UNIT 12 REFRIGERATION

Structure

12.1 INTRODUCTION

Refrigeration is the process of maintaining a body, or a zone in space, at a temperature lower than the local surroundings temperature. Hence, refrigeration requires the continuous transfer of heat from the body, or the zone, to be 'refrigerated' to the surroundings at a relatively high temperature. It has been explained in Unit 6 that such a job can be done by a 'Reversed Heat Engine' (a refrigerator). However, this is not the only method available for providing refrigeration. There are several other methods used in practice. The most commonly used methods are: (i) Vapour Compression Refrigeration System, (ii) Gas Compression Refrigeration System, (iii) Vapour Absorption Refrigeration System and (iv) Steam Jet Refrigeration System. In this Unit, while the first two systems in the above list are discussed in detail from the view point of thermodynamics, only a brief mention of the working of the other systems, their application advantages and disadvantages is made.

Objectives

After reading this Unit, you must be able to

* understand the working of various refrigeration systems and the principles on which they operate,
* represent the basic cycles on $T-s$ and $p-h$ diagrams and read the properties at each state using the property Tables or Charts,
* to analyse a cycle and evaluate the performance of the system using the principles of thermodynamics studied earlier,
* distinguish between wet compression and dry compression cycles,
* know the effect of irrversibilities on the system performance especially on the performance of the compressor in a VCRS,
* distinguish between VCRS and VARS,
* understand and analyse air cycle refrigeration system operating on reversed Brayton cycle,
* describe and analyse an aircraftcooling system, and
* understand the working of steam jet refrigeration system.

12.2 VAPOUR COMPRESSION REFRIGERATION SYSTEM

A vapour compression system is used in a majority of cases where refrigeration is required for different purposes. Its popularity is mainly because of its high performance characteristics and trouble free working for long periods of time.

Shown in Figure 12.1 is the schematic of a simple vapour compression refrigeration system (VCRS). This is essentially the same as that in Figure 6.3 that explains the working of a refrigerator. Here, the working substance, called the 'refrigerant', undergoes a thermodynamic cycle while transferring heat from the low temperature reservoir (refrigerated body or zone) to the high temperature reservoir (surroundings) with the help o external work.

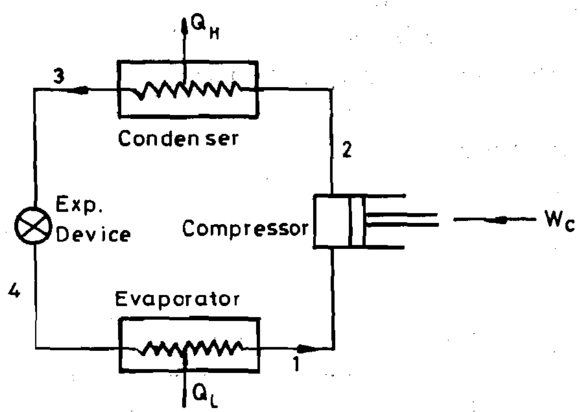

Figure 12.1 : Schematic of a simple VCRS

The VCRS, shown in Figure 12.1, has four major components, namely, the EVAPORATOR, the COMPRESSOR, the CONDENSER and the EXPANSION DEVICE. The ideal or the basic cycle on which such a system operates is shown in Figure 12.2 on both $T-s$ and $p-h$ diagrams.

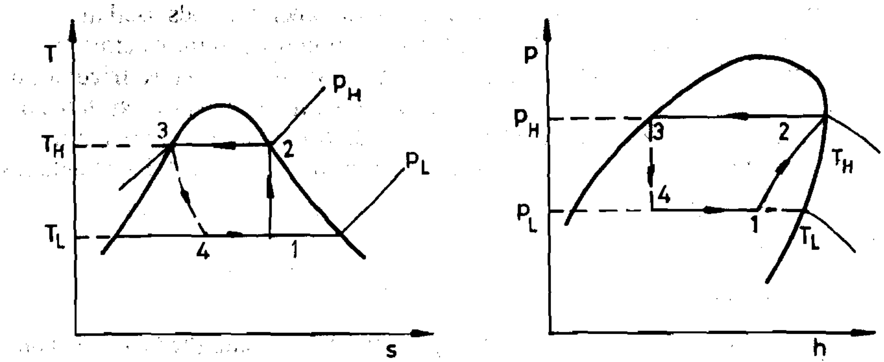

Figure 12.2 : VCR Cycle (Wet compression)

Wet refrigerant vapour at state 1 from the evaporator enters the compressor. This vapour is compressed reversibly and adiabatically to saturated vapour at state 2. Let the work required by the compressor be W. At state 2 the pressure is equal to the saturation pressure of the refrigerant corresponding to the condenser temperature. The high pressure vapour at state 2 is condensed to saturated liquid at the same pressure as it passes through the condenser and let Q_H be the heat transferred from the refrigerant to the surroundings at T_H. The saturated liquid at state 3 is then throttled through the expansion device to low quality vapour at state 4. At state 4 the pressure of the refrigerant vapour corresponds to its saturation pressure at the evaporator temperature. This low quality vapour, as it flows through the evaporator, picks up heat to maintain the refrigerated space at a temperature T_L, and then leaves it at state 1 as high quality vapour. Let Q_L be the heat transferred to the refrigerant at the evaporator.

12.2.1 Refrigerants

The design of a VCRS is greatly influenced by the properties of the refrigerant used in the system. The most widely used refrigerants are the halocarbons of the FREON family. The chemical composition of each halocarbon in the above family is identified by the number ascribed to it. For example the commonly used refrigerant in the domestic refrigerators is FREON-12 or R-12 which is nothing but 'dichloro difluoro methane ($CCl_2 F_2$)'. Similarly, FREON-22 or R-22 which is commonly used in window-airconditioners is 'manochloro difluoro methane ($CHClF_2$). The other refrigerants in the family are : R-11, R-13, R-14, R-22, R-113 and R-114. The selection of the refrigerant depends upon the evaporator temperature needed, the type of compressor used in the system and the capacity of the system.

It is not always the halocarbons that are used as refrigerants. Many inorganic substances such as NH_3, H_2O, Air, CO_2 and SO_2 are also used as refrigerants in certain specific situations.

Properties of the refrigerants are available in the form of Tables (similar to steam Tables in Unit 11) or charts in hand books on refrigeration and air-conditioning.

Given in Tables 12.1 and 12.2, at the end of this Unit, are the extracts of property Tables for Freon-12 and Ammonia.

12.2.2 Performance of Vapour Compression System

The performance of a VCRS is expressed, as explained in Unit 6, by its Coefficient of Performance (COP).

$$\text{COP} = \frac{(\text{Heat transfer to the refrigerant at the evaporator})}{\text{Work input}}$$

or

$$\text{COP} = \frac{Q_L}{W} \qquad (12.1)$$

Applying First Law to the cycle

$$W = Q_H - Q_L \text{ and hence,}$$

$$\text{COP} = \frac{Q_L}{(Q_H - Q_L)} \qquad (12.2)$$

The following relations that give the magnitudes of heat and work interactions may be obtained by the application of SFEE (based on unit mass) to the four components of the VCRS. The state points marked on the schematic (Figure 12.1) and the cycle diagram (Figure 12.2), are compatible.

$$Q_L = h_1 - h_4 \qquad (12.3)$$

$$W = h_2 - h_1 \qquad (12.4)$$

$$Q_H = h_2 - h_3 \qquad (12.5)$$

Also,

$$h_3 = h_4 \qquad (12.6)$$

Using equations (12.1) to (12.6) it may be further written that:

$$\text{COP} = \frac{(h_1 - h_4)}{(h_2 - h_1)} \qquad (12.7)$$

$$= \frac{(h_1 - h_4)}{[(h_2 - h_3) - (h_1 - h_4)]} \qquad (12.8)$$

Refrigerating Effect

In refrigeration technology, the cooling produced per kg refrigerant is referred to as the refrigerating effect. Therefore,

$$\text{Ref.effect} = (h_1 - h_4) \text{ kJ/kg}$$

Capacity of the VCRS

The capacity of a VCRS, expressed in kW, is the rate at which cooling is produced in the evaporator. Therefore,

$$\text{Capacity} = (\text{mass flow rate of the refrigerant}) \times (\text{Ref.effect}).$$

i.e.
$$\text{Capacity} = \dot{m} \times (h_1 - h_4) \text{ kW} \quad (12.9)$$

where $\dot{m}$ is in kg/s and h is in kJ/kg.

Ton of Refrigeration

This is the unit of capacity of the refrigeration system commonly used in British System of Units. A refrigeration system is said to have a capacity of one ton if it can provide cooling at a rate of 200 BTU/min. This is so because a system cooling at this rate can completely freeze one ton (2000 lbs) of water at 32° F to ice at the same temperature in a day (24 hours). This can easily be checked assuming the enthalpy of fusion of water to be 144 BTU/lb.

$$\text{capacity} = \frac{(\text{cooling produced})}{(\text{time taken})}$$

$$1 \text{ ton capacity} = \frac{(2000 \text{ lb} \times 144 \text{ BTU/lb})}{24 \times 60 \text{ min}}$$

$$= 200 \text{ BTU/min.}$$

Because of its long time continuous use in the refrigeration industry, this unit of capacity has come to stay and is used widely everywhere irrespective of the system of units used. It is advised not to use this unit of capacity in SI system of Units. Here the capacity is best expressed in kW. However, to have a feel for the relative magnitudes of '1 ton' and '1 kW' the following conversion factor can be used.

$$1 \text{ ton} = 200 \text{ BTU/min} = 3.5 \text{ kW.}$$

Compressor Power

This is the power to be supplied to the system from the surroundings .

$$\text{Power} = (\text{mass flow rate of the refrigerant}) \times (\text{work per kg})$$

i.e.
$$P = \dot{m} \times (h_2 - h_1) \text{ kW} \quad (12.10)$$

where $\dot{m}$ is in kg/s and h is in kJ/kg.

SAQ 1

The COP of a wet compression ideal VCRS of capacity 5 tons is given to be 3. Work supplied to the compressor is 20 kJ/kg. Find the mass flow rate of the refrigerant and the refrigerating effect.

12.2.3 Carnot Refrigeration Cycle

The simple VCRS shown in Figure 12.1 is irreversible in nature mainly because of the throttling process 3-4. If this process could be replaced by a reversible expander and the work obtained by this could supplement the work required by the compressor we get a reversed Carnot cycle operating between T_H and T_L. Under such circumstances the COP of

the refrigeration cycle shall be the maximum. Such a reversed Carnot cycle is shown in the $T-s$ diagram of Figure 12.3. In this Figure while 1-2-3-4*-1 is the Carnot cycle, 1-2-3-4-1 is the corresponding simple VCRS, both operating between the same temperature limits of T_H and T_L. The COP of the Carnot cycle has to be more than that of the VCRS because of both increased refrigerating effect [from $(h_1 - h_4)$ to $(h_1 - h_4^*)$] and decreased work input

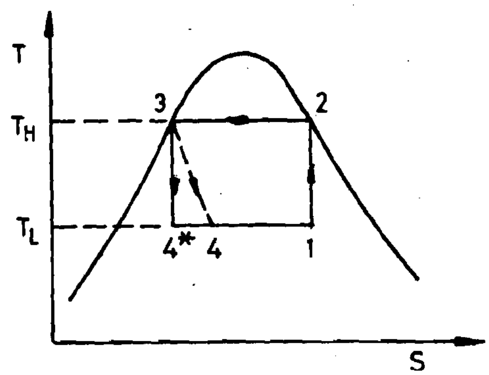

Figure 12.3 : Carnot refrigeration cycle

from $(h_2 - h_1)$ to $[(h_2 - h_1) - (h_3 - h_4^*)]$.

The Carnot cycle is a reversible engine and hence the ratio of heat transfers is equal to the ratio of the condensing and evaporating temepratures.

By its definition,

$$\text{COP} = \frac{Q_L}{Q_H - Q_L} \tag{12.2}$$

$$= \frac{1}{\frac{Q_H}{Q_L} - 1}$$

Replacing the ratio of heat transfer by the ratio of temperatures and rearranging,

$$\text{COP}_{\text{carnot}} = \text{COP}_{\max} = \frac{T_L}{(T_H - T_L)} \tag{12.11}$$

No refrigeration cycle operating between given T_H and T_L can have a COP greater than that given by equation (12.11).

Although Carnot refrigeration cycle can be thought of as a model of perfect refrigeration cycle, it is impracticable as the work obtained by expanding the high pressure liquid in the expander is so insignificant that it does not warrant the use of the expander at an additional cost and added problems of its operation and maintenance.

In actual practice, while a short capillary tube can do the job of expansion in small VCRS, a simple expansion valve does the same job in large VCRS. It is for this reason that attempts are not made to replace them by expanders.

SAQ 2

What is the maximum COP of a VCRS operating between an evaporator temperature of -3°C and a condenser temperature of 27°C.

SAQ 3

Why is reversed Carnot cycle not considered as the ideal cycle for a VCRS ?

12.2.4 Wet and Dry Compressions

The basic or ideal cycle for the VCRS shown in Figure 12.2 is called the 'Wet Compression Cycle' because the condition of vapour at inlet to the compressor is wet. Shown in Figure 12.4 is the basic cycle on $T-s$ and $p-h$ diagrams for the 'Dry Compression Cycle' Here the vapour enters the compressor as saturated vapour at the evaporator temperature. The dry compression is normally preferred mainly for the following two reasons: (i) The compressor efficiency is high and (ii) No damage to the compressor by the entering slugs of liquid.

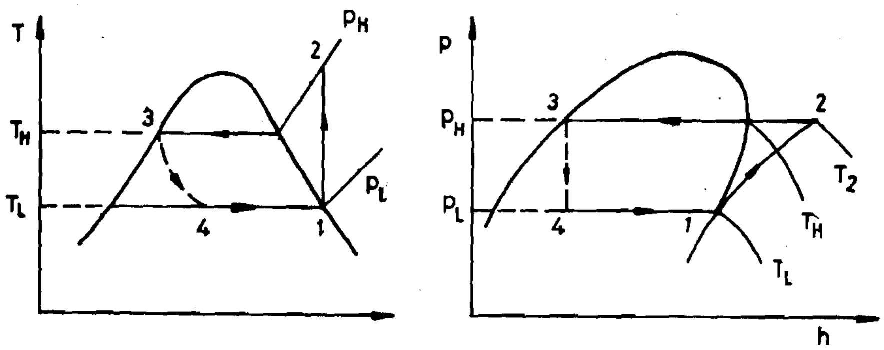

Figure 12.4 : Vapour compression refrigeration cycle (dry compression)

In the absence of any specific information, in the analysis of a simple VCRS, the system is assumed to work on basic or ideal dry compression cycle shown in Figure 12.4. In such a case the vapour at entry to the compressor is saturated vapour at the evaporator temperature and the compression of this vapour to state 2 is through reversible and adiabtic compression in the compressor. Also, at state 2 the pressure is the saturation pressure of the refrigerant corresponding to the condenser temperature.

In actual practice the cycles do not strictly follow the basic cycles because of irrversibilities. Also many modifications are made on the basic or ideal cycles to improve the system performance. Explanation of all these improved cycles is beyond the scope of this unit and hence the reader is advised to look into standard books on refrigeration for further details if required.

SAQ 4

Show on $T-s$ and $p-h$ diagrams the effect of irreversibilities on compressor work. Express adiabatic efficiency of the compressor in terms of enthalpies.

12.3 GAS COMPRESSION REFRIGERATION SYSTEM

Gas compression refrigeration system is mainly used in liquefaction of gases. In such a case the gas to be liquefied is compressed to high pressures through multistage compression, cooled in heat exchangers at constant pressure, throttled to very low pressures and the resulting liquid and vapour are separated. A particular type of this refrigeration system with air as the working substance, known as 'air cycle refrigeration system' is a popular system used in cooling aircraft cabins. Such a system works on reversed Brayton or Joule Cycle - Brayton Cycle being the ideal cycle on which a closed cycle gas turbine power plant works.

12.3.1 Brayton or Joule Cycle

The Brayton or the Joule Cycle, as already mentioned, is the ideal cycle on which a closed cycle gas turbine power plant operates. The operation of the plant is explained in Unit 6 with the help of the schematic diagram in Figure 6.2. The same is reproduced here in Figure 12.5. Figure 12.6 gives the cycle, on $T-s$ diagram, on which the plant operates. This

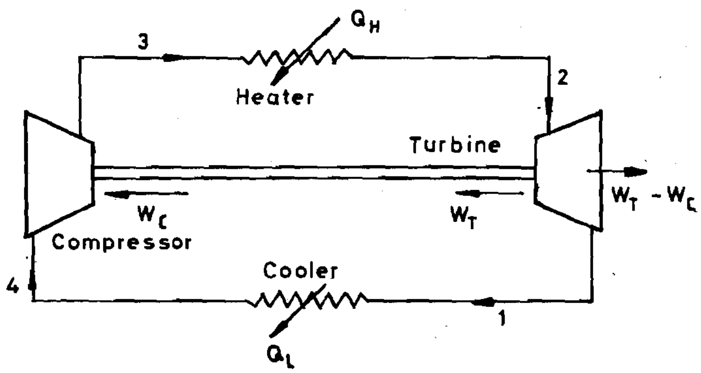

Figure 12.5 : Closed cycle gas turbine power plant

is the Brayton or Joule Cycle. The states of air marked in the schematic and cycle diagrams are compatible. The Brayton Cycle, as shown in Figure 12.6, consists of the following processes: 1 – 2, reversible adiabatic compression; 2 – 3, reversible constant pressure heating; 3 – 4, reversible adiabatic expansion and 4 – 1 reversible constant pressure cooling. The heat transfer to the system is Q_H and the net work output of the system is $(W_T - W_C)$.

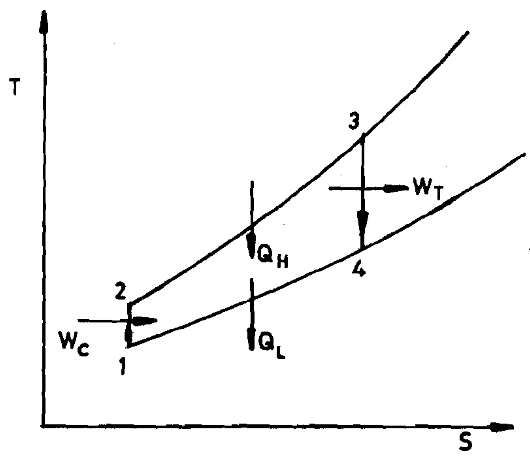

Figure 12.6 : Brayton cycle

12.3.2 Reversed Brayton Cycle

While Figure 12.7 shows the schematic of the air cycle refrigeration system, Figure 12.8 shows the cycle on which the system works. It is easy to comprehend from these Figures that they are exactly the same as those in Figures 12.5 and 12.6 but for the directions of heat and work interactions. It is for this reason that the ideal cycle on which air cycle refrigeration works is named as the 'reversed Brayton or Joule Cycle'.

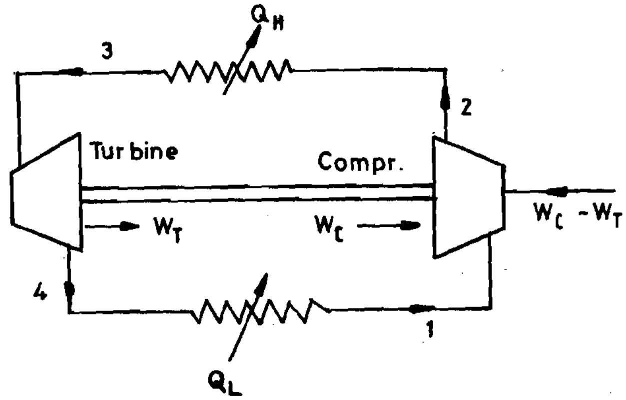

Figure 12.7 : Air cycle refrigeration system

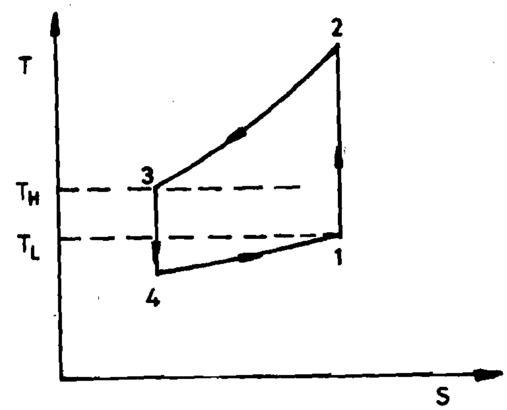

Figure 12.8 : Reversed Brayton cycle

Air leaving the refrigerated space at state 1 is compressed to state 2 reversibly and adiabatically. The compressed air is then cooled at constant pressure, p_h, to state 3 by passing the air through the cooler. The air at state 3 is then expanded reversibly and adiabatically in the turbine to state 4. The work output of the turbine, W_T, aids the compressor to the extent it can, thereby reducing the net work to be supplied from outside from W_C to $(W_C - W_T)$. Low pressure, low temperature air at state 4 enters the space to be refrigerated to pick up the heat to maintain the refrigerated space at a temperature corresponding to state 1.

Here, per kg air, the magnitudes of heat and work are,

$$Q_L = h_1 - h_4 = c_p(t_1 - t_4) \quad (12.12)$$

$$W_C = h_2 - h_1 = c_p(t_2 - t_1) \quad (12.13)$$

$$Q_H = h_2 - h_3 = c_p(t_2 - t_3) \quad (12.14)$$

and

$$W_T = h_3 - h_4 = c_p(t_3 - t_4). \quad (12.15)$$

Also,

$$\left(\frac{p_h}{p_l}\right) = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{T_3}{T_4}\right)^{\frac{\gamma}{\gamma-1}} \quad (12.16)$$

$$\text{COP} = \frac{Q_L}{(W_C - W_T)} = \frac{(T_1 - T_4)}{[(T_2 - T_1) - (T_3 - T_4)]} \quad (12.17)$$

$$\text{Refrigeration capacity} = \dot{m}\, c_p(T_1 - T_4) \quad (12.18)$$

$$\text{Compressor Power} = \dot{m}\, c_p(T_2 - T_1) \quad (12.19)$$

The above cycle is only an ideal cycle and hence the irreversibilities in the compressor and turbine are not considered. In actual practice, because of irreversibilities the temperatures at the end of compression and expansion increases, leading to the decrease in COP of the system.

SAQ 5

An airconditioning system operates on reversed Brayton cycle maintaining the cold room at 20°C while the surroundings are at 40°C. Show the cycle on $T-s$ diagram and identify the states corresponding to the above two temperatures.

SAQ 6

Air enters the compressor of an ideal air cycle refrigeration system at 1 bar, 20°C. The discharge pressure from the compressor is 3 bar. Find the temperature of air at the exit of the compressor.

12.3.3 Aircraft Cooling System

The COP of the air cycle refrigeration system is low. In spite of this it is used in air craft cooling as it has the advantage of lower weight compared to an equivalent capacity vapour refrigeration system and also beacause there is no need for an additional compressor for the cooling system as the required compressed air can be bled from the main compressor of the jet engine.

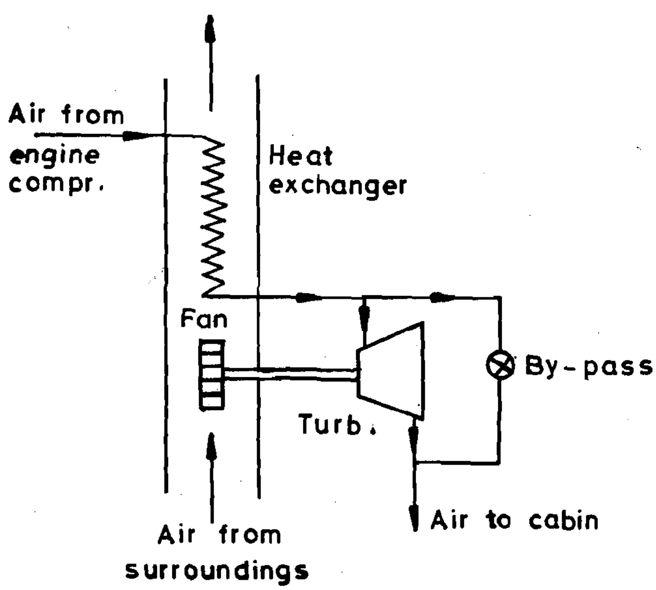

Figure 12.9 : Aircraft cooling system

Shown in Figure 12.9 is the schematic of a simple system used for cooling aircraft cabins. Air bled from the jet engine compressor is passed through the heat exchanger cooled by the outside air. This cooled high pressure air is then expanded in a small turbine. The work from the turbine can be used to run the fan that sucks the air over the heat exchanger. The air emerging out of the turbine at low temperature is passed through the cabin. The by-pass line is used for temperature control.

12.4 VAPOUR ABSORPTION REFRIGERATION SYSTEM

Shown in Figure 12.10 is the schematic of a simple vapour absorption refrigeration system (VARS). A careful perusal of the Figure reveals that the VARS system is essentially the same as the simple VCRS, except that the compressor in the latter is replaced by the components inside the dotted boundary in the schematic of the former. Hence, a VARS can be viewed as a VCRS in which the compressor is replaced by the absorber - generator assembly. In the VARS the work to be supplied for the liquid pump is insignificantly small compared to the work required by the compressor in a VCRS - a distinct advantage of the VARS. But, a large amount of heat required to be supplied in the generator leads to poor performance of the VARS. The VARS can hence be used advantageously and economically only in such cases where either waste heat is available as a by-product or low grade energy such as solar energy is used for providing cooling.

The working of the VARS is explained briefly with the help of the schematic diagram in Figure 12.10.

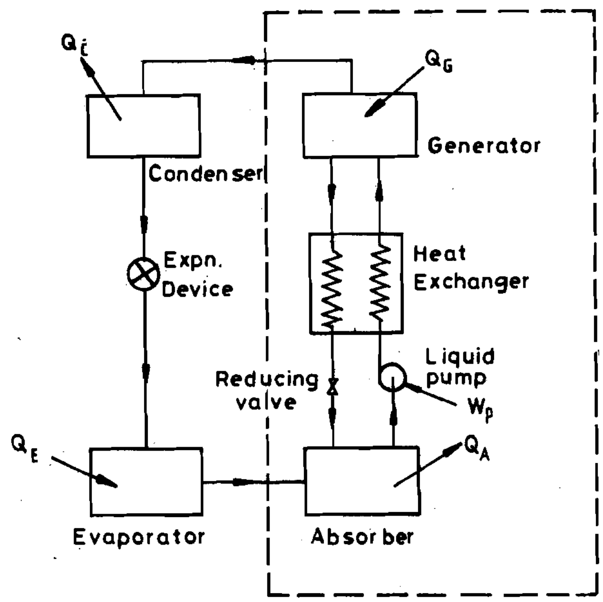

Figure 12.10 :Vapour absorption refrigeration system

The working substance in the system is a refrigerant - absorbent solution. The refrigerant and absorbent selected are such that the refrigerant is a low boiling point liquid and that it has a high affinity to the absorbent.

A strong refrigerant-absorbent solution (rich in the former) is heated in the generator. Let the heat transfer in the generator be Q_G. The refrigerant vapour leaving the generator at high pressure is cooled and condensed in the water cooled condenser. Let Q_C be the heat transferred from the refrigerant in the condenser. The liquid refrigerant from the condenser is throttled through the expansion device. Low quality refrigerant vapour entering the evaporator picks up heat (Q_E) from it to become saturated vapour and thereby produces the required cooling as in the VCRS. The low pressure, low temperature vapour leaving the evaporator enters the water cooled absorber where it meets the weak cool solution from the generator, the pressure difference between the generator and the absorber being maintained by the pressure reducing valve (The generator and condenser are on the high-pressure side of the system, while the evaporator and the absorber are on the low pressure side). The refrigerant vapour is absorbed by the weak solution and the resulting strong solution is then pumped to the generator through the liquid to liquid heat exchanger. This heat exchanger is an important component in the system. In the heat exchanger heat is transferred from the hot but weak solution coming from the generator to the strong solution going to the generator. Thus, the heat exchanger ensures that low temperature weak solution enters the absorber to improve its performance, and high temperature strong solution enters the generator to decrease the heat transfer in it, both contributing to the improvement in performance of the system.

The COP of the system is given by

$$\text{COP} = \frac{Q_E}{(W_p + Q_G)}.$$

The COP is low because although W_P is very small for a given Q_E, the Q_G required is very large.

In the simple system explained above it is assumed that pure refrigerant vapour leaves the generator to be condensed in the condenser. This is barely a fact in practice. The vapour leaving the generator is only rich in the refrigerant component. It also carries a small fraction of absorbent. This absorbent also gets condensed in the condenser and when it enters the throttle valve gives rise to several problems in the expansion device and the evaporator. Hence in the actual system provision must be made to rectify the vapour leaving the generator to assure that only refrigerant vapour leaves the generator and the absorbent vapours are separated, condesned and returned back to the generator. This is accomplished by installing a rectifying equipment (analyser and a rectifier) on the generator.

Among the various refrigerant-absorbent combinations used in VARS the most important commercial combination has been the Ammonia-water combination in which ammonia is the refrigerant. A VARS working with this combination is called the Aqua-ammonia absorption refrigeration system.

Another popular combination from the air-conditioning point is the water-Lithium Bromide combination in which water is the refrigerant.

12.4.1 Maximum COP of the VARS

If the absorption system were to operate under ideal conditions then its COP shall be the maximum. To imagine such an ideal situation all processes must be assumed to be reversible. The system operates continuously (in a cycle) with five interactions:

Heat transfer Q_G to the system at generator temperature T_G;

Heat transfer Q_E to the system at evaporator temperature T_E;

Heat transfer Q_C from the system at condenser temperature T_C;

Heat transfer Q_A from the system at absorber temperature T_A;

Work transfer W_P to the system at the pump.

As the condenser and the absorber are cooled by the water from the surroundngs, for ideal conditions, $T_C = T_A = T_o$ the ambient temperture, and the net heat transfer to the surroundings $= Q_o$ where

$$Q_o = Q_C + Q_A. \qquad \text{(i)}$$

Applying the first law of thermodynamics to the system

$$Q_o = Q_G + Q_E + W_P. \qquad \text{(ii)}$$

If the system were to be working reversibly the net entropy change of the system and surroundings would be zero. There can be no change in entropy of the working substance as it undergoes a thermodynamic cycle. Hence the net entropy change of the surroundings must be equal to zero.

i.e. $$\left(-\frac{Q_G}{T_G}\right) + \left(-\frac{Q_E}{T_E}\right) + \left(\frac{Q_o}{T_o}\right) = 0 \qquad \text{(iii)}$$

Substituting for Q_o in (iii) from (ii) and nelecting W_P as small,

$$\left(-\frac{Q_G}{T_G}\right) + \left(-\frac{Q_E}{T_E}\right) + \left(\frac{Q_G + Q_E}{T_o}\right) = 0 \qquad \text{(iv)}$$

Equation (iv) can be simplified and rearranged as

$$\frac{Q_E}{Q_G} = \left[\frac{T_G - T_o}{T_G}\right]\left[\frac{T_E}{T_o - T_E}\right] \qquad \text{(v)}$$

In equation (v) while the term on the left hand side is the COP of the system, the first term on the right side is the Carnot efficiency of a heat engine operating between the Generator and the ambient temperature and the second term is the COP of a Carnot refrigeration system operating between the evaporator and the ambient temperatures. Thus, the maximum COP of an absorption system is equal to the product of the Carnot efficiency of a heat engine operating between T_G and T_o and the COP of a Carnot refrigerator operating between T_E and T_c.

SAQ 7

What is the maximum COP of an absorption refrigeration system, if it has to maintain an evaporator at − 3°C, while having heat transfer from a high temperature reservoir at 127°C at a place where the ambient temperature is 27°C ?

12.5 STEAM JET REFRIGERATION SYSTEM

This system, also called a flash cooling system, is generally used for chilling water. The system turns out to be an economical proposition, in comparison to VCRS, in such situations where steam, to be used in the ejector of the system, is available cheaply in large quantities.

Figure 12.11 shows the schematic of a steam jet refrigeration system.

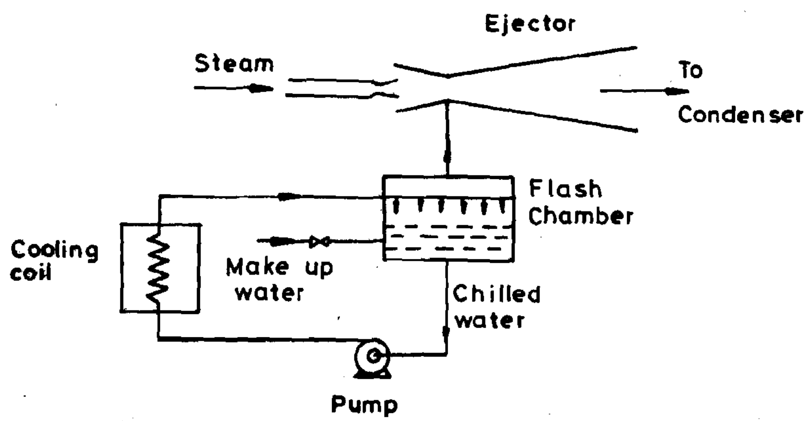

Figure 12.11 : Steam jet refrigeration system

Water to be chilled is sprayed in the flash chamber. This chamber is well insulated and maintained at low pressure with the help of the steam ejector which continuously removes the water vapour flashing in the chamber. Because the chamber is insulated, the energy required for vapourising a part of spray water has to be supplied by the non-vaporising liquid part. Hence the temperature of liquid water in the flash chamber decreases to provide the required chilled water. The chilled water is pumped out to an external heat exchanger, through the cooling coils, where it cools the air to be conditioned. The water from the cooling coils is again passed to the flash chamber. As water vapour is being continuously removed from the flash chamber there is a need for supplying make up water to the system.

The steam from the ejector is condensed in a large condenser which is cooled by water from the local source. The condenser is maintained at the required low pressure, normally sub-atmospheric, by wet and dry pumps.

12.6 ILLUSTRATIVE PROBLEMS

Example 12.1 :

An ideal wet compression refrigeration cycle, with R – 12 as the refrigerant, operates between an evaporator temperature of - 10°C and a condenser temperature of 40°C. Calculate the following:

(a) refrigerating effect,

(b) compressor work, and

(c) COP.

Solution :

The system is operating on the cycle in Figure 12.2, where $T_H = 40°C$ and $T_L = -10°C$. The required properties of R – 12 can be read from Table 12.1.

$h_2 = h_g$ at 40°C = 203.05 kJ/kg

$h_3 = h_f$ at 40°C = 74.53 kJ/kg

$h_4 = h_3 = 74.53$ kJ/kg

To find the value of h_1:

$s_2 = s_g$ at 40°C = 0.6820 = s_1.

But $s_1 = [(1 - x_1)\, s_f + x_1\, s_g]$ at – 10°C

$\therefore\ 0.6820 = (1 - x_1) \times 0.1079 + x_1 \times 0.7014$ or $x_1 = 0.967$.

Now $h_1 = [(1 - x_1)\, h_f + x_1\, h_g]$ at – 10°C.

$h_1 = 0.033 \times 26.85 + 0.967 \times 183.06 = 177.9$ kJ/kg.

(a) Ref.effect = $h_1 - h_4 = 177.9 - 74.53 = 103.37$ kJ/kg.

(b) Comp.Work = $h_2 - h_1 = 203.05 - 177.9 = 25.15$ kJ/kg.

(c) $\text{COP} = \dfrac{\text{ref. effect}}{\text{comp. work}} = \dfrac{103.37}{25.15} = 4.11$

Example 12.2 :

(a) What is the maximum COP for a refrigeration system operating between - 10°C and 40°C ? (b) If the system in 12.2 (a) is operating with R-12 as the refrigerant find the refrigerating effect and the compressor work. (c) Compare the results of (a) and (b) with those of Prob.12.1.

Solution:

(a) For maximum COP the system must be operating in a reversible cycle, i.e. on a reversed Carnot cycle. In such a case,

$$\text{COP} = \frac{T_L}{T_H - T_L} \qquad (12.11)$$

$$= \frac{273 - 10}{40 + 10} = 5.26$$

(b) The cycle can be assumed to be the same as 1-2-3-4*-1 in Figure 12.3.

Also
$$\text{COP} = 5.26 = \frac{(h_1 - h_4^*)}{[(h_2 - h_1) - (h_3 - h_4^*)]}, \qquad (12.8)$$

where, h_1, h_2 and h_3 have the same values as in Prob.12.1.

Substituting these in the above relation,

$$5.26 = \frac{(177.9 - h_4^*)}{[(203.5 - 177.9) - (74.53 - h_4^*)]},$$

$$\therefore \ h_4^* = 69.53 \text{ kJ/kg}.$$

$$\text{Ref.effect} = h_1 - h_4^* = 177.9 - 69.53 = 108.37 \text{ kJ/kg}.$$

$$\text{Net work} = \frac{\text{Ref.effect}}{\text{COP}} = \frac{108.37}{5.26} = 20.6 \text{ kJ/kg}.$$

(c) The results of Prob. 12.1 and 12.2 are presented in the following Table:

	Prob.12.1	Prob.12.2
COP	4.11	5.26
Ref.effect (kJ/kg)	103.37	108.37
Work Input (kJ/kg)	25.15	20.6

The above Table reveals that while the ref. effect and COP in Prob.12.2 are greater than those in Prob.12.1, the work input is smaller. The difference between the two is because while the cycle in Prob.12.2 is reversible the cycle in Prob.12.1 is not.

Example 12.3 :

Re-do Problem 12.1 assuming dry compression, and that the enthalpy after compression is 209.4 kJ/kg.

Solution :

The $T-s$ diagram of Figure 12.4 may be used for the solution of this problem. Reading the required properties from the property Table:

$$h_1 = h_g \text{ at } -10°\text{C} = 183.06 \text{ kJ/kg}$$
$$h_2 = 209.4 \text{ kJ/kg (from data)}$$
$$h_3 = h_f \text{ at } 40°\text{C} = 74.53 \text{ kJ/kg}$$
$$h_4 = h_3 = 74.53 \text{ kJ/kg}$$

(a) Ref.effect $= h_1 - h_4 = 183.06 - 74.53 = 108.53$ kJ/kg.

(b) Comp. work $= h_2 - h_1 = 209.4 - 183.06 = 26.34$ kJ/kg.

(c) $$\text{COP} = \frac{\text{Ref. effect}}{\text{Comp. work}} = \frac{108.53}{26.34} = 4.12$$

Example 12.4 :

Re-do problem 12.3 assuming compressor efficiency is 0.8.

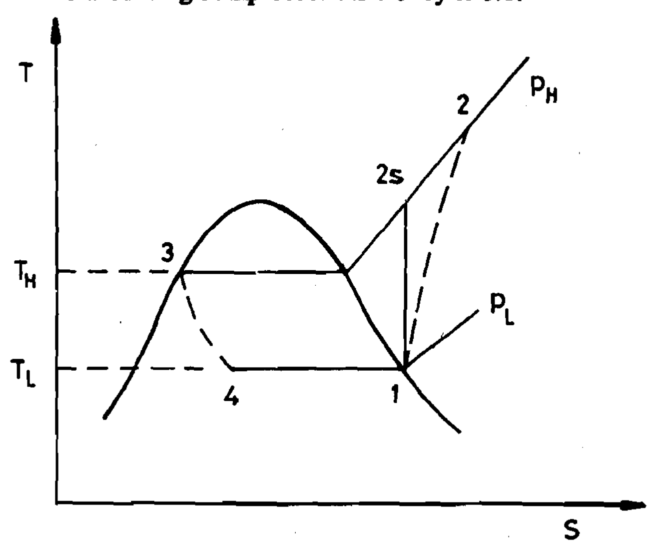

Figure 12.12 : Figure for ill.prob.12.4.

Solution:

Here the compression process is not reversible as its efficiency is not 100 % , but is given to be 80 %. Hence, more work is required by the compressor during this irreversible adiabatic compression process. The efficiency of a compressor is defined as the ratio of the reversible adiabatic work of compression to the actual work, the compression occurring in both the cases from the same initial state to the same discharge pressure.

In Figure 12.12 while $1-2s$ is the reversible adiabatic compression process, 1 - 2 is the actual compression process.

Thus,

$$\frac{W_{\text{rev.ad.}}}{W_{\text{actual}}} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}$$

State 2s is the same as state 2 in Prob.12.3. Therefore, $W_{\text{rev.ad.}} = h_{2s} - h_1 = W_{\text{prob.12.3}} = 26.34$ kJ/kg, and the values of h_1 (from Tables or from Prob.12.3) = 183.06 kJ/kg.

$$0.8 = \frac{26.34}{W_{\text{act}}}, \text{ or } W_{\text{act}} = 32.955 \text{ kJ/kg.}$$

Ref.effect = $h_1 - h_4$ = same as in Prob.12.3 = 108.53 kJ/kg.

$$\text{COP} = \frac{\text{Ref.effect}}{\text{Comp.Work}} = \frac{108.53}{32.955} = 3.3$$

(This problem illustrates that the irrversibility in compression leads to increased work of compression, thereby decreasing the COP of the system)

Example 12.5 :

An R -12 refrigeration plant of capacity 17.5 kW maintains a cold store at -20°C. The condenser cooling water is available in plenty at 30°C. The enthalpy of refrigerant vapour at the end of isentropic compression is 206 kJ/kg. Find (a) COP, (b) mass flow rate of the refrigerant, (c) compressor power and (d) compressor cylinder dimensions assuming stroke = 1.2 diameter, speed = 900 rev/min and volumetric efficiency = 0.92.

Solution :

Figure 12.4 can be assumed to represent the cycle on which the system operates. Properties at various states can be found using the Table of properties.

$$h_1 = 178.61 \text{ kJ/kg}$$

$$h_2 = 206 \text{ kJ/kg (from data)}$$

$$h_3 = 64.54 \text{ kJ/kg} = h_4$$

$$v_1 = 0.10885 \text{ m}^3\text{/kg}$$

Ref.effect = $h_1 - h_4$ = 114.07 kJ/kg.

Comp.Work = $h_2 - h_1$ = 27.39 kJ/kg.

(a) $\text{COP} = \frac{\text{Ref.effect}}{\text{Comp.work}} = \frac{114.07}{27.39} = 4.16.$

(b) $\text{Mass flow rate} = \frac{\text{capacity}}{\text{Ref.effect}}$

$$= \frac{17.5}{114.07} = 0.153 \text{ kg/ s}$$

(c) Comp.power = mass flow rate × comp.work/kg

$$= 0.153 \times 27.39 = 4.19 \text{ kW.}$$

(d) Speed of compr. = 900 rev/min = 15 rev/s.

Specific volume of refrigerant vapour at entry to compr.= v_1 = 0.10885 m³/kg.

Actual volume of vapour handled per second = mass flow rate × v_1

$$= 0.153 \times 0.10885 = 0.01665 \text{ m}^3/\text{s}$$

As the compressor runs 15 times/second, the actual volume of vapour handled by the compr. per cycle = 0.01665/15 m^3.

$$\text{The piston displacement volume} = \left(\frac{\pi}{4}\right) d^2 \times 1.2d,$$

where d is the diameter of the cylinder and 1.2 d is the stroke as per data provided.

$$\text{Volumetric eff.} = \frac{\text{actual volume handled}}{\text{p.d.volume}} = 0.92$$

$$\therefore \quad 0.92 = \frac{\dfrac{0.01665}{15}}{\dfrac{\pi}{4} d^2 \times 1.2d} \quad \text{or} \quad d = 0.1085 \text{ m.}$$

Diameter of the compressor cylinder = 10.85 cm.

Stroke length = $1.2d = 1.2 \times 11.18 = 13.02$ cm.

Example 12.6 :

In an ideal air cycle refrigeration system, air enters the compressor at 1 bar, 5°C and is compressed to 3 bar. The air is then cooled at constant pressure to 50°C and then expanded in a turbine to 1 bar. The cooling capacity of the system is 10 kW. Assume air behaves as a perfect gas with $c_p = 1.005$ kJ/kg K and $c_v = 0.718$ kJ/kg K.

Find COP, mass flow rate of air and the power required by the system.

Solution :

Figure 12.8 represents the cycle on which the system operates.

From data, $T_1 = 273 + 5 = 278$ K.

$p_L = 1$ bar, $p_h = 3$ bar.

$T_3 = 273 + 50 = 323$ K.

Capacity = 10 kW and

$$\gamma = \frac{c_p}{c_v} = \frac{1.005}{0.718} = 1.4.$$

As process 1-2 is reversible and adiabatic,

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{p_h}{p_L}\right)^{\frac{\gamma-1}{\gamma}} \qquad (12.16)$$

$$\text{or} \qquad T_2 = 278\left(\frac{3}{1}\right)^{\frac{0.4}{1.4}} = 380.5 \text{ K.}$$

Similarly for the reversible adiabatic expansion process 3-4,

$$\left(\frac{T_3}{T_4}\right) = \left(\frac{p_h}{p_L}\right)^{\frac{\gamma-1}{\gamma}} \qquad (12.16)$$

$$\text{i.e.} \qquad T_4 = \frac{323}{(3)^{0.4/1.4}} = 236 \text{ K.}$$

$$\text{Ref.effect} = c_p(T_1 - T_4) \qquad (12.12)$$

$$= 1.005(278 - 236) = 42.21 \text{ kJ/kg.}$$

$$W = W_C - W_T$$

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

$$= 1.005[(380.5 - 278) - (323 - 236)] = 15.6 \text{ kJ/kg}.$$

$$\text{COP} = \frac{\text{Ref.effect}}{\text{Work}} = \frac{42.21}{15.6} = 2.7.$$

$$\text{Mass flow rate} = \frac{\text{Capacity}}{\text{ref.eff.}} = \frac{10}{42.21} = 0.237 \text{ kg/s}.$$

$$\text{Power required} = \text{mass flow rate} \times \text{work/kg}$$
$$= 0.237 \times 15.6 = 3.7 \text{ kW}.$$

Example 12.7 :

Re-do Problem 12.6 assuming the efficiencies of compressor and turbine to be 0.7 and 0.75 respectively.

Solution :

The cycle is shown in Figure 12.13. Here while 1-2s-3-4s-1 is the ideal cycle, 1-2-3-4-1 is the actual cycle. Also states 2s and 4s are the same as states 2 and 4 in Prob.12.6.

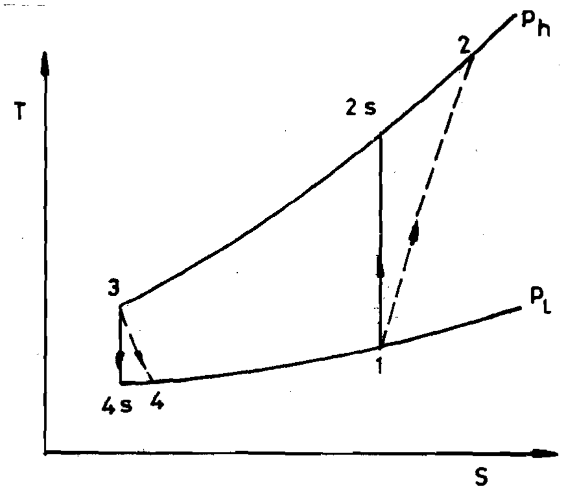

Figure 12.13 : Figure for ill.prob.12.7

Hence, $T_1 = 278$ K

$T_{2s} = 380.5$ K

$T_3 = 303$ K

and $T_{4s} = 236$ K.

$$\text{Comp.eff} = \frac{W_{1-2s}}{W_{1-2}} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$$\therefore\ 0.7 = \frac{(380.5 - 278)}{(T_2 - 278)} \text{ or } T_2 = 424.4 \text{ K}.$$

$$\text{Similarly Turb.eff} = \frac{W_{3-4}}{W_{3-4s}} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}.$$

$$\therefore\ 0.75 = \frac{(323 - T_4)}{(323 - 236)}, \text{ or } T_4 = 258 \text{ K}.$$

$$\text{COP} = \frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)}$$

$$= \frac{(278 - 258)}{(424.4 - 278) - (323 - 258)} = 0.245$$

$$\text{Mass flow rate} = \frac{\text{capacity}}{\text{ref.effect}} = \frac{10}{c_p(T_1 - T_4)}$$

$$= \frac{10}{1.005(278-258)} = 0.497 \text{ kg/s.}$$

Power required = mass flow rate × work/kg

$$= 0.497 \times c_p[(T_2 - T_1) - (T_3 - T_4)]$$

$$= 0.497 \times 1.005[(424.4 - 278) - (323 - 258)]$$

$$= 40.65 \text{ kW.}$$

12.7 SUMMARY

Most commonly used refrigearting system is the VCRS.

Dry compression is preferred to wet compression.

Refrigerating effect is the cooling produced per kg refrigerant.

COP of a simple VCRS is equal to the ratio of refrigerating effect and the compressor work per kg.

Capacity of a refrigerating system is the rate at which cooling is produced and is given by the product of refrigerating effect and the mass flow rate of the refrigerant.

A system is said to be of 1 ton capacity if it cools at a rate of 3.5 kW.

Irreversibilities in a VCRS decrease the COP.

Unless and otherwise mentioned an ideal vapour compression refrigeration cycle is assumed to operate on dry compression with saturated vapour entering the compressor.

The COP of a VARS is low.

VARS can be used advantageously wherever thermal energy is available in plenty with negligible cost.

The maximum COP of a VARS, maintaining an evaporator at T_E at a place where surroundings are at T_o and having heat interaction with a high temperature reservoir at T_G is equal to the product of the Carnot efficiency of a heat engine operating between T_G and T_o and the COP of a reversed Carnot refrigerator operating between T_E and T_o.

While R – 12 is a very commonly used refrigerant in domestic refrigerators, R – 22 is used in window airconditioners.

Most VARS operate with ammonia as the refrigerant and water as the absorbent.

When VARS is used for airconditioning normally water is used as the refrigerant and lithium bromide is used as the absorbent.

12.8 GLOSSARY

Refrigeration	:	Process of maintaining temperature less than the local surroundings temperature.
Refrigerator	:	A device that provides refrigeration.
Refrigerant	:	Working substance in a refrigerator.
VCRS	:	Vapour Compression Refrigeration System.
VARS	:	Vapour Absorption Refrigeration System.
Dry Compression	:	Vapour at entry to compressor of a VCRS is saturated.
Wet Compression	:	Vapour at entry to compressor of a VCRS is wet.
Refrigerating Effect	:	Cooling produced per kg refrigerant.
Capacity of refrigerating System	:	Rate at which cooling is produced. Also equal to the product of refrigerating effect and mass flow rate of the refrigerant.
COP	:	Coefficient of performance of a refrigeration system.

12.9 ANSWERS /SOLUTIONS TO SAQs

SAQ 1

0.2917 kg/s, 60 kJ/kg.

SAQ 2

9.

SAQ 3

Reversed Carnot cycle is not used as the reference cycle as the turbine in the system is not practical. Also the work output from it is insignificant and does not warrant its use.

SAQ 4

In the $T-s$ and $p-h$ digrams shown below, in Figure 12.14, state 1 is the state of the refrigerant at entry to the compressor irrespective of whether the compression is reversible or not. But, the end state after compression is 2 if the compression is irreversible and adiabatic and 2s if the compression is reversible and adiabatic. The adiabatic effiiciency of the compressor is given by

$$= \frac{W_{1-2s}}{W_{1-2}} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}.$$

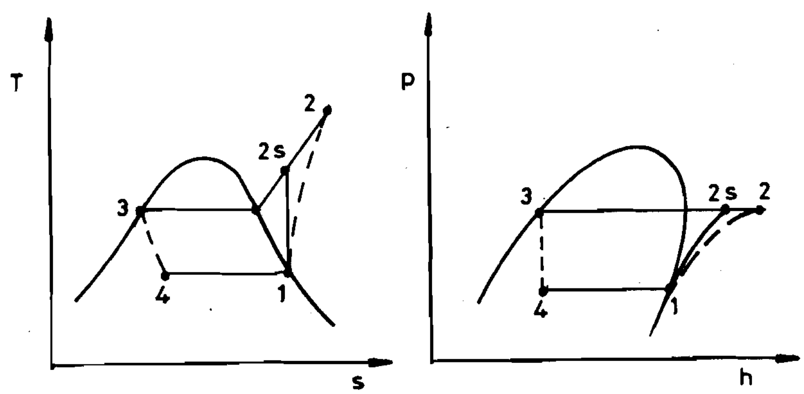

Figure 12.14 : Figure for SAQ 4

SAQ 5

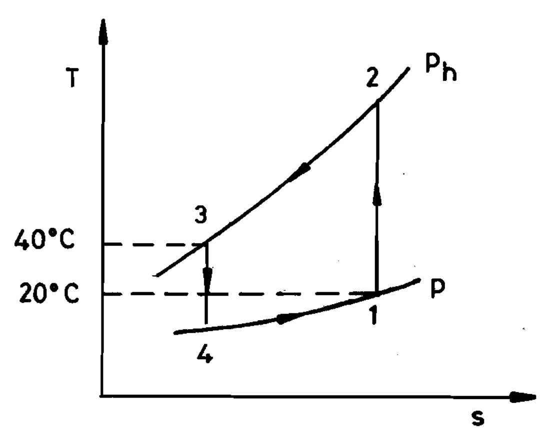

Figure12.15 : Figure for SAQ 5

SAQ 6

128°C.

SAQ 7

2.25.

Table 12.1 : R -12 Liquid - Vapour Saturation, Pressure Table

Temp. °C	Abs.Pr. MPa	v_f m³/kg	v_g	h_f kJ/kg	h_g	s_f kJ/kg K	s_g
-50	0.0391	0.00065	0.38311	-8.77	164.84	-0.0384	0.7396
-40	0.0642	0.00066	0.24191	-0.00	169.48	-0.0000	0.7269
-30	0.1004	0.00067	0.15938	8.85	174.08	0.0371	0.7165
-20	0.1509	0.00069	0.10885	17.80	178.61	0.0730	0.7082
-10	0.2191	0.00070	0.07665	26.85	183.06	0.1079	0.7014
0	0.3086	0.00072	0.05539	36.02	187.40	0.1418	0.6960
10	0.4233	0.00073	0.04091	45.34	191.60	0.1750	0.6916
20	0.5673	0.00075	0.03078	54.83	195.64	0.2076	0.6879
30	0.7449	0.00077	0.02351	64.54	199.48	0.2397	0.6848
40	0.9607	0.00080	0.01817	74.53	203.05	0.2716	0.6820
50	1.2193	0.00083	0.01417	84.87	206.30	0.3034	0.6792

TABLE 12.2 : Ammonia Liquid - Vapour Saturation, Pressure Table

Temp. °C	Abs.Pr. kPa	v_f m³/kg	v_g	h_f kJ/kg	h_g	s_f kJ/kg K	s_g
-50	40.88	0.00142	2.6254	-44.3	1372.4	-0.1942	6.1561
-40	71.77	0.00145	1.5521	0.0	1389.0	0.0000	5.9589
-30	119.55	0.00148	0.9635	44.7	1404.6	0.1873	5.7815
-20	190.22	0.00150	0.6237	89.7	1419.0	0.3684	5.6205
-10	290.85	0.00153	0.4185	135.2	1432.0	0.5440	5.4730
0	429.44	0.00157	0.2895	181.1	1443.5	0.7145	5.3369
10	614.95	0.00160	0.2056	227.6	1453.3	0.8808	5.2104
20	857.12	0.00164	0.1494	274.9	1461.5	1.0434	5.0920
30	1166.49	0.00168	0.1106	322.9	1467.9	1.2028	4.9805
40	1554.33	0.00173	0.0833	371.7	1472.2	1.3591	4.8740
50	2032.62	0.00178	0.0635	421.7	1473.7	1.5135	4.7696