
or, $0 = a \cdot 0 + b$
$\Rightarrow$ $b = 0$
At boiling point, temperature = 100°C, 1000°X
                $1000 = a \cdot 100 + b$
$\Rightarrow$ $a = 10$
Conversion relation

$$\boxed{X = 10 \cdot C}\ \textbf{Ans.}$$

Absolute zero temperature in °C = – 273.15°C
Absolute zero temperature in °X = – 2731.5°X

$$\boxed{-2731.5°X}\ \textbf{Ans.}$$

7. Free expansion, as the name implies refers to the unrestrained expansion of a gas. Let as take an insulated tank having two compartments separated by a partition, say A and B. Let us assume that compartment A is filled with gas while B is having vacuum. If now the partition is removed and gas allowed to occupy the whole volume of tank, then the gas expands to fill the complete volume space. New pressure of gas will be lesser as compared to initial pressure of gas occupying the compartment A. A close look at the expansion process shows that the expansion due to removal of partition is unresisted expansion due to gas expanding in vacuum. This is also known as free expansion. The reverse of free expansion is impossible and so it is an irreversible process.
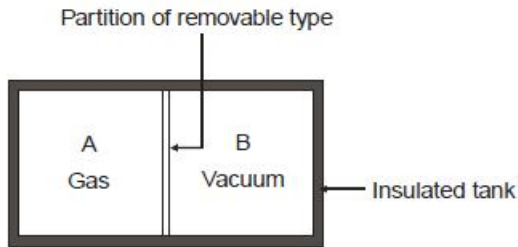
Figure : Free Expansion

8. **Flow process** is the one in which fluid enters the system and leaves it after work interaction, which means that such processes occur in the systems having open boundary permitting mass interaction across the system boundary.

    Non-flow process is the one in which there is no mass interaction across the system boundaries during the occurrence of the process.

9. (i) First law of thermodynamics does not differentiate between heat and work and assures full convertibility of one into other whereas full conversion of work into heat is possible but the vice-versa is not possible.

    (ii) First law of thermodynamics does not explain the direction of a process. Such as theoretically it shall permit even heat transfer from low temperature body to high temperature body which is not practically feasible. Spontaneity of the process is not taken care of by the first law of thermodynamics.

10. "It is impossible for a device operating in a cycle to produce net work while exchanging heat with bodies at single fixed temperature".

11. According to the third law of thermodynamics entropy of crystalline structure substance is zero at absolute zero temperature.

12. Pure substance refers to the "substance with chemical homogeneity and constant chemical composition." H_2O is a pure substance as it meets both the above requirements. Any substance, which undergoes a chemical reaction, cannot be pure substance.

13. The maximum available useful work is called Exergy while the minimum amount of energy that is not useful is called Anergy.

14. This refrigeration effect is defined by the unit of refrigeration called 'Ton' of refrigeration. 'Ton' as unit of refrigeration has been defined based on formation of ice as described ahead. "One 'Ton' of refrigeration can be defined by the amount of heat being removed from one ton of water at 0ºC to form one ton of ice at 0ºC within 24 hours." Thus, a Ton of refrigeration shall quantify the latent heat required to be removed for solidification of water at 0ºC.

15.
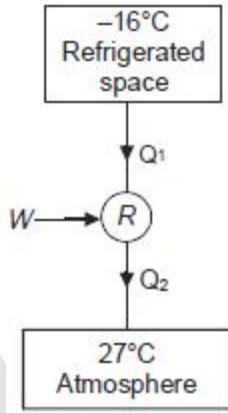
For refrigerator working on reversed Carnot cycle.

$$\frac{Q_1}{(-16+273)} = \frac{Q_2}{(27+273)}$$

$$\frac{Q_1}{Q_2} = \frac{257}{300}$$

Also it is given that $Q_1 = 500$ kJ/min
                             $Q_2 = 583.66$ kJ/min
Work input required, $W = Q_2 - Q_1$
                             $= 83.66$ kJ/min

| Work input = 83.66 kJ/min | Ans.

16. This is given by the ratio of the mass of water vapour to the mass of dry air. It can also be defined as mass of water vapour present in per kg of dry air. It is given in terms of grams per kg of dry air. Mathematically, humidity ratio or specific humidity,

$$\omega = \frac{m_v}{m_a}$$

where m_v and m_a are mass of water vapour and dry air respectively.

17. Relative humidity gives an account of moisture content in an actual mixture as compared to the mixture in saturated state at same temperature and pressure. It can be given by the ratio of actual mass of water vapour in given volume to the mass of water vapour if the air is saturated at the same temperature and pressure.

Mathematically, relative humidity (ϕ) can be given as:

$$\phi = \frac{\text{Actual mass of water vapour in given volume of mixture}}{\begin{pmatrix} \text{Maximum mass of water vapour in same volume of} \\ \text{mixture being staurated at same temperature and pressure} \end{pmatrix}}$$

18. Refrigerant is the working fluid used in refrigeration/air conditioning equipments having capability of carrying heat/rejecting heat in the form of sensible heat or latent heat. Refrigerants can be classified into two main categories as 'Primary refrigerants' and 'Secondary refrigerants'. The primary refrigerants are those refrigerants which are directly involved in refrigeration system while secondary refrigerants are first cooled by primary refrigerants and then used for imparting refrigeration. The primary refrigerants are used in vapour compression systems while secondary refrigerants are liquids used for transporting low-temperature heat energy from one place to another as done by 'brine', antifreeze agents etc.

19. Dryness fraction: It is the mass fraction of vapour in a mixture of liquid and vapour at any point in liquid-vapour mixture region. It is generally denoted by 'x'. It is also called quality of steam Wetness fraction: It is the mass fraction of liquid in a mixture of liquid and vapour at any point in liquid-vapour mixture region. It is generally denoted by 'y'.

20. Amount of superheating is quantified by degree of superheating. Degree of superheating is given by difference between temperature of steam and saturation temperature at given pressure.

Degree of superheating = Temperature of steam – Saturation temperature at given pressure

[PART : B]

21. Initial states : 100×10^3 Pa, 300 K, 5 m^3

Final states : 50×10^3 Pa, 280 K, 5 m^3.

Let initial and final mass of air be m_1 and m_2. From perfect gas equation of air,

$$m_1 = \frac{p_1 V_1}{RT_1} : m_2 = \frac{p_2 V_2}{RT_2}$$

$$m_1 = \frac{100 \times 10^3 \times 5}{287 \times 300} : m_2 = \frac{50 \times 10^3 \times 5}{287 \times 280}$$

Mass of removed, $(m_1 - m_2)$

$$= \left(\frac{100 \times 10^3 \times 5}{287 \times 300}\right) - \left(\frac{50 \times 10^3 \times 5}{287 \times 280}\right)$$

$$m_1 - m_2 = 2.696 \text{ kg}$$

Volume of this mass of air at initial states i.e 100 kPa and 300 K:

$$V = \frac{(m_1 - m_2) \cdot RT_1}{p_1}$$

$$= \frac{2.696 \times 287 \times 300}{100 \times 10^3}$$

Volume = 2.32 m³

Mass of air removed = **2.696 kg**

Volume of air at initial states = **2.32 m³**   **Ans.**

22. Isothermal process :

$$p_1 V_1 = p_2 V_2 = \text{Constant, or, } P = \frac{P_1 V_1}{V}$$

so work involved, $$W_{1-2} = \int_{V_1}^{V_2} P \cdot dV$$

$$W_{1-2} = \int_{V_1}^{V_2} \frac{P_1 V_1}{V} dV = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$W_{1-2} = P_1 V_1 \ln r$$

where r = ratio of final and initial volumes.

Pressure ⟶

23. If the energy at macroscopic level as discussed above could be separated from the total stored energy E, then the amount of energy left shall be called internal energy.

Mathematically,

Internal energy, U = (Stored energy) – (Kinetic energy) – (Potential energy) – (Magnetic energy) – (Electrical energy) – (Surface tension energy) – (Solid distortion energy).

Therefore, stored energy is summation of internal energy, potential energy, kinetic energy, magnetic, electrical, surface tension, solid distortion etc. types of energy.

Enthalpy (H) of a substance at any point is quantification of energy content in it, which could be given by summation of internal energy and flow energy. Enthalpy is very useful thermodynamic property for the analysis of engineering systems.

Mathematically, it is given as,

$$H = U + PV$$

On unit mass basis, the specific enthalpy could be given as,

$$h = u + pv$$

24. (a) It is a closed system. If the pressure on face of piston is uniform, then the work done on piston can obtained as,

$$W = p \int_1^2 dV$$

$$= 689 \times 10^3 (0.045 - 0.04)$$

Work done on piston, $W = 3445$ J or 3.445 kJ

Work done on piston = **3.445 kJ Ans.**

(b) Paddle work done on the system = – 4.88 kJ

Net work of system $\Rightarrow W_{net} = W_{piston} + W_{paddle}$

$$= 3445 - 4880$$

$$W_{net} = -1435 \text{ J}$$

Work done on system = 1435 J or 1.435 kJ.

Work done on system = **1.435 kJ Ans.**

25. Thus $\dfrac{Q_1}{Q_2} = \dfrac{673}{288}$

and $Q_1 - Q_2 = 200$ kJ

From equations 1 and 2, upon solving

$$Q_1 = 349.6 \text{ kJ}$$

and $Q_2 = 149.6$ kJ

Heat to be supplied = 349.6 kJ

26. *Vapour compression system*

Merits :

1. Thermodynamic cycle involved in vapour compression system is very close to Carnot cycle hence its COP is quite high of the order of 3–4   while for air refrigeration COP is generally less than unity.

2. Refrigerant required is smaller in vapour compression system because the heat is carried away by latent heat of vapour and so the amount of liquid refrigerant circulated per ton is less. Also size of evaporator is small.

3. Operating cost of vapour compression system is very less as compared to air refrigeration system on ground.

4. Refrigerated space temperature can be easily regulated by controlling the throttle valve.

Demerits :

1. Initial investment is too large as compared to air refrigeration system.

2. There are chances of leakage of refrigerant as large pressure differential exists throughout the cycle.

3. Air leakage into refrigerant offers reduced performance of vapour compression system and so it should be prevented.

Merits and demerits of Vapour absorption system over Vapour compression system.

The merits and demerits of vapour absorption system over vapour compression system are listed as under:

Merits

1. Vapour absorption system has no moving component throughout cycle, so it works quietly and has reduced maintenance requirement.

2. Lesser work input is required for pump as compared to compression in vapour compression system

3. Vapour absorption system may be run with some installation rejecting waste heat, which may be used to run generator, thus energy conservation.

4. Vapour absorption system is well suited for large refrigeration capacity applications such as even above 1000 tons, which is difficult in vapour compression systems.

5. Vapour absorption systems offer smaller variation with load whereas COP of vapour compression system varies largely with load. At smaller loads the COP of absorption system is more than that of vapour absorption system.

Demerits

1. Size of absorption system is large and is bulky to handle.

2. If heating source (such as electric heating) is costly then the vapour absorption system becomes costly than vapour compression system.

3. At very high loads the COP of vapour compression system is more than that of vapour absorption system.

27. Ideal gas equation, $Pv = RT$

Let us consider two variable (v, T) to be independent and P as dependent variable

$$P = P(v, T) = \frac{RT}{v}$$

By cyclic relation,

$$\boxed{\left(\frac{\partial P}{\partial v}\right)_T \cdot \left(\frac{\partial v}{\partial T}\right)_p \cdot \left(\frac{\partial T}{\partial P}\right)_v = -1}$$

Let us find the three partial derivatives separately and then substitute.

$$\left(\frac{\partial P}{\partial v}\right)_T = \frac{-RT}{v^2}, \left(\frac{\partial v}{\partial p}\right)_p = \frac{R}{p}, \left(\frac{\partial T}{\partial P}\right)_v = \frac{v}{R}$$

Substituting

$$\left(\frac{-RT}{v^2}\right)\left(\frac{R}{P}\right)\left(\frac{v}{R}\right) = \frac{-RT}{Pv} = -1 \text{ Hence proved.}$$

28. $(COP)_{HP} = \dfrac{Q_1}{W}$

$W = Q_1 - Q_2$

$$(COP)_{refrigerator} = \frac{\text{Desired effect}}{\text{Net work}} = \frac{Q_2}{W}$$

Here

$$W = Q_1 - Q_2$$

or

$$(COP)_{refrigerator} = \frac{Q_2}{Q_1 - Q_2}$$

COP values of heat pump and refrigerator can be interrelated as:

$$(COP)_{HP} = (COP)_{refrigerator} + 1$$

29. A refrigerant being used in refrigeration should have following thermodynamic properties, physical properties and working properties.

(i) Boiling temperature of refrigerant should be quite low at atmospheric conditions for effective refrigeration. For refrigerants having higher boiling temperatures at atmospheric conditions the compressor is run at higher vacuum.

(ii) For an ideal refrigerant the freezing temperature of refrigerant should be quite low so as to prevent its freezing at evaporator temperature. Freezing point temperature should be less than evaporator temperature. For example, refrigerant R–22 has freezing point of – 160ºC and normally most of refrigerants have freezing point below – 30ºC.

(iii) Critical temperature of the ideal refrigerant should be higher than the condenser temperature for the ease of condensation.

(iv) Refrigerant should have large latent heat at evaporator temperature as this shall increase the refrigerating capacity per kg of refrigerant.

(v) Refrigerant should have small specific volume at inlet to compressor as this reduces compressor size for same refrigeration capacity.

(vi) Specific heat of refrigerant in liquid form should be small and it should be large for refrigerant in vapour form, since these increase the refrigerating capacity per kg of refrigerant.

(vii) Thermal conductivity of refrigerant should be high.

(viii) Viscosity of refrigerants should be small for the ease of better heat transfer and small pumping work requirement.

(ix) Refrigerant should be chemically inert and non toxic.

   (x) Refrigerant should be non-flammable, non explosive and do not have any harmful effect upon coming in contact with material stored in refrigeration space.

   (xi) Refrigerant may have pleasant distinct odour so as to know about its leakage.

   (xii) Refrigerant should be readily available at lesser price.

30. ***Dry air:*** Atmospheric air having 79% nitrogen and 21% oxygen by volume is considered dry air. Its, molecular weight is taken as 29.

   Moist air: Moist air is the mixture of dry air and water vapour in which dry air is treated as if it were pure component. Quantity of water vapour present in the mixture depends upon the temperature of air and it may vary from zero in dry air to the maximum quantity when mixture is saturated of water vapour (called saturation capacity of air). Moist air is assumed to behave as ideal gas for the purpose of analysis. Mixture pressure is the sum of partial pressures of dry air and water vapour. When the partial pressure of water vapour corresponds to the saturation pressure of water at mixture temperature then mixture is said to be saturated. Saturated air is the mixture of dry air and saturated water vapour. When the temperature of mixture of air and vapour is above the saturation temperature of water vapour then the vapour is called superheated vapour.

31.

$$COP = \frac{T_1}{T_2 - T_1} = \frac{246}{T_2 - 246}$$

Also,

$$COP = \frac{\text{Refrigeration effect}}{\text{Work input}}$$
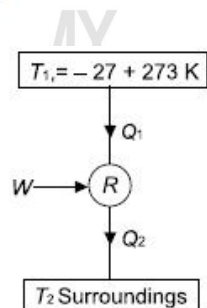
$$= \frac{1 \times 3.5}{3 \times 0.7457}$$

$$COP = 1.56$$

$$\boxed{COP = 1.56} \text{ Ans.}$$

Equating COP,

$$1.56 = \frac{246}{T_2 \times 246}$$

$$T_2 = 403.69 \text{ K}$$

$$\boxed{\text{Temperature of surroundings} = 403.69 \text{ K}} \text{ Ans.}$$

$T_1 = -27 + 273$ K

$W \rightarrow$ R

Q_1

Q_2

T_2 Surroundings

32. This type of cooling is needed in hot and dry climates. Evaporative cooling has arrangement for spray of spraying liquid water into air or passing air through a pad soaked with water. Due to less humidity of air it shall evaporate some amount of water in its contact and thus reduce its temperature because of heat extracted for evaporation of water. This air leaving evaporative cooler shall have temperature less than inlet air temperature and also due to moisture being picked up the humidity ratio gets increased.

[PART : C]

33. Carnot cycle is a reversible thermodynamic cycle comprising of four reversible processes.

Thermodynamic processes constituting Carnot cycle are;

(i) Reversible isothermal heat addition process, (1–2, Qadd)

(ii) Reversible adiabatic expansion process (2–3, Wexpn +ve)

(iii) Reversible isothermal heat release process (3–4, Qrejected)

(iv) Reversible adiabatic compression process (4–1, Wcompr –ve)

Carnot cycle is shown on P–V diagram between states 1, 2, 3 4, and 1. A reciprocating pistoncylinder assembly is also shown below P–V diagram.

Process 1 –2 is isothermal heat addition process of reversible type in which heat is transferred to system isothermally. In the piston cylinder arrangement heat Qadd can be transferred to gas from a constant temperature source T1 through a cylinder head of conductor type. First law of thermodynamics applied on 1–2 yields;

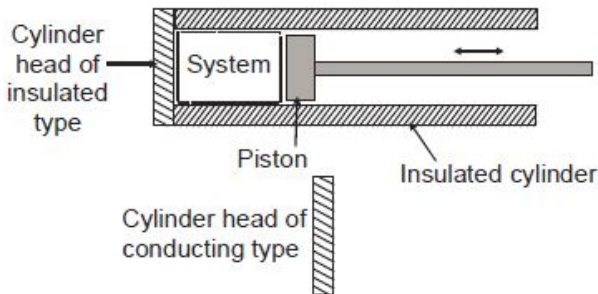$$Qadd = U2 – U1 + W1–2$$

Reversible adiabatics
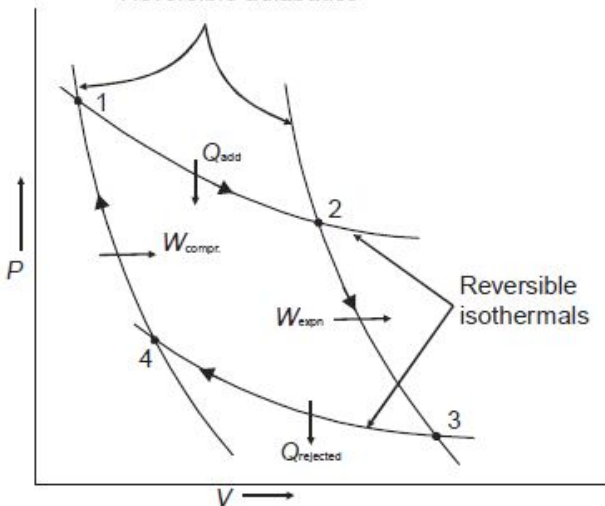
Figure : Carnot Cycle

For the perfect gas as working fluid in isothermal process no change in internal energy occurs,

therfore $U2 = U1$

and $Qadd = W1–2$

Process 2–3 is reversible adiabatic expansion process which may be had inside cylinder with cylinder head being replaced by insulating type cylinder head so that complete arrangement is insulated and adiabatic expansion carried out.

During adiabatic expansion say work Wexpn is available,

$Q2–3 = 0$

From first law of thermodynamics;

$0 = (U3 – U2) + Wexpn$

or $Wexpn = (U2 – U3)$

Process 3–4 is reversible isothermal heat rejection for which cylinder head of insulating type may be replaced by conducting type as in 1–2 and heat (Qrejected) be extracted out isothermally. From first law of thermodynamics applied on process 3–4,

$–Qrejected = (U4 – U3) + (–W3–4)$

for perfect gas internal energy shall remain constant during isothermal process. Thus, $U3 = U4$

$–Qrejected = –W3–4$

or $Qrejected = W3–4$

Process 4–1 is the reversible adiabatic compression process with work requirement for compression.

In the piston cylinder arrangement cylinder head of conducting type as used in 3–4 is replaced by insulating type, so that the whole arrangement becomes insulated and adiabatic compression may be realized,

From first law applied on process 4–1

For adiabatic process; Q4–1 = 0

Ò! 0 = (U1 – U4) + (–Wcompr)

or Wcompr = (U1 "U4)

Efficiency of reversible heat engine can be given as;

$$\eta_{rev,}\ HE\ =\ \frac{Network}{Heat\ supplied}$$

Here, Network = W$_{expn}$ – W$_{compr}$

and heat is supplied only during process 1–2, therefore

$$heat\ supplied = Q_{add}$$

Substituting in the expression for efficiency.

$$\eta_{rev,\ HE}\ =\ \frac{W_{expn} - W_{compr}}{Q_{add}}$$

Also for a cycle

$$\sum_{cycle} W\ =\ \sum_{cycle} Q$$

so

$$W_{net} = Q_{add} - Q_{rejected}$$

Hence

$$\eta_{rev,\ HE}\ =\ 1 - \frac{Q_{rejected}}{Q_{add}}$$

As the heat addition takes place at high temperature, while heat rejection takes place at low temperature, so writing these heat interactions as Q$_{high}$, Q$_{low}$ we get,

$$\boxed{\eta_{rev,\ HE} = 1 - \frac{Q_{low}}{Q_{high}}}$$

$$\boxed{\eta_{Carnot} = 1 - \frac{Q_{low}}{Q_{high}}}$$

34. Vapour compression cycle based refrigeration systems are extensively used in refrigeration systems. These cycles are used for most of small domestic and large industrial applications.

The vapour compression cycle has the refrigeration being circulated in closed circuit through compressor, condenser, throttle valve or expansion valve and evaporator as shown in figure. 18.5. Refrigeration (in gas/vapour) is compressed isentropically in compressor from state 1 to 2. High pressure and high temperature refrigeration enters the condenser at state 2 where its condensation occurs and refrigerant is available in liquid form at state 3. Refrigerant in the form of saturated liquid at high pressure is passed through expansion valve where isenthalpic expansion occurs. Refrigerant leaving expansion valve at state 4 is in the form of low pressure wet mixture of liquid and vapour. Low pressure liquid vapour mixture is passed through evaporator section in which refrigerant picks up heat from surroundings thereby showing refrigeration effect. As a result of this heat absorption the liquid vapour mixture refrigerant gets transformed into dry gaseous refrigerant in case of dry compression as shown in Fig. 18.5(b). Compression of dry and saturated or superheated refrigerant is called dry compression.
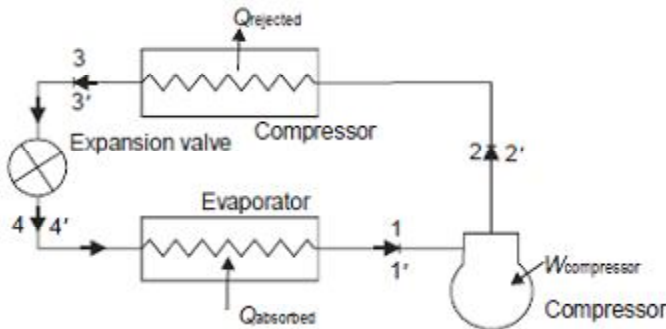
Compression of dry refrigerant yields superheated state of refrigerant as shown by state 2. (Fig. 18.5 (b))

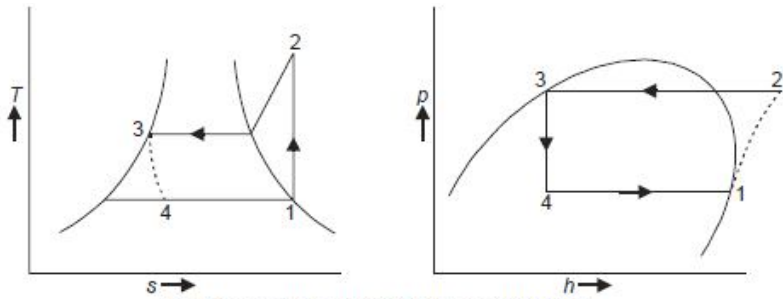1 – 2 or 1' – 2' :- Isentropic compression.

2 – 3 or 2' – 3' :- Isentropic heat rejection.

3 – 4 or 3' – 4' :- Isenthalpic expansion process or throtting process.
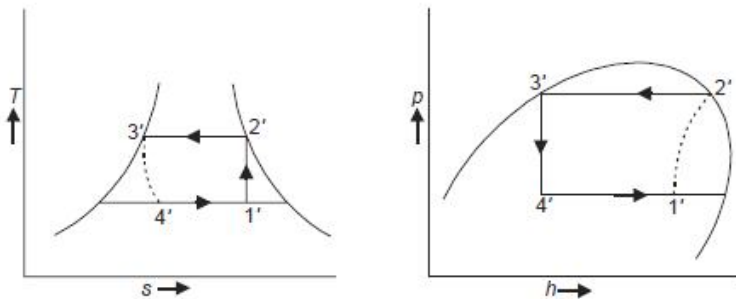
4 – 1 or 4' – 1' :- Isobaric heat absorption.

(a) Schematic for vapour compression cycle

(b) T-s and p-h diagram for dry compression

(c) T-s and p-h diagram for wet compression

Figure : Vapour Compression Cycle

Wet Compression :

It is also possible that the refrigerant is in wet state i.e. liquid-vapour mixture at inlet of compressor, state 1'. Compression of wet mixture gets transformed into dry refrigerant (gaseous form) as shown by state 2'. Dry refrigerant at high pressure and high temperature is passed through condenser where refrigerant gets condensed and condensate is available in saturated liquid form at high pressure. Subsequently refrigerant is throuttled from high pressure to low pressure inside expansion valve from state 3' to 4'. Low pressure refrigerant in wet state is passed through evaporator from state 4' to 1' where it pick up heat from surrounding and some of its liquid fraction gets transformed into vapour but it does not become dry(gas) refrigerant at inlet to compressor. this wet refrigerant is compressed inside compressor.

T-s and p-h diagram for vapour compression cycle shows that the refrigeration capacity can be increased by subcooling the condensate before it enters the expansion valve and also by increasing the degree of expansion in expansion valve.

Here in second option the condensation temperature is fixed and evaporator temperature is lowered which increases expansion ratio and compression ratio. This increase in compression ratio shows poor volumetric efficiency in single stage dry compression.

The temperature after compression also ets increased with increased compression ratio which may be harmful to refrigerant properties. To regulate the excessive rise in refrigerant temperature the intercooling during compression with multistage compression is used. This intercooled compression may be devised using refrigerant itself as intercooling medium.

Thermodynamic analysis :

For sample vapour compression cycle shown in Fig. 18.6 the COP, refrigeration effect and work input can be estimated based on following assumptions :

(i)   All process of refrigeration cycle are internally reversible expect the expansion through valve which is throttling process and is irreversible.

(ii) Compressor and expansion value have no heat interaction with surroundings during their operation i.e. they operate adiabatically.

(iii) Refrigerant leaving condenser is saturated liquid.

(iv) Refrigerant entering compressor is saturated vapour in case of 'dry compression' and 'liquid-vapour mixture in case of' 'wet compression.'

(v)   Changes in kinetic energy and potential energy are negligible.

For mass flow rate of refrigerant being given by 'm' kg/s the refrigeration capacity or refrigeration effect. $Q_{absorbed} = m(h_1 - h_4)$, in dry compression.

$$Q_{absorbed} = m(h_{1'} - h_{4'}) \text{ in wet compression.}$$

Net work = Work input

$$W_{compressor} = m(h_2 - h_1), \text{ in dry compression.}$$

$$W_{compressor} = m(h_{2'} - h_{1'}) \text{ in wet compression.}$$

Therefore, $COP = \dfrac{m(h_1 - h_4)}{m(h_2 - h_1)} = \left\{\dfrac{(h_1 - h_4)}{(h_2 - h_1)}\right\}$, in dry compression

$$COP = \dfrac{m(h_{1'} - h_{4'})}{m(h_{2'} - h_{1'})} = \left\{\dfrac{(h_{1'} - h_{4'})}{(h_{2'} - h_{1'})}\right\}, \text{ in dry compression}$$

35.

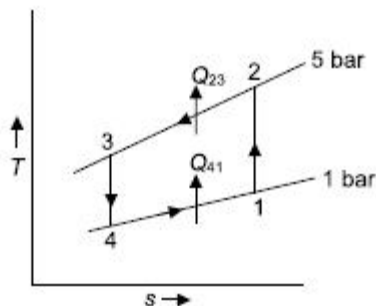Fig. 18.25

Given

$$T_1 = 7°C \text{ or } 280 \text{ K}$$
$$T_3 = 27°C \text{ or } 300 \text{ K}$$
$$P_1 = 1 \text{ bar}, P_2 = 5 \text{ bar}$$

For isentropic compression process:

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1}$$

$$T_2 = 280 \times (5)^{\frac{(1.4-1)}{1.4}}$$
$$T_2 = 443.47 \text{ K}$$

For isentropic expansion process:

$$\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{T_3}{T_4}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\left(\frac{T_3}{T_4}\right) = (5)^{\frac{(1.4-1)}{1.4}}$$

$$T_4 = 189.42 \text{ K}$$

Heat rejected during process 2–3, $Q_{23} = m \times C_p \times (T_2 - T_3)$

$$= m \times 1.005 \times (443.47 - 300)$$

$$Q_{23} = 144.19 \ m$$

Refrigeration process, heat picked during process 4–1, $Q_{41} = m \times C_p \times (T_1 - T_4)$

$$= m \times 1.005 \times (280 - 189.42)$$

$$Q_{41} = 91.03 \ m$$

Net work $= Q_{23} - Q_{41}$

$$= (144.19 - 91.03) \ m$$

$$W = 53.16 \ m \text{ kJ}$$

$$COP = \frac{\text{Refrigeration effect}}{\text{Net work}} = \frac{91.03m}{53.16m} = 1.71$$

$$\boxed{COP = 1.71} \qquad \text{Ans.}$$

36.

Solution:

Here, $W_1 : W_2 : W_3 = 3 : 2 : 1$

Efficiency of engine, HE_1,

$$\frac{W_1}{Q_1} = \left(1 - \frac{T_2}{1100}\right) \Rightarrow Q_1 = \frac{1100.W_1}{(1100 - T_2)}$$

for HE_2 engine,

$$\frac{W_2}{Q_2} = \left(1 - \frac{T_3}{T_2}\right)$$

for HE_3 engine,

$$\frac{W_3}{Q_3} = \left(1 - \frac{300}{T_3}\right)$$

From energy balance on engine, HE_1

$$Q_1 = W_1 + Q_2 \Rightarrow Q_2 = Q_1 - W_1$$

Above gives,

$$Q_2 = \left\{\frac{1100\,W_1}{(1100 - T_2)} - W_1\right\} = W_1\left\{\frac{T_2}{1100 - T_2}\right\}$$

Substituting Q_2 in efficiency of HE_2

$$\frac{W_2}{W_1\left(\frac{T_2}{1100 - T_2}\right)} = \left(1 - \frac{T_3}{T_2}\right)$$

or

$$\frac{W_2}{W_1} = \left(\frac{T_2}{1100 - T_2}\right)\left(\frac{T_2 - T_3}{T_2}\right) = \left(\frac{T_2 - T_3}{1100 - T_2}\right)$$

or

$$\left\{\frac{2}{3} = \left(\frac{T_2 - T_3}{1100 - T_2}\right)\right\}$$

or

$$2200 - 2T_2 = 3T_2 - 3T_3$$

$$\boxed{5T_2 - 3T_3 = 2200}$$

Energy balance on engine HE_2 gives,

$$Q_2 = W_2 + Q_3$$

Substituting in efficiency of HE_2,

$$\frac{W_2}{(W_2 + Q_3)} = \left(\frac{T_2 - T_3}{T_2}\right)$$

or

$$W_2.\,T_2 = (W_2 + Q_3)\,(T_2 - T_3)$$

or

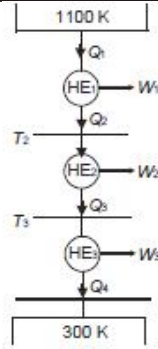$$Q_3 = \frac{W_2\,T_3}{(T_2 - T_3)}$$

Fig. 4.25

Substituting Q_3 in efficiency of HE_3,

$$\frac{W_3}{\left(\dfrac{W_2 T_3}{T_2 - T_3}\right)} = \frac{T_3 - 300}{T_3}$$

$$\frac{W_3}{W_2} = \left(\frac{T_3}{T_2 - T_3}\right)\left(\frac{T_3 - 300}{T_3}\right)$$

$$\frac{1}{2} = \frac{T_3 - 300}{T_2 - T_3}$$

$$3T_3 - T_2 = 600$$

Solving, equations of T_2 and T_3, $T_3 = 433.33$ K

$$T_2 = 700 \text{ K}$$

Intermediate temperatures: 700 K and 433.33 K Ans.

37. Above problem can be solved using steady flow energy equations upon hot water-flow,

$$Q + m_1\left(h_1 + \frac{C_1^2}{2} + gz_1\right) = w + m_2\left(h_2 + \frac{C_2^2}{2} + gz_2\right)$$

Here total heat to be supplied = 500×50

Heat lost by water (–ve), Q = –25000 kcal/hr.

There shall be no work interaction and change in kinetic energy, so, steady flow energy equation shall be,

$$Q + m_1(h_1 + gz_1) = m_2(h_2 + gz_2)$$

Here, $m_1 = m_2 = m$

$h_1 = 80$ kcal/kg,

$h_2 = 45$ kcal/kg

$Q + m(h_1 - h_2) = m(gz_2 - gz_1)$

$-(25000 \times 10^3 \times 4.18) + m(80 - 45) \times 10^3 \times 4.18 = m \times (9.81 \times 10)$

m = 714.76 kg/hr

or Water circulation rate = 11.91 kg/min.

38. Reversible processes as described "the thermodynamic processes occurring in the manner that states passed through are always in thermodynamic equilibrium and no dissipative effects are present." Any reversible process occurring between states 1–2 upon reversal, while occurring from 2–1 shall not leave any mark of process ever occurred as states traced back are exactly similar to those in forward direction. Reversible processes are thus very difficult to be realized and also called ideal processes. All thermodynamic processes are attempted to reach close to the reversible process in order to give best performance.

Thermodynamic process which does not fulfil conditions of a reversible process are termed irreversible processes. Irreversibilities are the reasons causing process to be irreversible. Generally, the irreversibilities can be termed as internal irreversibility and external irreversibility. Internal irreversibility is there because of internal factors whereas external irreversibility is caused by external factors at the system-surrounding interface. Generic types of irreversibilities are due to;

(i) Friction,

(ii) Electrical resistance,

(iii) Inelastic solid deformations,

(iv) Free expansion

(v) Heat transfer through a finite temperature difference,

(vi) Non equilibrium during the process, etc.

39. Here, $\otimes S_{universe} = \otimes S_{block\ 1} + \otimes S_{block\ 2}$

Two blocks at different temperatures shall first attain equilibrium temperature. Let equilibrium temperature be T_f.

Then from energy conservation

$1 \times 0.393 \times (423.15 - T_f) = 0.5 \times 0.381 \times (T_f - 273.15)$

$T_f = 374.19$ K

Hence, entropy change in block 1, due to temperature changing from 423.15 K to 374.19 K.

$$\Delta S_1 = 1 \times 0.393 \times \ln\left(\frac{374.19}{423.15}\right) = -0.0483 \text{ kJ/K}$$

Entropy change in block 2

$$\Delta S_2 = 0.5 \times 0.381 \times \ln\left(\frac{374.19}{423.15}\right) = 0.0599 \text{ kJ/K}$$

Entropy change of universe = 0.0599 – 0.0483 = 0.0116 kJ/K

Entropy change of universe = 0.0116 kJ/K