Refrigeration and Air Conditioning

Introduction

Air cycle refrigeration systems belong to the general class of gas cycle refrigeration systems, in which a gas is used as the working fluid. The gas does not undergo any phase change during the cycle, consequently, all the internal heat transfer processes are sensible heat transfer processes. Gas cycle refrigeration systems find applications in air craft cabin cooling and also in the liquefaction of various gases. In the present chapter gas cycle refrigeration systems based on air are discussed.

Air Standard Cycle analysis

Air cycle refrigeration system analysis is considerably simplified if one makes the following assumptions:

i. The working fluid is a fixed mass of air that behaves as an ideal gas
ii. The cycle is assumed to be a closed loop cycle with all inlet and exhaust processes of open loop cycles being replaced by heat transfer processes to or from the environment
iii. All the processes within the cycle are reversible, i.e., the cycle is internally reversible
iv. The specific heat of air remains constant throughout the cycle

An analysis with the above assumptions is called as cold Air Standard Cycle (ASC) analysis. This analysis yields reasonably accurate results for most of the cycles and processes encountered in air cycle refrigeration systems. However, the analysis fails when one considers a cycle consisting of a throttling process, as the temperature drop during throttling is zero for an ideal gas, whereas the actual cycles depend exclusively on the real gas behavior to produce refrigeration during throttling.

Basic concepts

The temperature of an ideal gas can be reduced either by making the gas to do work in an isentropic process or by sensible heat exchange with a cooler environment. When the gas does adiabatic work in a closed system by say, expanding against a piston, its internal energy drops. Since the internal energy of the ideal gas depends only on its temperature, the temperature of the gas also drops during the process, i.e.,

$$W = m(u_1 - u_2) = mc_v(T_1 - T_2) \quad (1)$$

where m is the mass of the gas, u_1 and u_2 are the initial and final internal energies of the gas, T_1 and T_2 are the initial and final temperatures and c_v is the specific heat at constant volume. If the expansion is reversible and adiabatic, by using the ideal gas equation and the equation for isentropic process the final temperature

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad (2)$$

is related to the initial temperature (T_1) and initial and final pressures (P_1 and P_2) by the equation:

$$\gamma = \left(\frac{c_p}{c_v}\right) \quad (3)$$

where γ is the coefficient of isentropic expansion given by:
Isentropic expansion of the gas can also be carried out in a steady flow in a turbine which gives a net work output. Neglecting potential and kinetic energy changes, the work output of the turbine is given by:

$$W = m(h_1 - h_2) = mc_p(T_1 - T_2) \quad (4)$$

The final temperature is related to the initial temperature and initial and final pressures by Eq. (2)

Reversed Carnot cycle employing a gas

Reversed Carnot cycle is an ideal refrigeration cycle for constant temperature external heat source and heat sinks. Figure 9.1(a) shows the schematic of a reversed Carnot refrigeration system using a gas as the working fluid along with the cycle diagram on T-s and P-Vcoordinates. As shown, the cycle consists of the following four processes:

Process 1-2: Reversible, adiabatic compression in a compressor
Process 2-3: Reversible, isothermal heat rejection in a compressor
Process 3-4: Reversible, adiabatic expansion in a turbine
Process 4-1: Reversible, isothermal heat absorption in a turbine

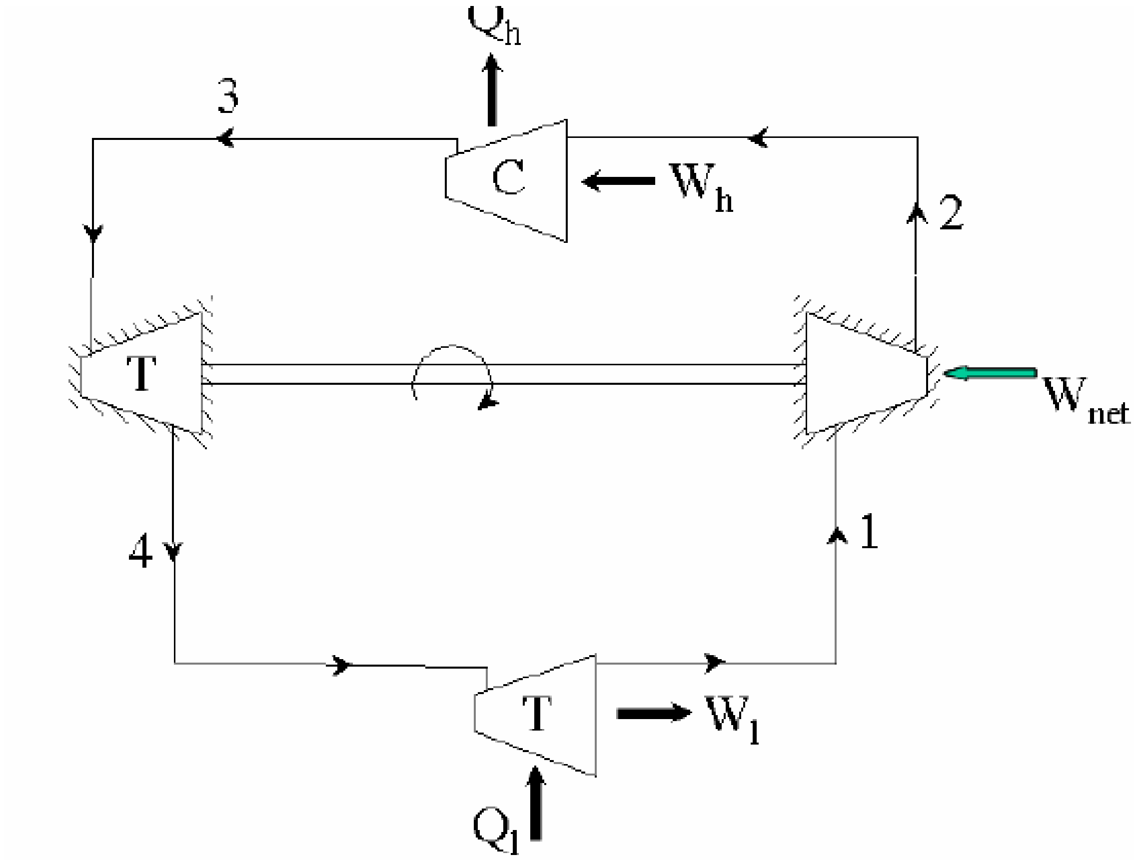


Fig.1(a). *Schematic of a reverse Carnot refrigeration system*

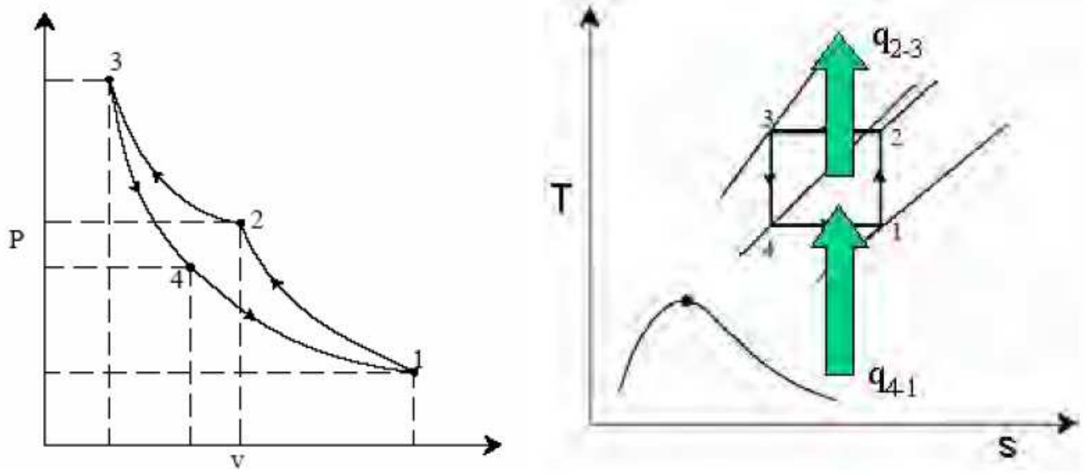
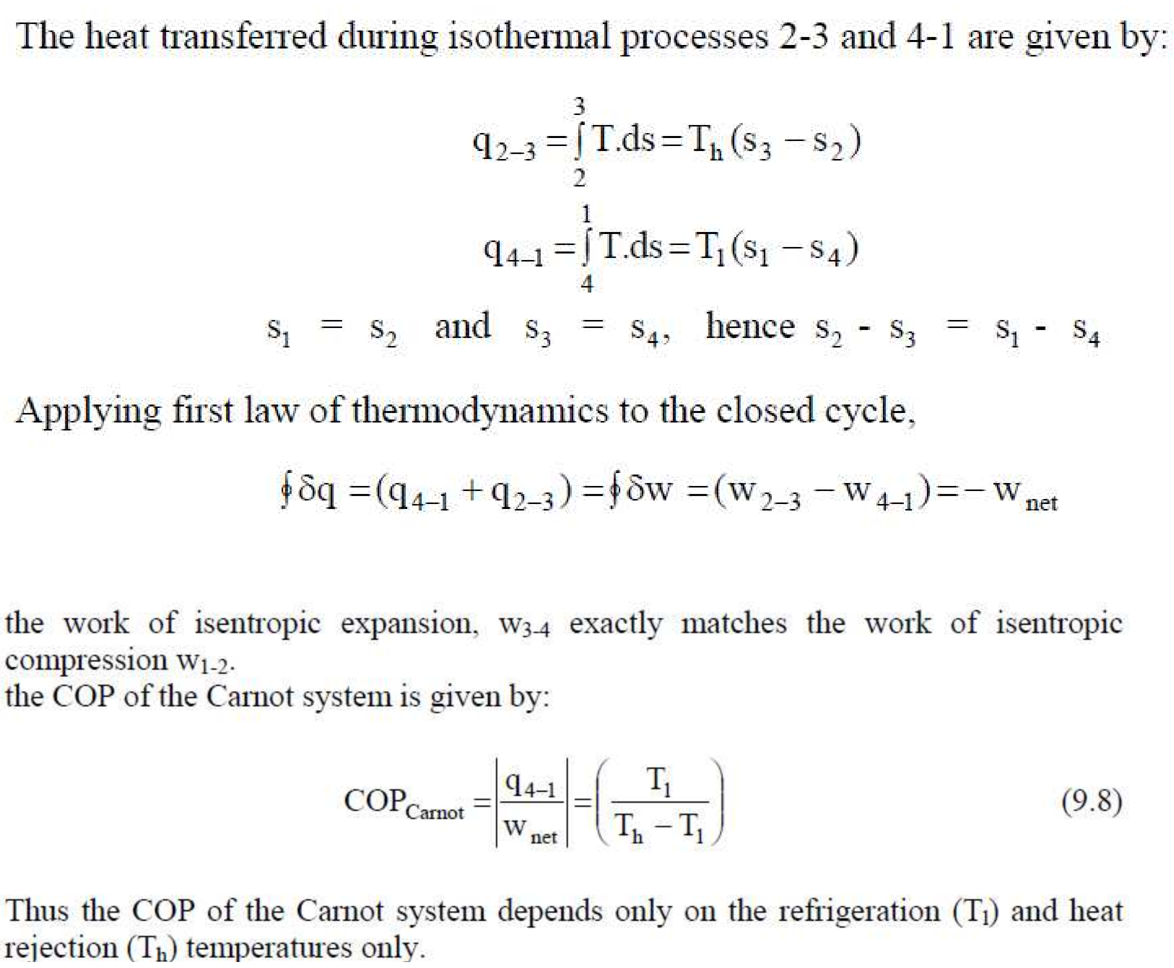


Fig. 1(b). *Reverse Carnot refrigeration system in P-v and T-s coordinates*

The heat transferred during isothermal processes 2-3 and 4-1 are given by:



The heat transferred during isothermal processes 2-3 and 4-1 are given by:

$$q_{2-3} = \int_2^3 T.ds = T_h(s_3 - s_2)$$

$$q_{4-1} = \int_4^1 T.ds = T_l(s_1 - s_4)$$

$s_1 = s_2$ and $s_3 = s_4$, hence $s_2 - s_3 = s_1 - s_4$

Applying first law of thermodynamics to the closed cycle,

$$\oint \delta q = (q_{4-1} + q_{2-3}) = \oint \delta w = (w_{2-3} - w_{4-1}) = -w_{net}$$

the work of isentropic expansion, w_{3-4} exactly matches the work of isentropic compression w_{1-2}.
the COP of the Carnot system is given by:

$$COP_{Carnot} = \left|\frac{q_{4-1}}{w_{net}}\right| = \left(\frac{T_l}{T_h - T_l}\right) \qquad (9.8)$$

Thus the COP of the Carnot system depends only on the refrigeration (T_l) and heat rejection (T_h) temperatures only.

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Limitations of Carnot cycle:

Carnot cycle is an idealization and it suffers from several practical limitations. One of the main difficulties with Carnot cycle employing a gas is the difficulty of achieving isothermal heat transfer during processes 2-3 and 4-1. For a gas to have heat transfer isothermally, it is essential to carry out work transfer from or to the system when heat is transferred to the system (process 4-1) or from the system (process 2-3). This is difficult to achieve in practice. In addition, the volumetric refrigeration capacity of the Carnot system is very small leading to large compressor displacement, which gives rise to large frictional effects. All actual processes are irreversible, hence completely reversible cycles are idealizations only.

Ideal reverse Brayton cycle

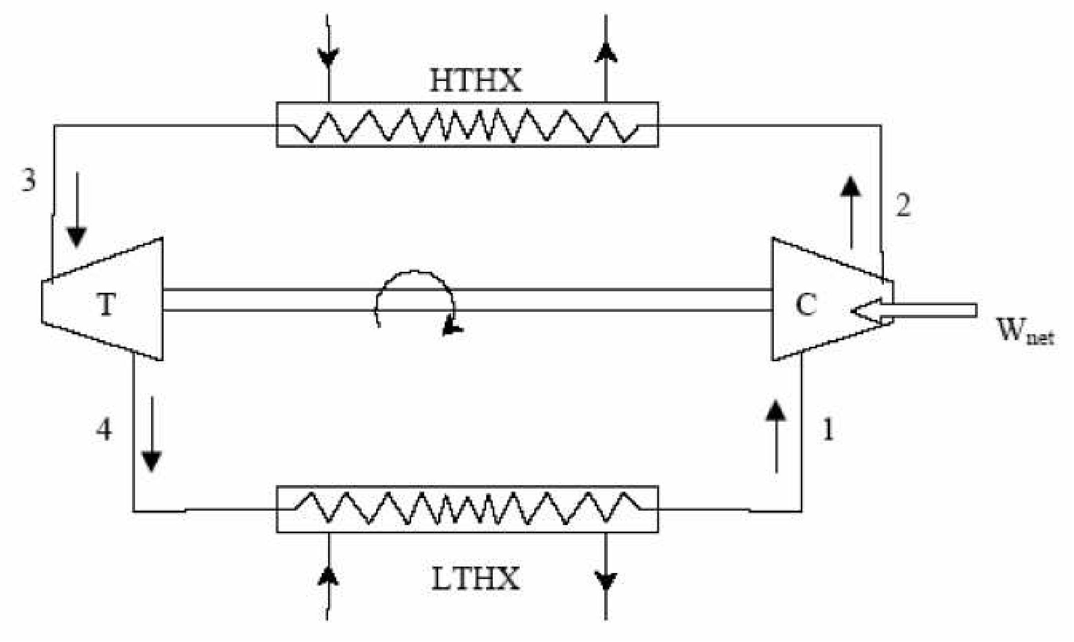

Fig. 2(a). *Schematic of a closed reverse Brayton cycle*

This is an important cycle frequently employed in gas cycle refrigeration systems. This may be thought of as a modification of reversed Carnot cycle, as the two isothermal processes of Carnot cycle are replaced by two isobaric heat transfer processes. This cycle is also called as Joule or Bell-Coleman cycle. Figure 2(a) and (b) shows the schematic of a closed, reverse Brayton cycle and also the cycle on T-s diagram. As shown in the figure, the ideal cycle consists of the following four processes:

Process 1-2: Reversible, adiabatic compression in a compressor
Process 2-3: Reversible, isobaric heat rejection in a heat exchanger
Process 3-4: Reversible, adiabatic expansion in a turbine
Process 4-1: Reversible, isobaric heat absorption in a heat exchanger

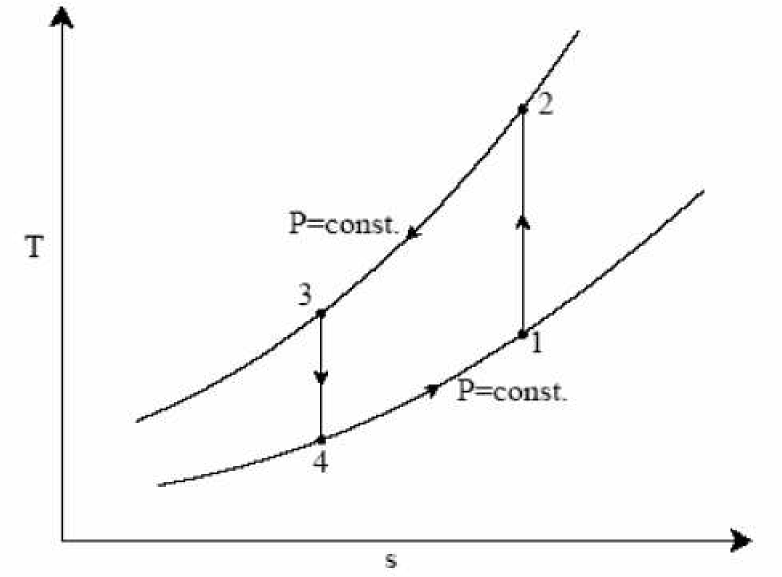

Fig. 2(b). *Reverse Brayton cycle in T-s plane*

Process 3-4: High pressure gas from the heat exchanger flows through a turbine, undergoes isentropic expansion and delivers net work output. The temperature of the gas drops during the process from T_3 to T_4. From steady flow energy equation:

$$\begin{gathered} W_{3-4} = \dot{m}(h_3 - h_4) = \dot{m} c_p (T_3 - T_4) \\ s_3 = s_4 \\ \text{and } T_3 = T_4 \left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}} = T_4 \, r_p^{\frac{\gamma-1}{\gamma}} \end{gathered} \tag{9.11}$$

where $r_p = (P_3/P_4)$ = pressure ratio

Process 4-1: Cold and low pressure gas from turbine flows through the low temperature heat exchanger and extracts heat sensibly and isobarically from a heat source, providing a useful refrigeration effect. The enthalpy and temperature of the gas rise during the process due to heat exchange, no work transfer takes place and the entropy of the gas increases. Again applying steady flow energy equation and second T ds equation:

$$\begin{gathered} Q_{4-1} = \dot{m}(h_1 - h_4) = \dot{m} c_p (T_1 - T_4) \\ s_4 - s_1 = c_p \ln \frac{T_4}{T_1} \\ P_4 = P_1 \end{gathered} \tag{9.12}$$

From the above equations, it can be easily shown that:

$$\left(\frac{T_2}{T_1} \right) = \left(\frac{T_3}{T_4} \right) \tag{9.13}$$

Applying 1st law of thermodynamics to the entire cycle:

$$\oint \delta q = (q_{4-1} - q_{2-3}) = \oint \delta w = (w_{3-4} - w_{1-2}) = -w_{net} \tag{9.14}$$

The COP of the reverse Brayton cycle is given by:

$$COP = \left| \frac{q_{4-1}}{w_{net}} \right| = \left(\frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \right) \tag{9.15}$$

using the relation between temperatures and pressures, the COP can also be written as:

$$COP = \left(\frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \right) = \left(\frac{T_4}{T_3 - T_4} \right) = \left(\frac{(T_1 - T_4)}{(T_1 - T_4)(r_p^{\frac{\gamma-1}{\gamma}} - 1)} \right) = (r_p^{\frac{\gamma-1}{\gamma}} - 1)^{-1} \tag{9.16}$$

From the above expression for COP, the following observations can be made:

a) For fixed heat rejection temperature (T_3) and fixed refrigeration temperature (T_1), the COP of reverse Brayton cycle is always lower than the COP of reverse Carnot cycle (Fig. 9.3), that is

$$COP_{Brayton} = \left(\frac{T_4}{T_3 - T_4}\right) < COP_{Carnot} = \left(\frac{T_1}{T_3 - T_1}\right)$$

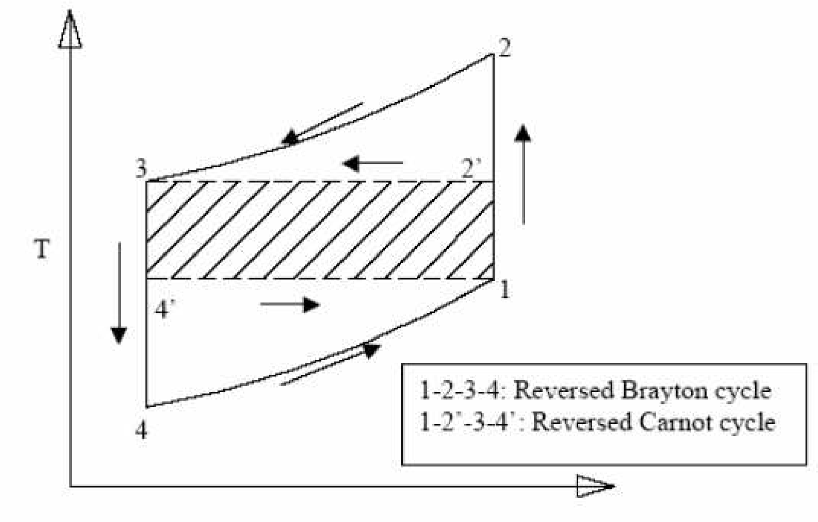

***Fig. 9.3**. Comparison of reverse Carnot and reverse Brayton cycle in T-s plane*

b) COP of Brayton cycle approaches COP of Carnot cycle as T_1 approaches T_4 (thin cycle), however, the specific refrigeration effect [$c_p(T_1-T_4)$] also reduces simultaneously.

c) COP of reverse Brayton cycle decreases as the pressure ratio r_p increases

Actual reverse Brayton cycle:

The actual reverse Brayton cycle differs from the ideal cycle due to:

i. Non-isentropic compression and expansion processes
ii. Pressure drops in cold and hot heat exchangers

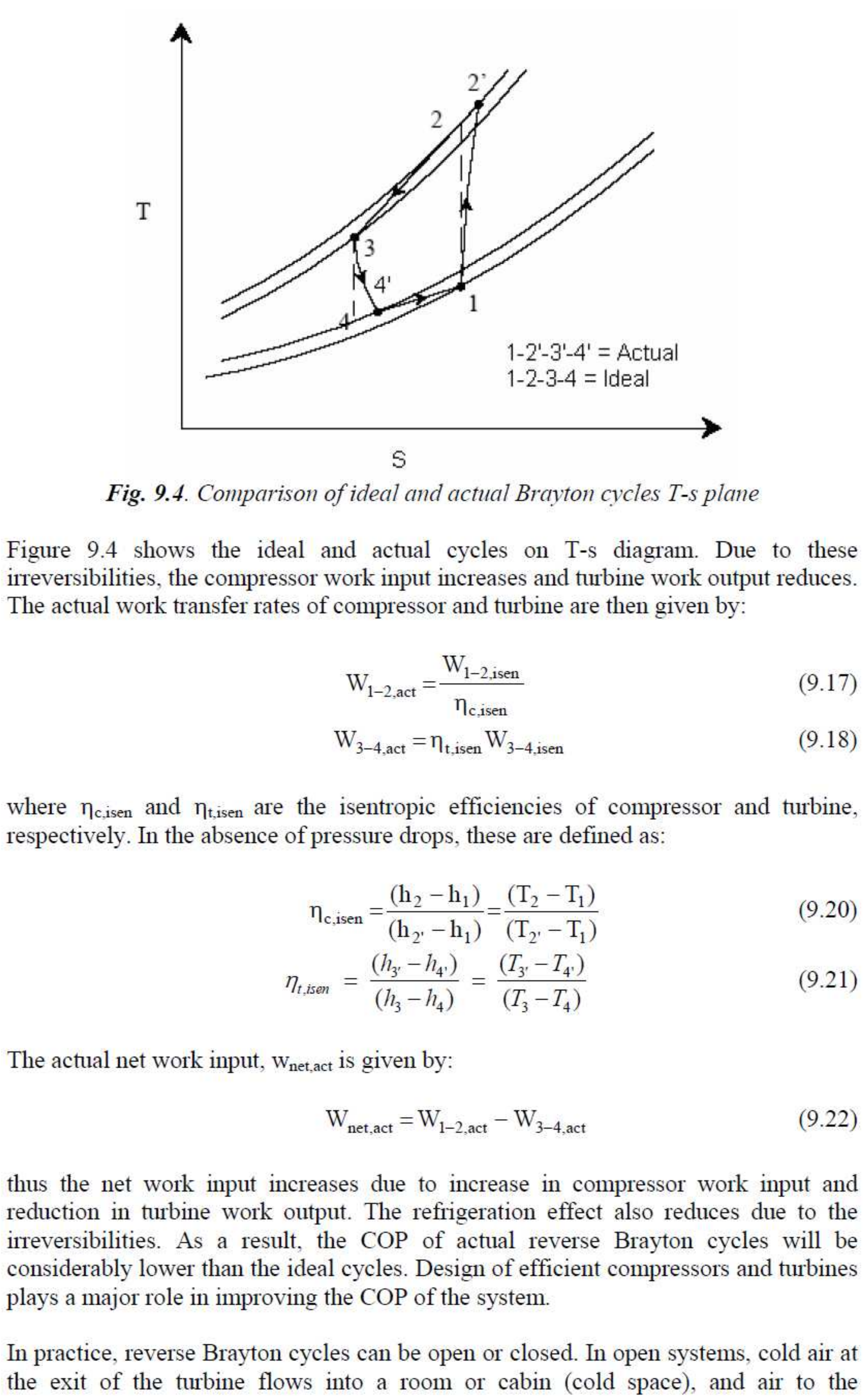

***Fig. 9.4**. Comparison of ideal and actual Brayton cycles T-s plane*

Figure 9.4 shows the ideal and actual cycles on T-s diagram. Due to these irreversibilities, the compressor work input increases and turbine work output reduces. The actual work transfer rates of compressor and turbine are then given by:

$$W_{1-2,act} = \frac{W_{1-2,isen}}{\eta_{c,isen}} \qquad (9.17)$$

$$W_{3-4,act} = \eta_{t,isen} W_{3-4,isen} \qquad (9.18)$$

where $\eta_{c,isen}$ and $\eta_{t,isen}$ are the isentropic efficiencies of compressor and turbine, respectively. In the absence of pressure drops, these are defined as:

$$\eta_{c,isen} = \frac{(h_2 - h_1)}{(h_{2'} - h_1)} = \frac{(T_2 - T_1)}{(T_{2'} - T_1)} \qquad (9.20)$$

$$\eta_{t,isen} = \frac{(h_{3'} - h_{4'})}{(h_3 - h_4)} = \frac{(T_{3'} - T_{4'})}{(T_3 - T_4)} \qquad (9.21)$$

The actual net work input, $w_{net,act}$ is given by:

$$W_{net,act} = W_{1-2,act} - W_{3-4,act} \qquad (9.22)$$

thus the net work input increases due to increase in compressor work input and reduction in turbine work output. The refrigeration effect also reduces due to the irreversibilities. As a result, the COP of actual reverse Brayton cycles will be considerably lower than the ideal cycles. Design of efficient compressors and turbines plays a major role in improving the COP of the system.

In practice, reverse Brayton cycles can be open or closed. In open systems, cold air at the exit of the turbine flows into a room or cabin (cold space), and air to the

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In practice, reverse Brayton cycles can be open or closed. In open systems, cold air at the exit of the turbine flows into a room or cabin (cold space), and air to the compressor is taken from the cold space. In such a case, the low side pressure will be atmospheric. In closed systems, the same gas (air) flows through the cycle in a closed manner. In such cases it is possible to have low side pressures greater than atmospheric. These systems are known as *dense air systems*. Dense air systems are advantageous as it is possible to reduce the volume of air handled by the compressor and turbine at high pressures. Efficiency will also be high due to smaller pressure ratios. It is also possible to use gases other than air (e.g. helium) in closed systems.

Aircraft cooling systems

In an aircraft, cooling systems are required to keep the cabin temperatures at a comfortable level. Even though the outside temperatures are very low at high altitudes, still cooling of cabin is required due to:

i. Large internal heat generation due to occupants, equipment etc.
ii. Heat generation due to skin friction caused by the fast moving aircraft
iii. At high altitudes, the outside pressure will be sub-atmospheric. When air at this low pressure is compressed and supplied to the cabin at pressures close to atmospheric, the temperature increases significantly. For example, when outside air at a pressure of 0.2 bar and temperature of 223 K (at 10000 m altitude) is compressed to 1 bar, its temperature increases to about 353 K. If the cabin is maintained at 0.8 bar, the temperature will be about 332 K. This effect is called as ram effect. This effect adds heat to the cabin, which needs to be taken out by the cooling system.
iv. Solar radiation

For low speed aircraft flying at low altitudes, cooling system may not be required, however, for high speed aircraft flying at high altitudes, a cooling system is a must.

Even though the COP of air cycle refrigeration is very low compared to vapour compression refrigeration systems, it is still found to be most suitable for aircraft refrigeration systems as:

i. Air is cheap, safe, non-toxic and non-flammable. Leakage of air is not a problem
ii. Cold air can directly be used for cooling thus eliminating the low temperature heat exchanger (open systems) leading to lower weight
iii. The aircraft engine already consists of a high speed turbo-compressor, hence separate compressor for cooling system is not required. This reduces the weight per kW cooling considerably. Typically, less than 50% of an equivalent vapour compression system
iv. Design of the complete system is much simpler due to low pressures. Maintenance required is also less.

Simple aircraft refrigeration cycle:

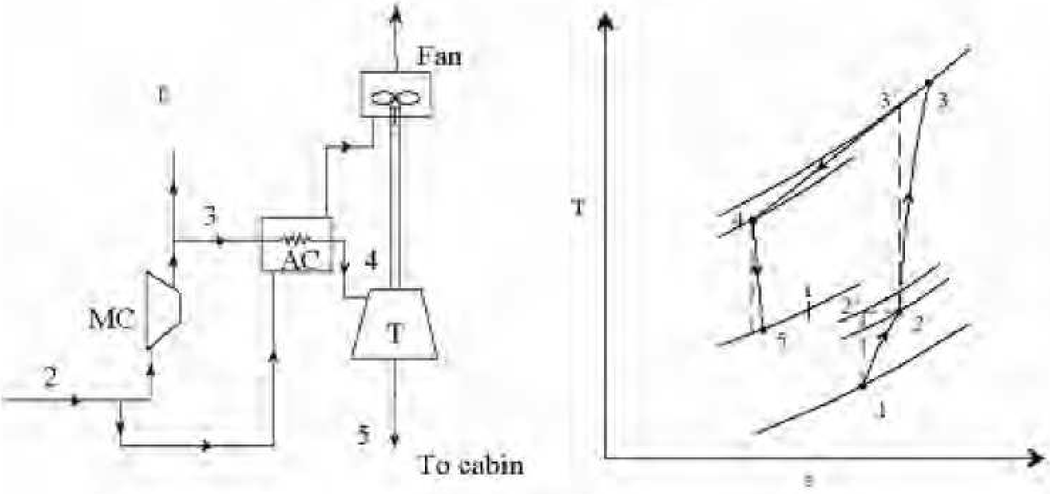


***Fig. 9.5**. Schematic of a simple aircraft refrigeration cycle*

Figure 9.5 shows the schematic of a simple aircraft refrigeration system and the operating cycle on T-s diagram. This is an open system. As shown in the T-s diagram, the outside low pressure and low temperature air (state 1) is compressed due to ram effect to ram pressure (state 2). During this process its temperature increases from 1 to 2. This air is compressed in the main compressor to state 3, and is cooled to state 4 in the air cooler. Its pressure is reduced to cabin pressure in the turbine (state 5), as a result its temperature drops from 4 to 5. The cold air at state 5 is supplied to the cabin. It picks up heat as it flows through the cabin providing useful cooling effect. The power output of the turbine is used to drive the fan, which maintains the required air flow over the air cooler. This simple system is good for ground cooling (when the aircraft is not moving) as fan can continue to maintain airflow over the air cooler.
By applying steady flow energy equation to the ramming process, the temperature rise at the end of the ram effect can be shown to be:

By applying steady flow energy equation to the ramming process, the temperature rise at the end of the ram effect can be shown to be:

$$\frac{T_{2'}}{T_1} = 1 + \frac{\gamma - 1}{2} M^2 \tag{9.23}$$

where M is the Mach number, which is the ratio of velocity of the aircraft (C) to the sonic velocity a ($a = \sqrt{\gamma R T_1}$), i.e.,

$$M = \frac{C}{a} = \frac{C}{\sqrt{\gamma R T_1}} \tag{9.24}$$

Due to irreversibilities, the actual pressure at the end of ramming will be less than the pressure resulting from isentropic compression. The ratio of actual pressure rise to the isentropic pressure rise is called as ram efficiency, η_{Ram}, i.e.,

$$\eta_{Ram} = \frac{(P_2 - P_1)}{(P_{2'} - P_1)} \tag{9.25}$$

The refrigeration capacity of the simple aircraft cycle discussed, $\dot{Q}$ is given by:

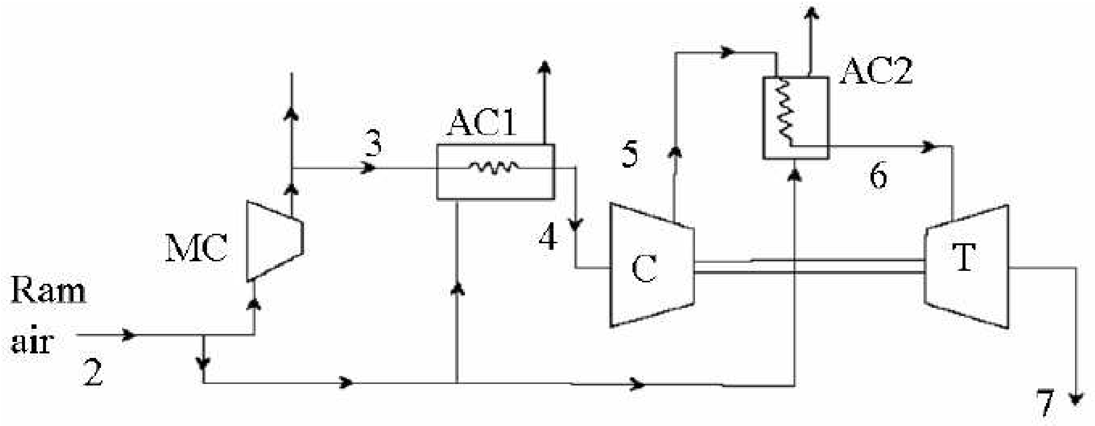
$$\dot{Q} = \dot{m} c_p (T_i - T_5) \tag{9.26}$$

where $\dot{m}$ is the mass flow rate of air through the turbine.

Bootstrap system:

Figure 9.6 shows the schematic of a bootstrap system, which is a modification of the simple system. As shown in the figure, this system consists of two heat exchangers (air cooler and aftercooler), instead of one air cooler of the simple system. It also incorporates a secondary compressor, which is driven by the turbine of the cooling system. This system is suitable for high speed aircraft, where in the velocity of the aircraft provides the necessary airflow for the heat exchangers, as a result a separate fan is not required. As shown in the cycle diagram, ambient air state 1 is pressurized to state 2 due to the ram effect. This air is further compressed to state 3 in the main compressor. The air is then cooled to state 4 in the air cooler. The heat rejected in the air cooler is absorbed by the ram air at state 2. The air from the air cooler is further compressed from state 4 to state 5 in the secondary compressor. It is then cooled to state 6 in the after cooler, expanded to cabin pressure in the cooling turbine and is supplied to the cabin at a low temperature T_7. Since the system does not consist of a separate fan for driving the air through the heat exchangers, it is not suitable for ground cooling. However, in general ground cooling is normally done by an external air conditioning system as it is not efficient to run the aircraft engine just to provide cooling when it is grounded.

Other modifications over the simple system are: regenerative system and reduced ambient system. In a regenerative system, a part of the cold air from the cooling turbine is used for precooling the air entering the turbine. As a result much lower temperatures are obtained at the exit of the cooling turbine, however, this is at the expense of additional weight and design complexity. The cooling turbine drives a fan similar to the simple system. The regenerative system is good for both ground cooling as well as high speed aircrafts. The reduced ambient system is well-suited for supersonic aircrafts and rockets.



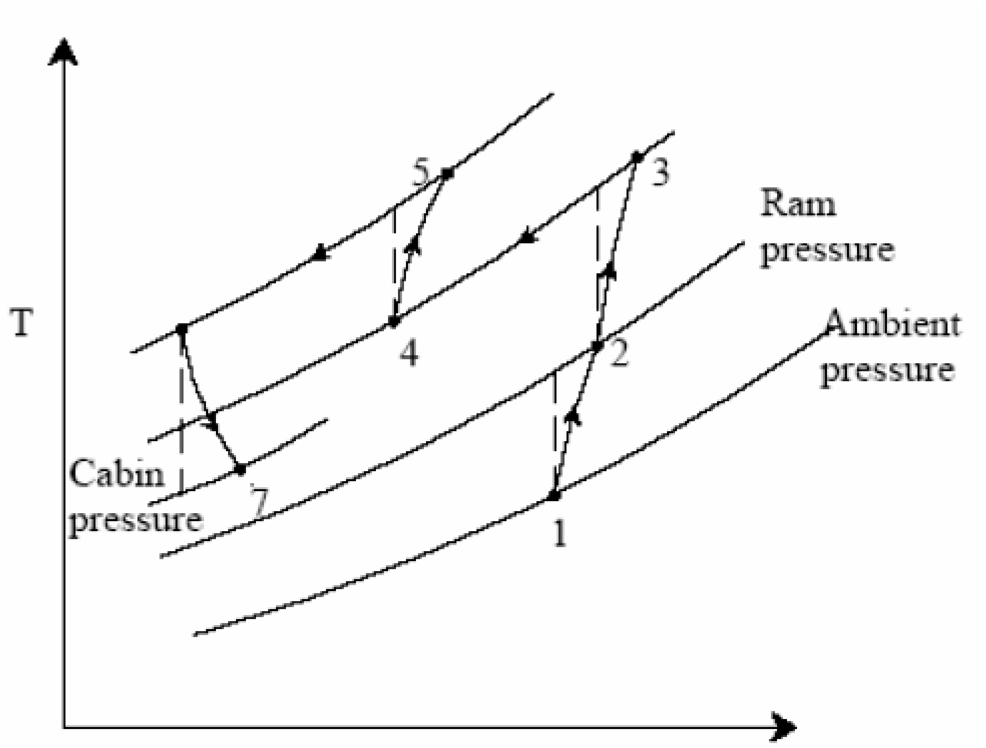


Fig. 9.6. *Schematic of a bootstrap system*

Dry Air Rated Temperature (DART):

The concept of Dry Air Rated Temperature is used to compare different aircraft refrigeration cycles. Dry Air Rated Temperature is defined as the temperature of the air at the exit of the cooling turbine in the absence of moisture condensation. For condensation not to occur during expansion in turbine, the dew point temperature and hence moisture content of the air should be very low, i.e., the air should be very dry. The aircraft refrigeration systems are rated based on the mass flow rate of air at the design DART. The cooling capacity is then given by:

$$\dot{Q} = \dot{m} c_p (T_i - T_{DART}) \tag{9.27}$$

is the mass flow rate of air, T_{DART} and T_i are the dry air rated temperature and cabin temperature, respectively.

A comparison between different aircraft refrigeration systems based on DART at different Mach numbers shows that:

i. DART increases monotonically with Mach number for all the systems except the reduced ambient system

ii. The simple system is adequate at low Mach numbers

iii. At high Mach numbers either bootstrap system or regenerative system should be used

iv. Reduced ambient temperature system is best suited for very high Mach number, supersonic aircrafts

Vapour Compression Refrigeration Systems

Introduction

Refrigeration may be defined as the process of achieving and maintaining a temperature below that of the surroundings, the aim being to cool some product or space to the required temperature. One of the most important applications of refrigeration has been the preservation of perishable food products by storing them at low temperatures. Refrigeration systems are also used extensively for providing thermal comfort to human beings by means of air conditioning. Air Conditioning refers to the treatment of air so as to simultaneously control its temperature, moisture content, cleanliness, odour and circulation, as required by occupants, a process, or products in the space. The subject of refrigeration and air conditioning has evolved out of human need for food and comfort, and its history dates back to centuries. The history of refrigeration is very interesting since every aspect of it, the availability of refrigerants, the prime movers and the developments in compressors and the methods of refrigeration all are a part of it.

Natural Refrigeration

In olden days refrigeration was achieved by natural means such as the use of ice or evaporative cooling. In earlier times, ice was either:

1. Transported from colder regions,
2. Harvested in winter and stored in ice houses for summer use or,
3. Made during night by cooling of water by radiation to stratosphere.

In Europe, America and Iran a number of icehouses were built to store ice. Materials like sawdust or wood shavings were used as insulating materials in these icehouses. Later on, cork was used as insulating material. Literature reveals that ice has always been available to aristocracy who could afford it. In India, the Mogul emperors were very fond of ice during the harsh summer in Delhi and Agra, and it appears that the ice used to be made by nocturnal cooling.

In 1806, Frederic Tudor, (who was later called as the “ice king”) began the trade in ice by cutting it from the Hudson River and ponds of Massachusetts and exporting it to various countries including India. In India Tudor’s ice was cheaper than the locally manufactured ice by nocturnal cooling. The ice trade in North America was a flourishing business. Ice was transported to southern states of America in train compartments insulated by 0.3m of cork insulation. Trading in ice was also popular in several other countries such as Great Britain, Russia, Canada, Norway and France. In these countries ice was either transported from colder

Comparison between gas cycles and vapour cycles

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. As mentioned in the previous chapter, in a typical gas cycle, the working fluid (a gas) does not undergo phase change; consequently the operating cycle will be away from the vapour dome. In gas cycles, heat rejection and refrigeration take place as the gas undergoes sensible cooling and heating. In a vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour absorption systems, vapour jet systems etc. Among these the vapour compression refrigeration systems are predominant.

Vapour Compression Refrigeration Systems

As mentioned, vapour compression refrigeration systems are the most commonly used among all refrigeration systems. As the name implies, these systems belong to the general class of vapour cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapour compression refrigeration system, refrigeration is obtained as the refrigerant evaporates at low temperatures. The input to the system is in the form of mechanical energy required to run the compressor. Hence these systems are also called as mechanical refrigeration systems. Vapour compression refrigeration systems are available to suit almost all applications with the refrigeration capacities ranging from few Watts to few megawatts. A wide variety of refrigerants can be used in these systems to suit different applications, capacities etc. The actual vapour compression cycle is based on Evans-Perkins cycle, which is also called as reverse Rankine cycle. Before the actual cycle is discussed and analysed, it is essential to find the upper limit of performance of vapour compression cycles. This limit is set by a completely reversible cycle.

The Carnot refrigeration cycle

Carnot refrigeration cycle is a completely reversible cycle, hence is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink. It is used as reference against which the real cycles are compared. Figures 10.1 (a) and (b) show the schematic of a Carnot vapour compression refrigeration system and the operating cycle on T-s diagram.

The basic Carnot refrigeration system for pure vapour consists of four components: compressor, condenser, turbine and evaporator. Refrigeration effect ($q_{4-1} = q_e$) is obtained at the evaporator as the refrigerant undergoes the process of vaporization (process 4-1) and extracts the latent heat from the low temperature heat source. The low temperature, low pressure vapour is then compressed isentropically in the compressor to the heat sink temperature T_c. The refrigerant pressure increases from P_e to P_c during the compression process (process 1-2) and the exit vapour is saturated. Next the high pressure, high temperature saturated refrigerant undergoes the process of condensation in the condenser (process 2-3) as it rejects the heat of condensation ($q_{2-3} = q_c$) to an external heat sink at T_c. The high pressure saturated liquid then flows through the turbine and undergoes isentropic expansion (process 3-4). During this process, the pressure and temperature fall from P_c, T_c to P_e, T_e. Since a saturated liquid is expanded in the turbine, some amount of liquid flashes into vapour and the exit condition lies in the two-phase region. This low temperature and low pressure liquid-vapour mixture then enters the evaporator completing the cycle. Thus as shown in Fig.10.1(b), the cycle involves two isothermal heat transfer processes (processes 4-1 and 2-3) and two isentropic work transfer processes (processes 1-2 and 3-4). Heat is extracted isothermally at evaporator temperature T_e during process 4-1, heat is rejected isothermally at condenser temperature T_c during process 2-3. Work is supplied to the compressor during the isentropic compression (1-2) of *refrigerant vapour* from evaporator pressure P_e to condenser pressure P_c, and work is produced by the system as *refrigerant liquid* expands isentropically in the turbine from condenser pressure P_c to evaporator pressure P_e. All the processes are both internally as well as externally reversible, i.e., net entropy generation for the system and environment is zero.

Applying first and second laws of thermodynamics to the Carnot refrigeration cycle,

$$\oint \delta q = \oint \delta w$$

$$\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$$

$$\oint \delta w = w_{3-4} - w_{1-2} = w_T - w_C = - w_{net}$$

$$\Rightarrow (q_c - q_e) = w_{net}$$

now for the reversible, isothermal heat transfer processes 2-3 and 4-1, we can write:

$$q_c\ q_{2-3} = -T.ds\ T_c (s_2 - s_3)$$

$$q_e\ q_{4-1} = -T.ds\ T_e (s_1 - s_4)$$

where Te and Tc are the evaporator and condenser temperatures, respectively, and,

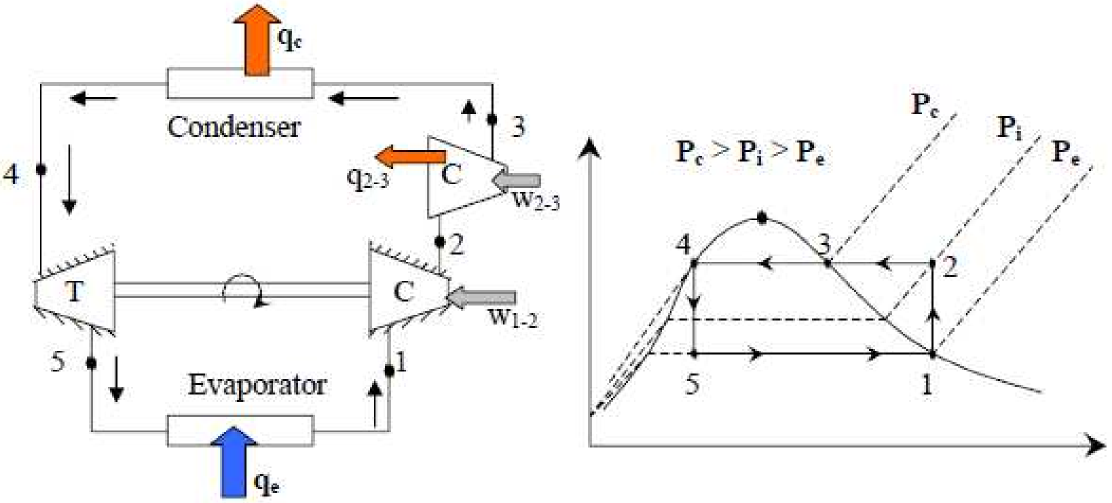
$$s_1 = s_2 \text{ and } s_3 = s_4$$

The Coefficient of Performance (COP) is given by:

$$COP_{carnot} = \text{Refrigerating Effect / Net work input} = q_e/W_{net}$$

$$= \frac{T_e(S_1 - S_4)}{T_c(S_2 - S_3) - T_e(S_1 - S_4)} = \frac{T_e}{T_c - T_e}$$

Thus the COP of Carnot refrigeration cycle is a function of evaporator and condenser temperatures only and is independent of the nature of the working substance. This is the reason why exactly the same expression was obtained for air cycle refrigeration systems operating on Carnot cycle .The Carnot COP sets an upper limit for refrigeration systems operating between two constant temperature thermal reservoirs (heat source and sink). From Carnot's theorems, for the same heat source and sink temperatures, no irreversible cycle can have COP higher than that of Carnot COP.



Carnot refrigeration system with dry compression

It can be seen from the above expression that the COP of a Carnot refrigeration system increases as the evaporator temperature increases and condenser temperature decreases. This can be explained very easily with the help of the T-s diagram (Fig.10.2). As shown in the figure, COP is the ratio of area a-1-4-b to the area 1-2-3-4. For a fixed condenser temperature T_c, as the evaporator temperature T_e increases, area a-1-4-b (q_e) increases and area 1-2-3-4 (w_{net}) decreases as a result, COP increases rapidly. Similarly for a fixed evaporator temperature T_e, as the condensing temperature T_c increases, the net work input (area 1-2-3-4) increases, even though cooling output remains constant, as a result the COP falls. Figure 10.3 shows the variation of Carnot COP with evaporator temperature for different condenser temperatures. It can be seen that the COP increases sharply with evaporator temperatures, particularly at high condensing temperatures. COP reduces as the condenser temperature increases, but the effect becomes marginal at low evaporator temperatures. It will be shown later that actual vapour compression refrigeration systems also behave in a manner similar to that of Carnot refrigeration systems as far as the performance trends are concerned.

Practical difficulties with Carnot refrigeration system:

It is difficult to build and operate a Carnot refrigeration system due to the following practical difficulties:

i. During process 1-2, a mixture consisting of liquid and vapour have to be compressed isentropically in the compressor. Such a compression is known as *wet compression* due to the presence of liquid. In practice, wet compression is very difficult especially with reciprocating compressors. This problem is particularly severe in case of high speed reciprocating compressors, which get damaged due to the presence of liquid droplets in the vapour. Even though some types of compressors can tolerate the presence of liquid in vapour, since reciprocating compressors are most widely is refrigeration, traditionally *dry compression* (compression of vapour only) is preferred to wet compression.

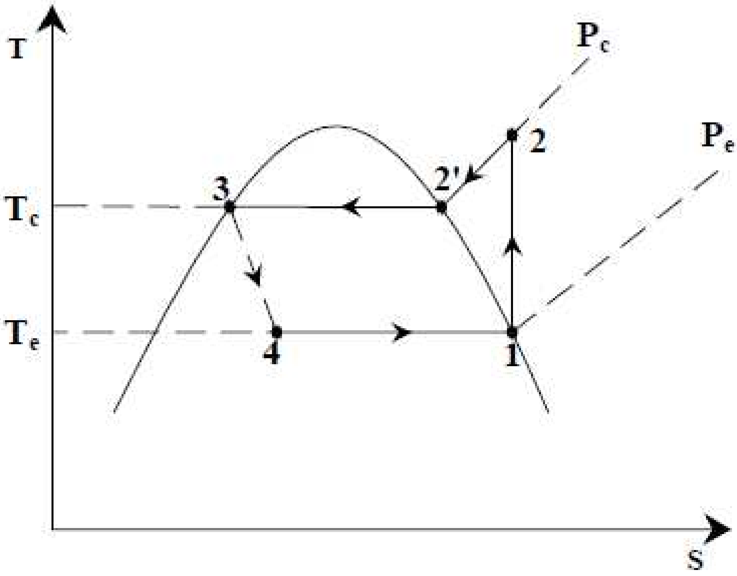
ii. The second practical difficulty with Carnot cycle is that using a turbine and extracting work from the system during the isentropic expansion of liquid refrigerant is not economically feasible, particularly in case of small capacity systems. This is due to the fact that the specific work output (per kilogram of refrigerant) from the turbine is given by:

$$w_{3-4} = \int_{P_e}^{P_c} v.dp$$

since the specific volume of liquid is much smaller compared to the specific volume of a vapour/gas, the work output from the turbine in case of the liquid will be small. In addition, if one considers the inefficiencies of the turbine, then the net output will be further reduced. As a result using a turbine for extracting the work from the high pressure liquid is not economically justified in most of the cases. One way of achieving dry compression in Carnot refrigeration cycle is to have two compressors – one isentropic and one isothermal as shown in Fig.

As shown in Fig.10.4, the Carnot refrigeration system with dry compression consists of one isentropic compression process (1-2) from evaporator pressure P_e to an intermediate pressure P_i and temperature T_c, followed by an isothermal compression process (2-3) from the intermediate pressure P_i to the condenser pressure P_c. Though with this modification the problem of wet compression can be avoided, still this modified system is not practical due to the difficulty in achieving true isothermal compression using high speed compressors. In addition, use of two compressors in place of one is not economically justified.

From the above discussion, it is clear that from practical considerations, the Carnot refrigeration system need to be modified. Dry compression with a single compressor is possible if the isothermal heat rejection process is replaced by isobaric heat rejection process. Similarly, the isentropic expansion process can be replaced by an isenthalpic throttling process. A refrigeration system, which incorporates these two changes is known as Evans-Perkins or reverse Rankine cycle. This is the theoretical cycle on which the actual vapour compression refrigeration systems are based regions or was harvested in winter and stored in icehouses for use in summer. The ice trade reached its peak in 1872 when America alone exported 225000 tonnes of ice to various countries as far as China and Australia. However, with the advent of artificial refrigeration the ice trade gradually declined.



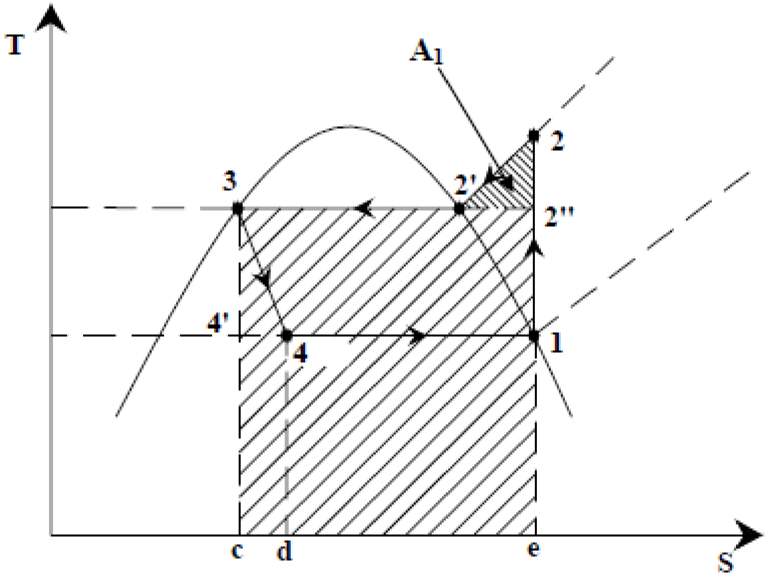
Standard Vapour compression refrigeration system

Standard Vapour Compression Refrigeration System (VCRS)

Figure 10.5 shows the schematic of a standard, saturated, single stage (SSS) vapour compression refrigeration system and the operating cycle on a T s diagram. As shown in the figure the standard single stage, saturated vapour compression refrigeration system consists of the following four processes:

Process 1-2: Isentropic compression of saturated vapour in compressor
Process 2-3: Isobaric heat rejection in condenser
Process 3-4: Isenthalpic expansion of saturated liquid in expansion device
Process 4-1: Isobaric heat extraction in the evaporator

By comparing with Carnot cycle, it can be seen that the standard vapour compression refrigeration cycle introduces two irreversibilities: 1) Irreversibility due to non-isothermal heat rejection (process 2-3) and 2) Irreversibility due to isenthalpic throttling (process 3- 4). As a result, one would expect the theoretical COP of standard cycle to be smaller than that of a Carnot system for the same heat source and sink temperatures. Due to these irreversibilities, the cooling effect reduces and work input increases, thus reducing the system COP. This can be explained easily with the help of the cycle diagrams on T- s charts. Figure below shows comparison between Carnot and standard VCRS in terms of
refrigeration effect.



Comparison between Carnot and standard VCRS

The heat extraction (evaporation) process is reversible for both the Carnot cycle and VCRS cycle. Hence the refrigeration effect is given by:

For Carnot refrigeration cycle (1-2’’-3-4’):

$$q_{e,Carnot} = q_{4'-1} = \int_{4'}^{1} T.ds = T_e(s_1 - s_{4'}) = \text{area } e-1-4'-c-e \tag{10.7}$$

For VCRS cycle (1-2-3-4):

$$q_{e,VCRS} = q_{4-1} = \int_4^1 T.ds = T_e(s_1 - s_4) = \text{area } e-1-4-d-e \tag{10.8}$$

thus there is a reduction in refrigeration effect when the isentropic expansion process of Carnot cycle is replaced by isenthalpic throttling process of VCRS cycle, this reduction is equal to the area d-4-4'-c-d (area A_2) and is known as *throttling loss*. The throttling loss is equal to the enthalpy difference between state points 3 and 4', i.e,

$$q_{e,Carnot} - q_{VCRS} = \text{area } d-4-4'-c-d = (h_3 - h_{4'}) = (h_4 - h_{4'}) = \text{area } A_2 \tag{10.9}$$

It is easy to show that the loss in refrigeration effect increases as the evaporator temperature decreases and/or condenser temperature increases. A practical consequence of this is a requirement of higher refrigerant mass flow rate.

The heat rejection in case of VCRS cycle also increases when compared to Carnot cycle.

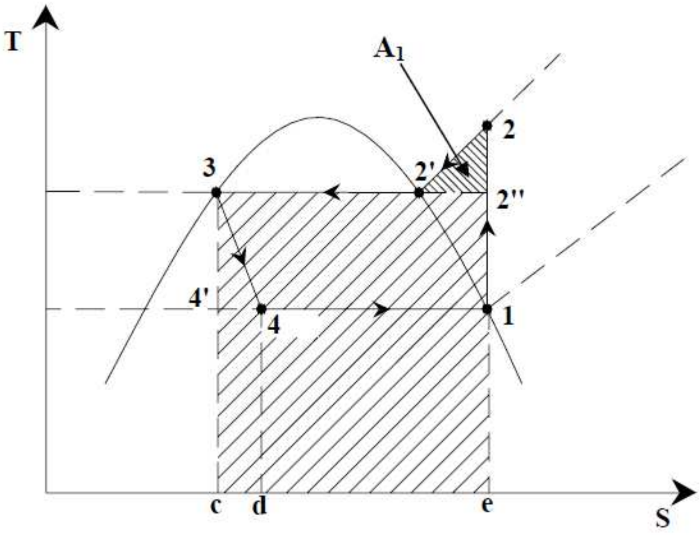


Fig.10.6(b). *Comparative evaluation of heat rejection rate of VCRS and Carnot cycle*

As shown in Fig.10.6(b), the heat rejection in case of Carnot cycle (1-2''-3-4') is given by:

$$q_{c,Carnot} = -q_{2''-3} = -\int_{2''}^{3} T.ds = T_c(s_{2''} - s_3) = \text{area } e-2''-3-c-e \tag{10.10}$$

In case of VCRS cycle, the heat rejection rate is given by:

$$q_{c,VCRS} = -q_{2-3} = -\int_2^3 T.ds = \text{area } e-2-3-c-e \tag{10.11}$$

Hence the increase in heat rejection rate of VCRS compared to Carnot cycle is equal to the area 2''-2-2' (area A_1). This region is known as *superheat horn*, and is due to the

If we define the cycle efficiency, η_R as the ratio of COP of VCRS cycle to the COP of Carnot cycle, then:

$$\eta_R = \frac{COP_{VCRS}}{COP_{Carnot}} = \left[\frac{1 - \left(\frac{\text{area } A_2}{q_{e,Carnot}} \right)}{1 + \left(\frac{\text{area } A_1 + \text{area } A_2}{w_{net,Carnot}} \right)} \right] \tag{10.18}$$

The cycle efficiency (also called as second law efficiency) is a good indication of the deviation of the standard VCRS cycle from Carnot cycle. Unlike Carnot COP, the cycle efficiency depends very much on the shape of T s diagram, which in turn depends on the nature of the working fluid.

If we assume that the potential and kinetic energy changes during isentropic compression process 1-2 are negligible, then the work input w_{1-2} is given by:

$$w_{1-2,VCRS} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f) \tag{10.19}$$

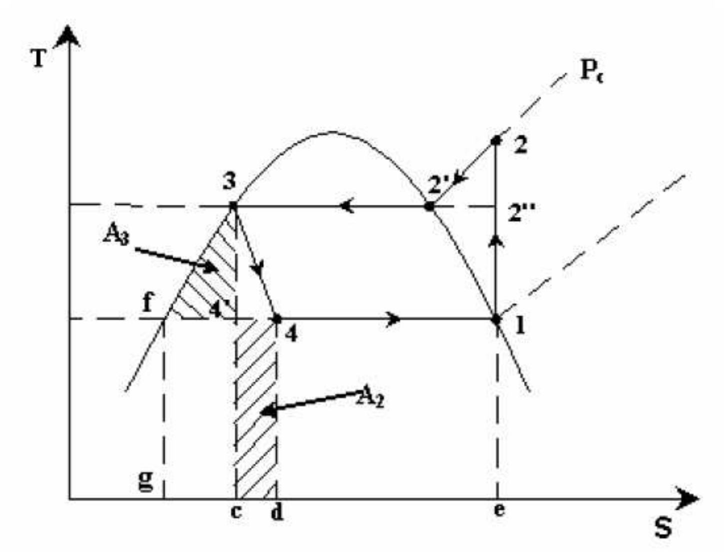


***Fig.10.7**. Figure showing saturated liquid line 3-f coinciding with the constant pressure line*

Now as shown in Fig.10.7, if we further assume that the saturated liquid line 3-f coincides with the constant pressure line P_c in the subcooled region (which is a reasonably good assumption), then from the 2nd Tds relation;

$$Tds = dh - v\, dP = dh; \text{ when P is constant}$$

$$\therefore (h_2 - h_f) = \int_2^f Tds = \text{area } e-2-3-f-g-e \tag{10.20}$$

$$\text{and, } (h_1 - h_f) = \int_1^f T\,ds = \text{area } e-1-f-g-e \tag{10.21}$$

Substituting these expressions in the expression for net work input, we obtain the compressor work input to be equal to area 1-2-3-f-1. Now comparing this with the earlier expression for work input (area 1-2-3-4'-c-d-4-1), we conclude that area A_2 is equal to area A_3.

As mentioned before, the losses due to superheat (area A_1) and throttling (area $A_2 \approx A_3$) depend very much on the shape of the vapor dome (saturation liquid and vapour curves) on T s diagram. The shape of the saturation curves depends on the nature of refrigerant. Figure 10.8 shows T s diagrams for three different types of refrigerants.

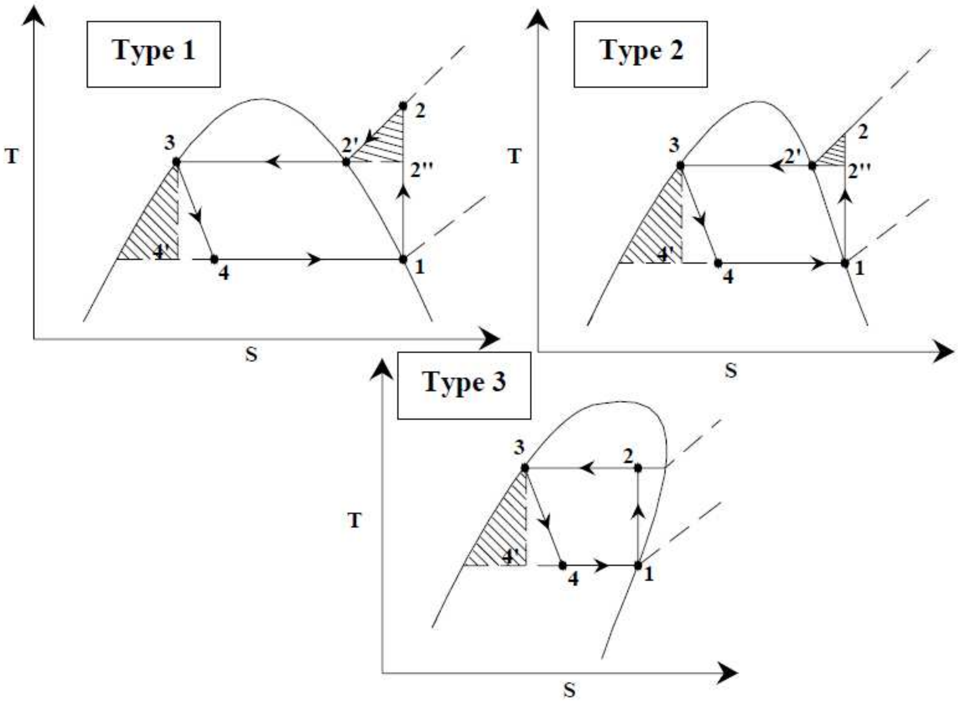


***Fig.10.8**. T-s diagrams for three different types of refrigerants*

Refrigerants such as ammonia, carbon di-oxide and water belong to Type 1. These refrigerants have symmetrical saturation curves (vapour dome), as a result both the superheat and throttling losses (areas A_1 and A_3) are significant. That means deviation of VCRS cycle from Carnot cycle could be significant when these refrigerants are used as working fluids. Refrigerants such as CFC11, CFC12, HFC134a belong to Type 2, these refrigerants have small superheat losses (area A_1) but large throttling losses (area A_3). High molecular weight refrigerants such as CFC113, CFC114, CFC115, iso-butane belonging to Type 3, do not have any superheat losses, i.e., when the compression inlet condition is saturated (point 1), then the exit condition will be in the 2-phase region, as a result it is not necessary to superheat the refrigerant. However, these refrigerants

$$\dot{Q}_c = \dot{m}_r(h_2 - h_3) \tag{10.25}$$

where h_3 and h_2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively.

The condenser pressure P_c is the saturation pressure corresponding to evaporator temperature T_c, i.e.,

$$P_c = P_{sat}(T_c) \tag{10.26}$$

Expansion device: For the isenthalpic expansion process, the kinetic energy change across the expansion device could be considerable, however, if we take the control volume, well downstream of the expansion device, then the kinetic energy gets dissipated due to viscous effects, and

$$h_3 = h_4 \tag{10.27}$$

The exit condition of the expansion device lies in the two-phase region, hence applying the definition of quality (or dryness fraction), we can write:

$$h_4 = (1 - x_4)h_{f,e} + x_4 h_{g,e} = h_f + x_4 h_{fg} \tag{10.28}$$

where x_4 is the quality of refrigerant at point 4, $h_{f,e}$, $h_{g,e}$, h_{fg} are the saturated liquid enthalpy, saturated vapour enthalpy and latent heat of vaporization at evaporator pressure, respectively.

The COP of the system is given by:

$$\text{COP} = \left(\frac{\dot{Q}_e}{\dot{W}_c}\right) = \left(\frac{\dot{m}_r(h_1 - h_4)}{\dot{m}_r(h_2 - h_1)}\right) = \frac{(h_1 - h_4)}{(h_2 - h_1)} \tag{10.29}$$

At any point in the cycle, the mass flow rate of refrigerant $\dot{m}_r$ can be written in terms of volumetric flow rate and specific volume at that point, i.e.,

$$\dot{m}_r = \dot{V}/v \tag{10.30}$$

applying this equation to the inlet condition of the compressor,

$$\dot{m}_r = \dot{V}_1/v_1 \tag{10.31}$$

where $\dot{V}_1$ is the volumetric flow rate at compressor inlet and v_1 is the specific volume at compressor inlet. At a given compressor speed, $\dot{V}_1$ is an indication of the size of the compressor. We can also write, the refrigeration capacity in terms of volumetric flow rate as:

experience significant throttling losses. Since the compressor exit condition of Type 3 refrigerants may fall in the two-phase region, there is a danger of wet compression leading to compressor damage. Hence for these refrigerants, the compressor inlet condition is chosen such that the exit condition does not fall in the two-phase region. This implies that the refrigerant at the inlet to the compressor should be superheated, the extent of which depends on the refrigerant.

Superheat and throttling losses:

It can be observed from the discussions that the superheat loss is fundamentally different from the throttling loss. The superheat loss increases only the work input to the compressor, it does not effect the refrigeration effect. In heat pumps superheat is not a loss, but a part of the useful heating effect. However, the process of throttling is inherently irreversible, and it increases the work input and also reduces the refrigeration effect.

10.5. Analysis of standard vapour compression refrigeration system

A simple analysis of standard vapour compression refrigeration system can be carried out by assuming a) Steady flow; b) negligible kinetic and potential energy changes across each component, and c) no heat transfer in connecting pipe lines. The steady flow energy equation is applied to each of the four components.

Evaporator: Heat transfer rate at evaporator or *refrigeration capacity*, $\dot{Q}_e$ is given by:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4) \tag{10.22}$$

where $\dot{m}_r$ is the refrigerant mass flow rate in kg/s, h_1 and h_4 are the specific enthalpies (kJ/kg) at the exit and inlet to the evaporator, respectively. $(h_1 - h_4)$ is known as specific refrigeration effect or simply *refrigeration effect*, which is equal to the heat transferred at the evaporator per kilogram of refrigerant. The evaporator pressure P_e is the saturation pressure corresponding to evaporator temperature T_e, i.e.,

$$P_e = P_{sat}(T_e) \tag{10.23}$$

Compressor: Power input to the compressor, $\dot{W}_c$ is given by:

$$\dot{W}_c = \dot{m}_r (h_2 - h_1) \tag{10.24}$$

where h_2 and h_1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, respectively. $(h_2 - h_1)$ is known as specific work of compression or simply *work of compression*, which is equal to the work input to the compressor per kilogram of refrigerant.

Condenser: Heat transfer rate at condenser, $\dot{Q}_c$ is given by:

$$\dot{Q}_e = \dot{m}_r(h_1 - h_4) = \dot{V}_1\left(\frac{h_1 - h_4}{v_1}\right) \tag{10.32}$$

where $\left(\frac{h_1 - h_4}{v_1}\right)$ is called as *volumetric refrigeration effect* (kJ/m^3 of refrigerant).

Generally, the type of refrigerant, required refrigeration capacity, evaporator temperature and condenser temperature are known. Then from the evaporator and condenser temperature one can find the evaporator and condenser pressures and enthalpies at the exit of evaporator and condenser (saturated vapour enthalpy at evaporator pressure and saturated liquid enthalpy at condenser pressure). Since the exit condition of the compressor is in the superheated region, two independent properties are required to fix the state of refrigerant at this point. One of these independent properties could be the condenser pressure, which is already known. Since the compression process is isentropic, the entropy at the exit to the compressor is same as the entropy at the inlet, s_1 which is the saturated vapour entropy at evaporator pressure (known). Thus from the known pressure and entropy the exit state of the compressor could be fixed, i.e.,

$$\begin{aligned} h_2 &= h(P_c, s_2) = h(P_c, s_1) \\ s_1 &= s_2 \end{aligned} \tag{10.33}$$

The quality of refrigerant at the inlet to the evaporator (x_4) could be obtained from the known values of h_3, $h_{f,e}$ and $h_{g,e}$.

Once all the state points are known, then from the required refrigeration capacity and various enthalpies one can obtain the required refrigerant mass flow rate, volumetric flow rate at compressor inlet, COP, cycle efficiency etc.

Use of Pressure-enthalpy (P-h) charts:

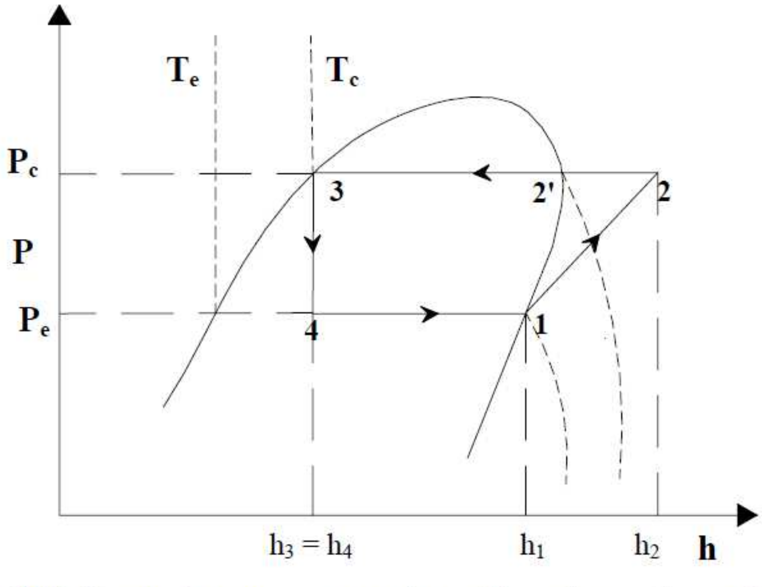


Fig.10.9. *Standard vapour compression refrigeration cycle on a P-h chart*

Since the various performance parameters are expressed in terms of enthalpies, it is very convenient to use a pressure – enthalpy chart for property evaluation and performance analysis. The use of these charts was first suggested by Richard Mollier. Figure 10.9 shows the standard vapour compression refrigeration cycle on a P-h chart. As discussed before, in a typical P-h chart, enthalpy is on the x-axis and pressure is on y-axis. The isotherms are almost vertical in the subcooled region, horizontal in the two-phase region (for pure refrigerants) and slightly curved in the superheated region at high pressures, and again become almost vertical at low pressures. A typical P-h chart also shows constant specific volume lines (isochors) and constant entropy lines (isentropes) in the superheated region. Using P-h charts one can easily find various performance parameters from known values of evaporator and condenser pressures.

In addition to the P-h and T-s charts one can also use thermodynamic property tables from solving problems related to various refrigeration cycles.

In addition to the basic components, an actual vapour compression refrigeration consists of several accessories for safe and satisfactory functioning of the system. These include: compressor controls and safety devices such as overload protectors, high and low pressure cutouts, oil separators etc., temperature and flow controls, filters, driers, valves, sight glass etc. Modern refrigeration systems have automatic controls, which do not require continuous manual supervision.

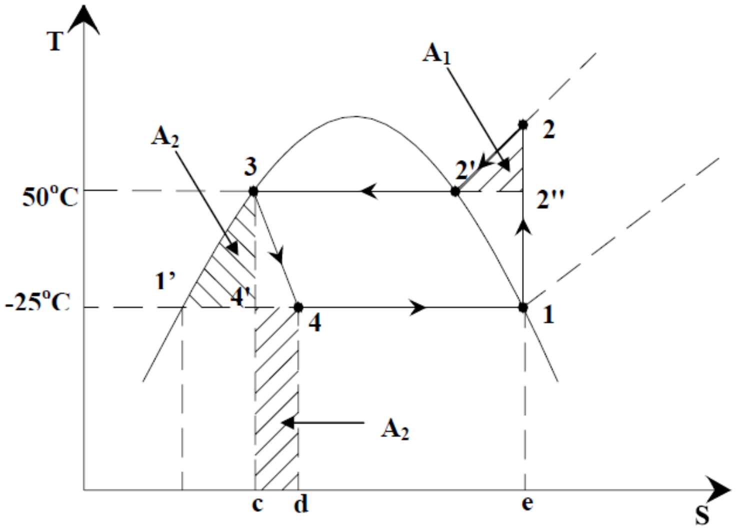
Problem:

9. A SSS vapour compression refrigeration system based on refrigerant R 134a operates between an evaporator temperature of –25°C and a condenser temperature of 50°C. Assuming isentropic compression, find:

a) COP of the system
b) Work input to compressor
c) Area of superheat horn (additional work required due to superheat)

Throttling loss (additional work input due to throttling in place of isentropic expansion) assuming the isobar at condenser pressure to coincide with saturated liquid line.

Ans.: Given: Refrigerant : R 134a
T_e = -25°C
T_c = 50°C



Using refrigerant R134a property data, required properties at various state points are:

State Point	**T (°C)**	**P (bar)**	**h (kJ/kg)**	**s (kJ/kg.K)**	**Quality**
1	-25.0	1.064	383.4	1.746	1.0
2	**60.7**	13.18	436.2	1.746	Superheated
3	50.0	13.18	271.6	1.237	0.0
4	-25.0	1.064	271.6	**1.295**	**0.4820**
1'	-25.0	1.064	167.2	**0.8746**	0.0
2'	50.0	13.18	423.4	1.707	**1.0**
2"	50.0	10.2	430.5	1.746	Superheated
4'	-25.0	1.064	257.1	1.237	0.4158

a) **COP = $(h_1-h_4)/(h_2-h_1)$ = 2.1174**

b) **Work input to compressor, $W_c = (h_2-h_1)$ = 52.8 kJ/kg**

c) Superheat horn area, area A_1:

Area A_1 = Area under 2-2' – Area under 2"-2'

Area under 2-2': Tds = (dh-vdP) = dh = h_2-$h_{2'}$ (dp = 0)

⇒ **Area under 2-2' =** h_2-$h_{2'}$ = 12.8 kJ/kg

Area under 2"-2' = Tds = Tc ($s_{2''}$-$s_{2'}$) = 12.6 kJ/kg

Superheat horn area = Area A_1 = (12.8 – 12.6) = 0.2 kJ/kg

d) Throttling loss, Area A_2 (assuming the saturated liquid line to coincide with isobar at condenser pressure):

Area A_2 = Area under 3-1'–Area under 4'-1' = (h_3–$h_{1'}$) – T_e(s_3-$s_{1'}$) ($s_3 = s_{4'}$)

Throttling area = (271.6–167.2) – 248.15(1.237–0.8746) = 14.47 kJ/kg

Alternatively:

Throttling area = Area under 4-4' = T_e(s_4-$s_{4'}$) = 248.15(1.295–1.237) = 14.4 kJ/kg

Check:

W_{sss} = W_{Carnot}+Area A_1+Area A_2

W_{Carnot} = (T_c-T_e)(s_1–$s_{4'}$) = 75(1.746-1.237) = 38.2 kJ/kg

W_{sss} = 38.2+14.4+0.2 = 52.8 kJ/kg

10. In a R22 based refrigeration system, a liquid-to-suction heat exchanger (LSHX) with an effectiveness of 0.65 is used. The evaporating and condensing temperatures are 7.2°C and 54.4°C respectively. Assuming the compression process to be isentropic, find:

a) Specific refrigeration effect
b) Volumic refrigeration effect
c) Specific work of compression
d) COP of the system
e) Temperature of vapour at the exit of the compressor

Comment on the use of LSHX by comparing the performance of the system with a SSS cycle operating between the same evaporator and condensing temperatures.

Ans.:

Given: Refrigerant : R 22
T_e = 7.2°C
T_c = 54.4°C
Effectiveness of LSHX,ε_X = 0.65

From the above data and using refrigerant property values for R 22 at various state points are:

State Point	T (°C)	P (bar)	h (kJ/kg)	s (kJ/kg.K)	v m^3/kg	Quality
1	7.2	6.254	407.6	1.741	0.03773	1.0
2	**37.88**	6.254	430.7	1.819	0.04385	Superheated
3	**104.9**	21.46	466.8	1.819	-	Superheated
4	54.4	21.46	269.5	**1.227**	-	0.0
5	**37.65**	21.46	**246.4**	**1.154**	-	Subcooled
6	7.2	6.254	246.4	**1.166**	-	**0.1903**
6'	7.2	6.254	269.5	**1.248**	-	**0.3063**
3'	**74.23**	21.46	438.6	**1.741**	-	Superheated
1'	7.2	6.254	208.5	**1.030**	-	0.0

With LSHX:

a) Refrigeration effect = (h_1-h_6) = **161.2 kJ/kg**

b) Volumic refrigeration effect = $(h_1-h_6)/v_2$ = **3676.2 kJ/m^3**

c) Work of compression = (h_3-h_2) = **36.1 kJ/kg**

d) COP = $(h_1-h_6)/\ (h_3-h_2)$ = **4.465**

e) Temperature at compressor exit (from P_c and $s_3=s_2$) = **104.9°C**

Without LSHX:

a) Refrigeration effect = $(h_1-h_{6'})$ = **138.1 kJ/kg**

b) Volumic refrigeration effect = $(h_1-h_{6'})/v_1$ = **3660.2 kJ/m^3**

c) Work of compression = $(h_{3'}-h_1)$ = **31.0 kJ/kg**

d) COP = $(h_1-h_{6'})/\ (h_{3'}-h_1)$ = **4.455**

e) Temperature at compressor exit (from P_c and $s_1=s_{3'}$) = **74.23°C**

Parameter	**With LSHX**	**Without LSHX**
Refrigeration effect, kJ/kg	**161.2**	**138.1**
Ref. quality at evaporator inlet	**0.1903**	**0.3063**
Vol. Refrigeration effect, kJ/m^3	**3676.2**	**3660.2**
Work of compression, kJ/kg	**36.1**	**31.0**
COP	**4.465**	**4.455**
Compressor exit temperature, ^{o}C	**104.9**	**74.23**

Comments:

a) There is no appreciable change in COP with the addition of LSHX
b) Quality of refrigerant at evaporator inlet is significantly lower with LSHX
c) Discharge temperature is significantly high with LSHX
d) For refrigerant R-22, use of LSHX does not improve the performance of the system significantly, however, the evaporator with LSHX performs better due to the lower vapour fraction at its inlet

MODULE-II

Compound Vapour Compression Refrigeration Systems

12.1. Introduction

A single stage vapour compression refrigeration system has one low side pressure (evaporator pressure) and one high side pressure (condenser pressure). The performance of single stage systems shows that these systems are adequate as long as the temperature difference between evaporator and condenser (*temperature lift*) is small. However, there are many applications where the temperature lift can be quite high. The temperature lift can become large either due to the requirement of very low evaporator temperatures and/or due to the requirement of very high condensing temperatures. For example, in frozen food industries the required evaporator can be as low as –40°C, while in chemical industries temperatures as low as –150°C may be required for liquefaction of gases. On the high temperature side the required condensing temperatures can be very high if the refrigeration system is used as a heat pump for heating applications such as process heating, drying etc. However, as the temperature lift increases the single stage systems become inefficient and impractical. For example, Fig. 12.1 shows the effect of decreasing evaporator temperatures on T s and P h diagrams. It can be seen from the T s diagrams that for a given condenser temperature, as evaporator temperature decreases:

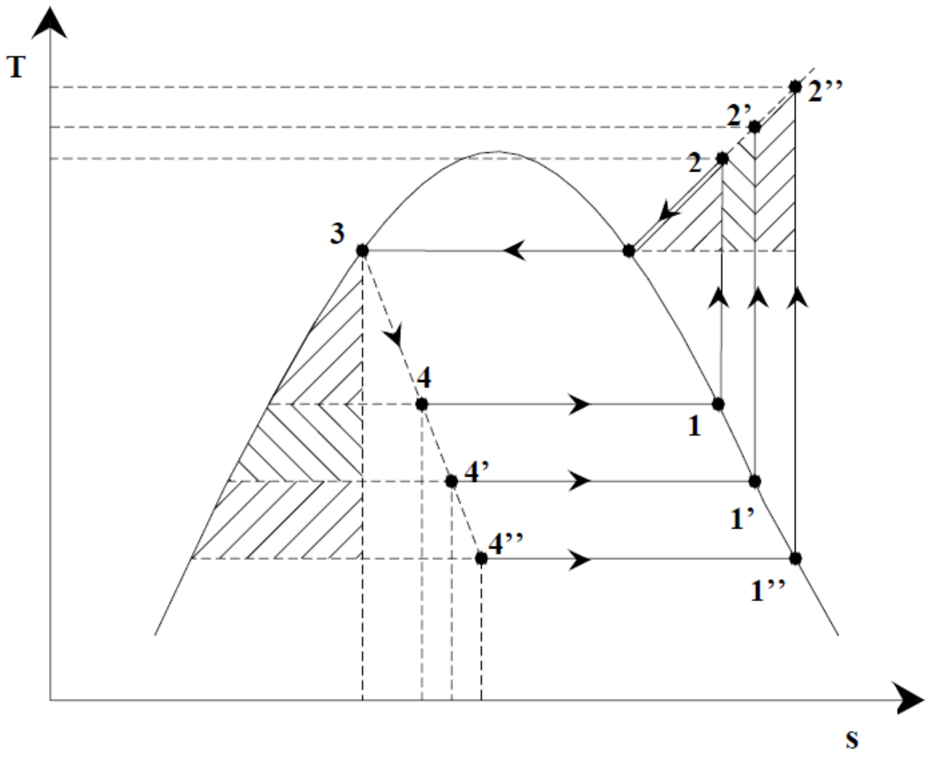


Fig.12.1(a): *Effect of evaporator temperature on cycle performance (T-s diagram)*

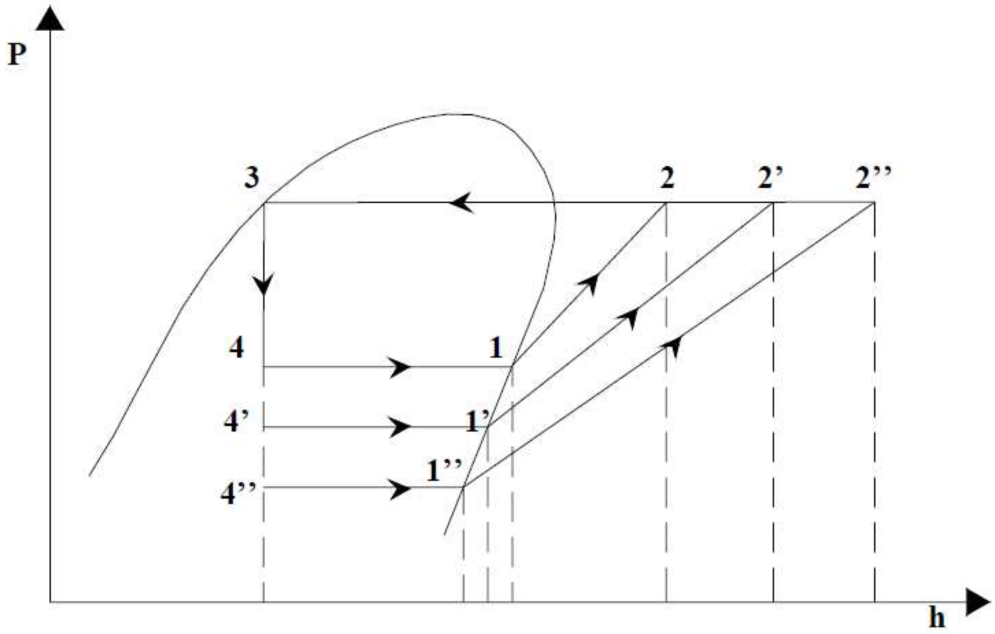


Fig.12.1(b): *Effect of evaporator temperature on cycle performance (P-h diagram)*

A multi-stage system is a refrigeration system with two or more low-side pressures. Multi-stage systems can be classified into:

a) Multi-compression systems
b) Multi-evaporator systems
c) Cascade systems, etc.

Two concepts which are normally integral to multi-pressure systems are, i) flash gas removal, and ii) intercooling. Hence these concepts will be discussed first.

12.2. Flash gas removal using flash tank

It is mentioned above that one of the problems with high temperature lift applications is the high quality of vapour at the inlet to the evaporator. This vapour called as *flash gas* develops during the throttling process. The flash gas has to be compressed to condenser pressure, it does not contribute to the refrigeration effect as it is already in the form of vapour, and it increases the pressure drop in the evaporator. It is possible to improve the COP of the system if the flash gas is removed as soon as it is formed and recompressed to condenser pressure. However, continuous removal of flash gas as soon as it is formed and recompressing it immediately is difficult in practice. One way of improving the performance of the system is to remove the flash gas at an intermediate pressure using a *flash tank*. Figure 12.2 shows the schematic of a flash tank and Fig.12.3 shows the expansion process employing flash tank. A flash tank is a pressure vessel, wherein the refrigerant liquid and vapour are separated at an intermediate pressure. The refrigerant from condenser is first expanded to an intermediate pressure corresponding to the pressure of flash tank, P_i using a low side float valve (process 6-7). The float valve also maintains a constant liquid level in the flash tank. In the flash tank, the refrigerant liquid and vapour are separated. The saturated liquid at point 8 is fed to the evaporator after throttling it to the required evaporator pressure, P_e (point 9) using an expansion valve. Depending upon the type of the system, the saturated vapour in the flash tank (point 3) is either compressed to the condenser pressure or throttled to the evaporator pressure. In the absence of flash tank, the refrigerant condition at the inlet to the evaporator would have been point 9', which has a considerably high vapour quality compared to point 9. As mentioned, the refrigerant liquid and vapour must get separated in the flash tank. This is possible when the upward velocity of the refrigerant vapour in the flash tank is low enough (< 1 m/s) for the refrigerant liquid droplets to fall back into the flash tank due to gravity. Thus the surface area of liquid in the flash tank can be obtained from the volumetric flow rate of refrigerant vapour and the required low refrigerant velocity.

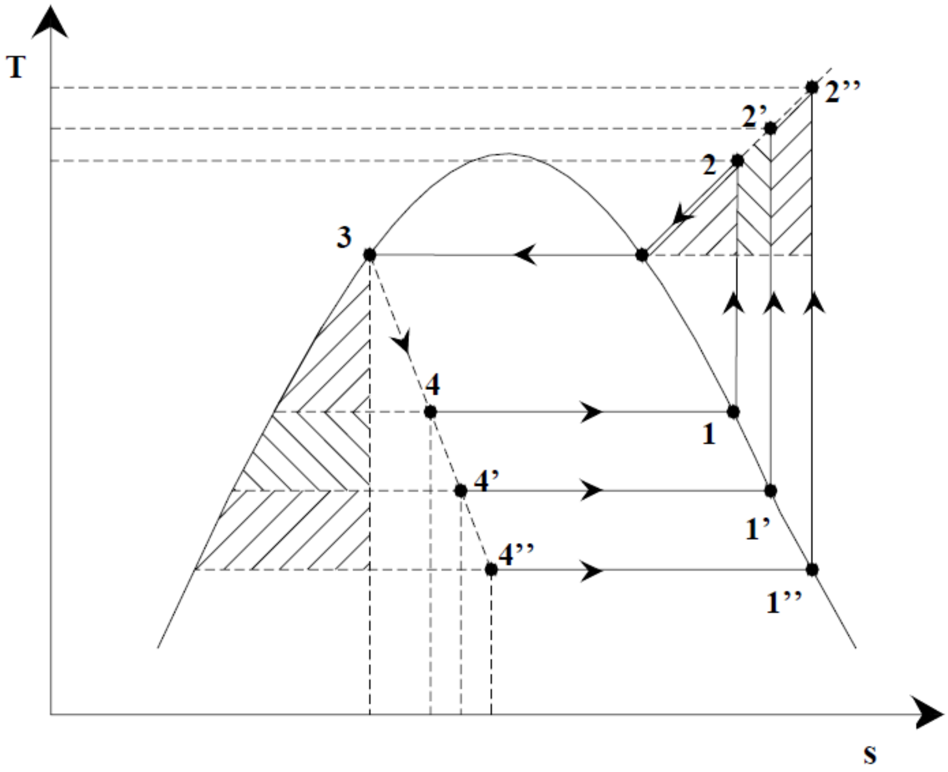


Fig.12.1(a): Effect of evaporator temperature on cycle performance (T-s diagram)

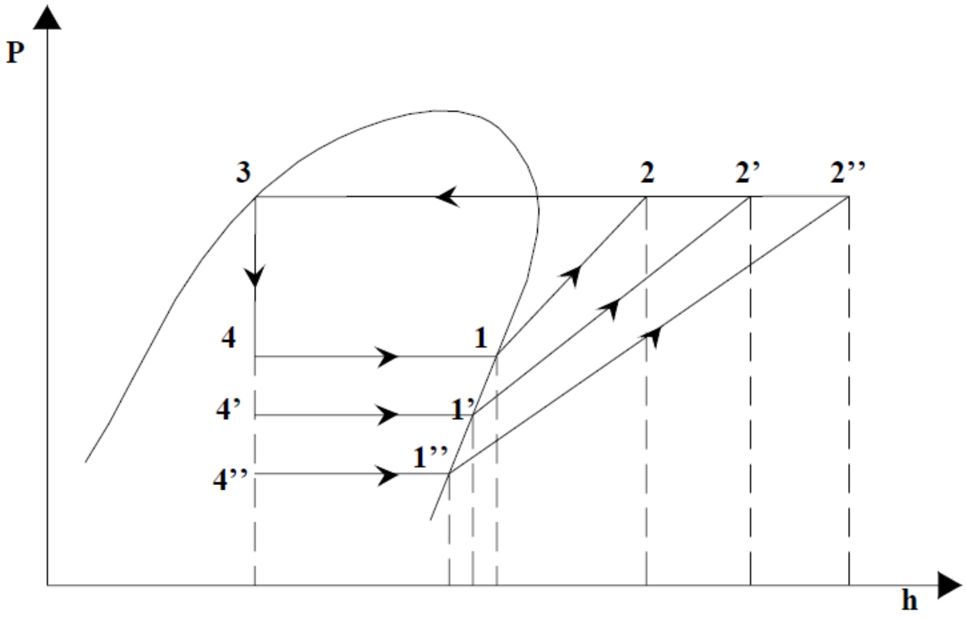


Fig.12.1(b): Effect of evaporator temperature on cycle performance (P-h diagram)

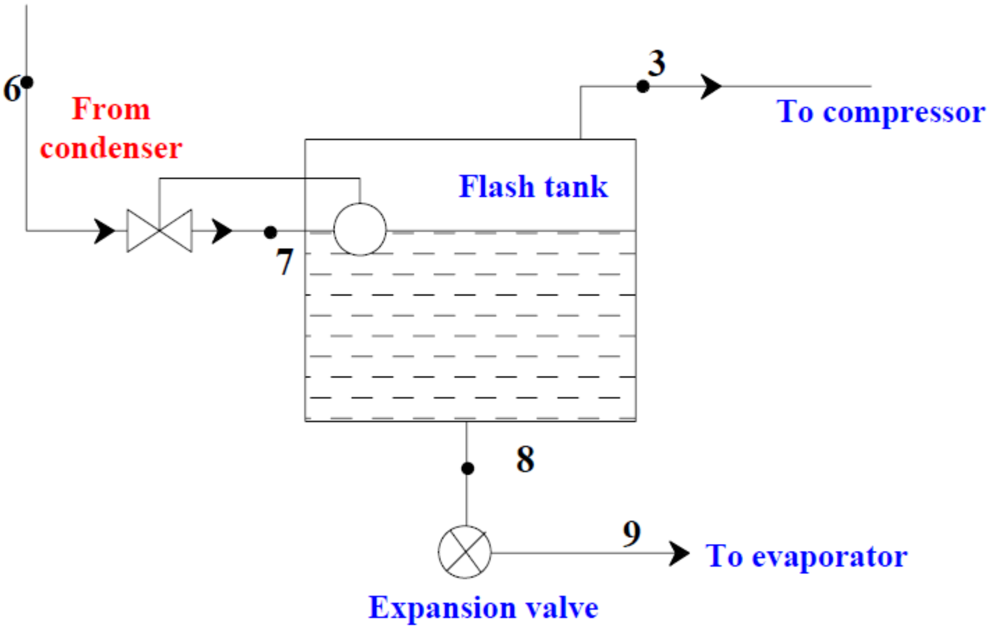


Fig.12.2(a): *Working principle of a flash tank*

12.3. Intercooling in multi-stage compression

The specific work input, w in reversible, polytropic compression of refrigerant vapour is given by:

$$w = -\int_1^2 v.dP = \left(\frac{n}{n-1}\right)P_1 v_1\left[1-\left(\frac{P_2}{P_1}\right)^{(n-1)/n}\right] \tag{12.1}$$

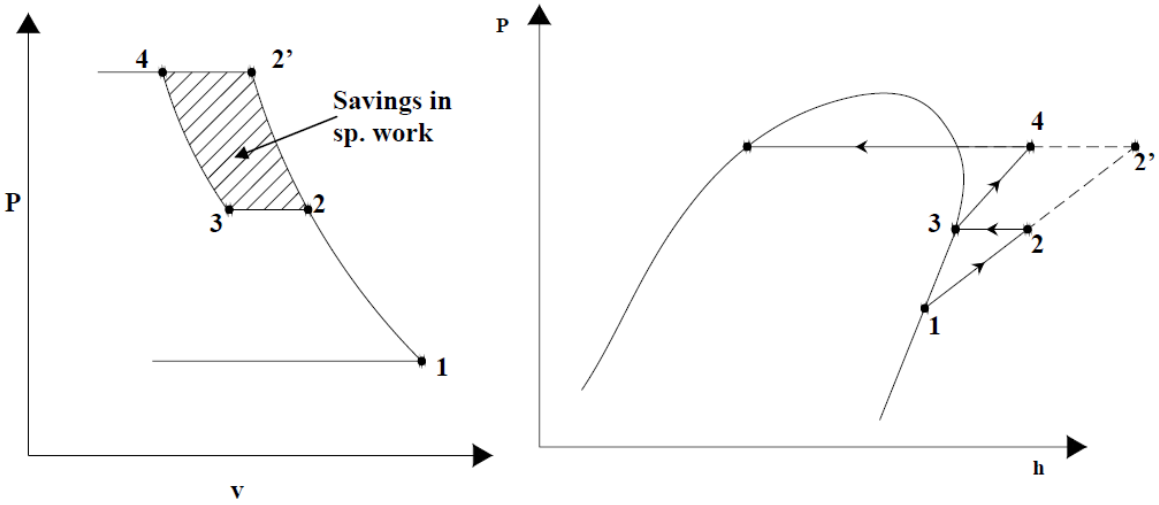


Fig.12.4(a) & (b): *Intercooling in two-stage compression*

where P_1 and P_2 are the inlet and exit pressures of the compressor, v_1 is the specific volume of the refrigerant vapour at the inlet to the compressor and n is the polytropic exponent. From the above expression, it can be seen that specific work input reduces as specific volume, v_1 is reduced. At a given pressure, the specific volume can be reduced by reducing the temperature. This is the principle behind intercooling in multi-stage compression. Figures 12.4 (a) and (b) show the process of intercooling in two-stage compression on Pressure-specific volume (P-v) and P-h diagrams.

As shown in the figures, in stead of compressing the vapour in a single stage from state 1 to state 2', if the refrigerant is compressed from state 1 to an intermediate pressure, state 2, intercooled from 2 to 3 and then compressed to the required pressure (state 4), reduction in work input results. If the processes are reversible, then the savings in specific work is given by the shaded area 2-3-4-2' on P-v diagram. The savings in work input can also be verified from the P-h diagram. On P-h diagram, lines 1-2-2' and 3-4 represent isentropes. Since the slope of isentropes on P-h diagram reduces (lines become flatter) as they move away from the saturated vapour line,

$$(h_4-h_3) < (h_{2'}-h_2) \Rightarrow (h_2-h_1)+(h_4-h_3) < (h_{2'}-h_1) \qquad (12.2)$$

Intercooling of the vapour may be achieved by using either a water-cooled heat exchanger or by the refrigerant in the flash tank. Figures 12.5(a) and (b) show these two systems. Intercooling may not be always possible using water-cooled heat exchangers as it depends on the availability of sufficiently cold water to which the refrigerant from low stage compressor can reject heat. Moreover, with water cooling the refrigerant at the inlet to the high stage compressor may not be saturated. Water cooling is commonly used in air compressors. Intercooling not only reduces the work input but also reduces the compressor discharge temperature leading to better lubrication and longer compressor life.

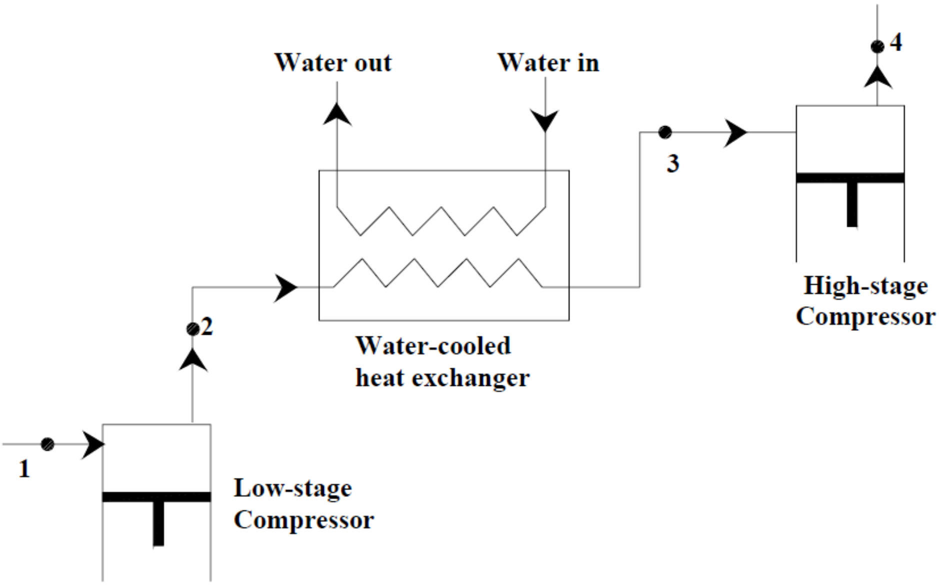


Fig.12.5(b): *Intercooling using external water cooled heat exchanger*

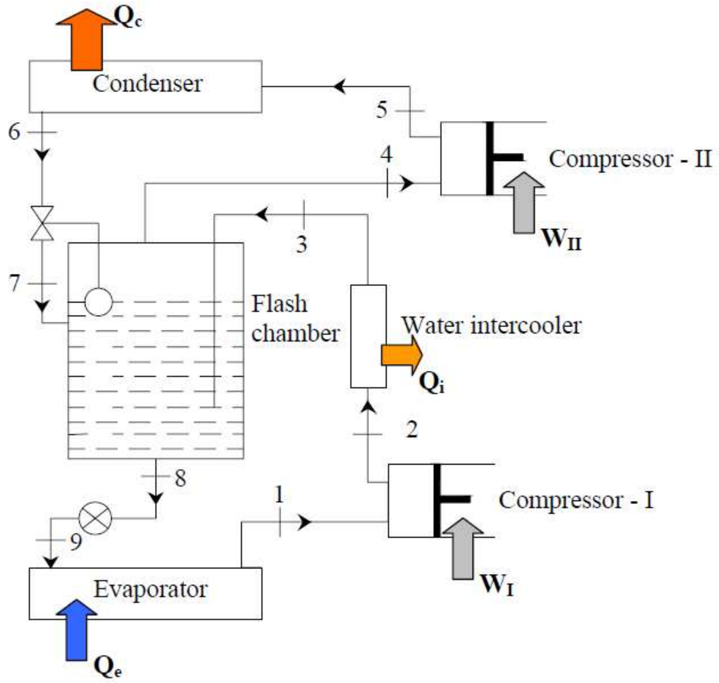


Fig.126(a): *Two-stage vapour compression refrigeration system with flash gas removal using a flash tank and intercooling*

Intercooling using liquid refrigerant from condenser in the flash tank may or may not reduce the power input to the system, as it depends upon the nature of the refrigerant. This is due to the fact that the heat rejected by the refrigerant during intercooling generates additional vapour in the flash tank, which has to be compressed by the high stage compressor. Thus the mass flow rate of refrigerant through the high stage compressor will be more than that of the low stage compressor. Whether total power input to the system decreases or not depends on whether the increased power consumption due to higher mass flow rate is

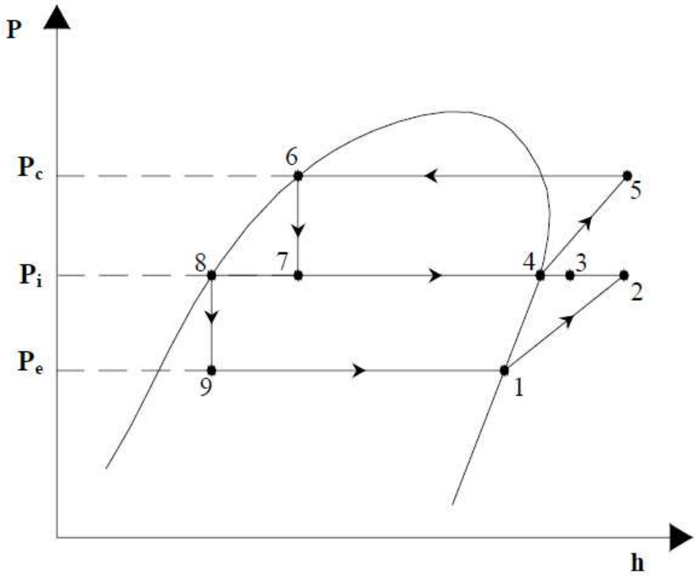


Fig.126(b): *Two-stage vapour compression refrigeration system with flash gas removal using a flash tank and intercooling – P-h diagram*

It can be seen from the above expression that the refrigerant flow through the high-stage compression $\dot{m}_{II}$ can be reduced by reducing the enthalpy of refrigerant vapour entering into the flash tank, h_3 from the water-cooled intercooler.

The amount of additional vapour generated due to de-superheating of the refrigerant vapour from the water-cooled intercooler is given by:

$$\dot{m}_{gen} = \dot{m}_{I}\left[\frac{h_3 - h_4}{h_4 - h_8}\right] \tag{12.24}$$

Thus the vapour generated $\dot{m}_{gen}$ will be zero, if the refrigerant vapour is completely de-superheated in the water-cooled intercooler itself. However, this may not be possible in practice.

For the above system, the COP is given by:

$$COP = \frac{Q_e}{W_I + W_{II}} = \frac{\dot{m}_{I}(h_1 - h_9)}{\dot{m}_{I}(h_2 - h_1) + \dot{m}_{II}(h_5 - h_4)} \tag{12.25}$$

The above system offers several advantages,

a) Quality of refrigerant entering the evaporator reduces thus giving rise to higher refrigerating effect, lower pressure drop and better heat transfer in the evaporator
b) Throttling losses are reduced as vapour generated during throttling from P_c to P_i is separated in the flash tank and recompressed by Compressor-II.
c) Volumetric efficiency of compressors will be high due to reduced pressure ratios
d) Compressor discharge temperature is reduced considerably.

However, one disadvantage of the above system is that since refrigerant liquid in the flash tank is saturated, there is a possibility of liquid flashing ahead of the expansion valve due to pressure drop or heat transfer in the pipelines connecting the flash tank to the expansion device. Sometimes this problem is tackled by using a system with a *liquid subcooler*. As shown in Fig.12.7, in a liquid subcooler the refrigerant liquid from the condenser is subcooled by exchanging heat with the refrigerant liquid in the flash tank. As a result, a small amount of refrigerant vapour is generated in the flash tank, which needs to be compressed in the high-stage compressor. Compared to the earlier system, the temperature of refrigerant liquid from the subcooler will be higher than the saturated refrigerant temperature in the flash tank due to indirect contact heat transfer. However, since the refrigerant at the inlet to the expansion valve is at high pressure and is subcooled, there is less chance of flashing of liquid ahead of expansion valve.

From mass and energy balance across expansion valve,

$$\dot{m}_8 = \dot{m}_9 \tag{12.7}$$

$$h_8 = h_9 \tag{12.8}$$

From mass and energy balance across evaporator:

$$\dot{m}_9 = \dot{m}_1 \tag{12.9}$$

$$Q_e = \dot{m}_1(h_1 - h_9) \tag{12.10}$$

From mass and energy balance across low-stage compressor, Compressor-I:

$$\dot{m}_9 = \dot{m}_1 = \dot{m}_I \tag{12.11}$$

$$W_I = \dot{m}_I\,(h_2 - h_1) \tag{12.12}$$

where $\dot{m}_I$ is the mass flow rate of refrigerant through Compressor-I.

From mass and energy balance across water-cooled intercooler:

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_I \tag{12.13}$$

$$Q_I = \dot{m}_I\,(h_2 - h_3) \tag{12.14}$$

where Q_I is the heat transferred by the refrigerant to the cooling water in the intercooler.

From mass and energy balance across high-stage compressor, Compressor-II:

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_{II} \tag{12.15}$$

$$W_{II} = \dot{m}_{II}\,(h_5 - h_4) \tag{12.16}$$

where $\dot{m}_{II}$ is the mass flow rate of refrigerant through Compressor-II.

Finally, from mass and energy balance across condenser:

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_{II} \tag{12.17}$$

$$Q_c = \dot{m}_{II}\,(h_5 - h_6) \tag{12.18}$$

Finally, from mass and energy balance across the float valve:

$$\dot{m}_6 = \dot{m}_7 = \dot{m}_{II} \tag{12.19}$$

$$h_6 = h_7 \tag{12.20}$$

From the above set of equations, it can be easily shown that for the flash tank:

$$\dot{m}_7 = \dot{m}_4 = \dot{m}_{II} \tag{12.21}$$

$$\dot{m}_3 = \dot{m}_8 = \dot{m}_I \tag{12.22}$$

$$\dot{m}_{II} = \dot{m}_I\left[\frac{h_3 - h_8}{h_4 - h_7}\right] \tag{12.23}$$

12.6. Use of flash tank for intercooling only

Sometimes the flash tank is used for intercooling of the refrigerant vapour between the low and high-stage compressors. It is not used for flash gas removal. Figures 12.9 (a) and (b) show the system schematic and P-h diagram of a two-stage compression system where the flash tank is used for intercooling only.

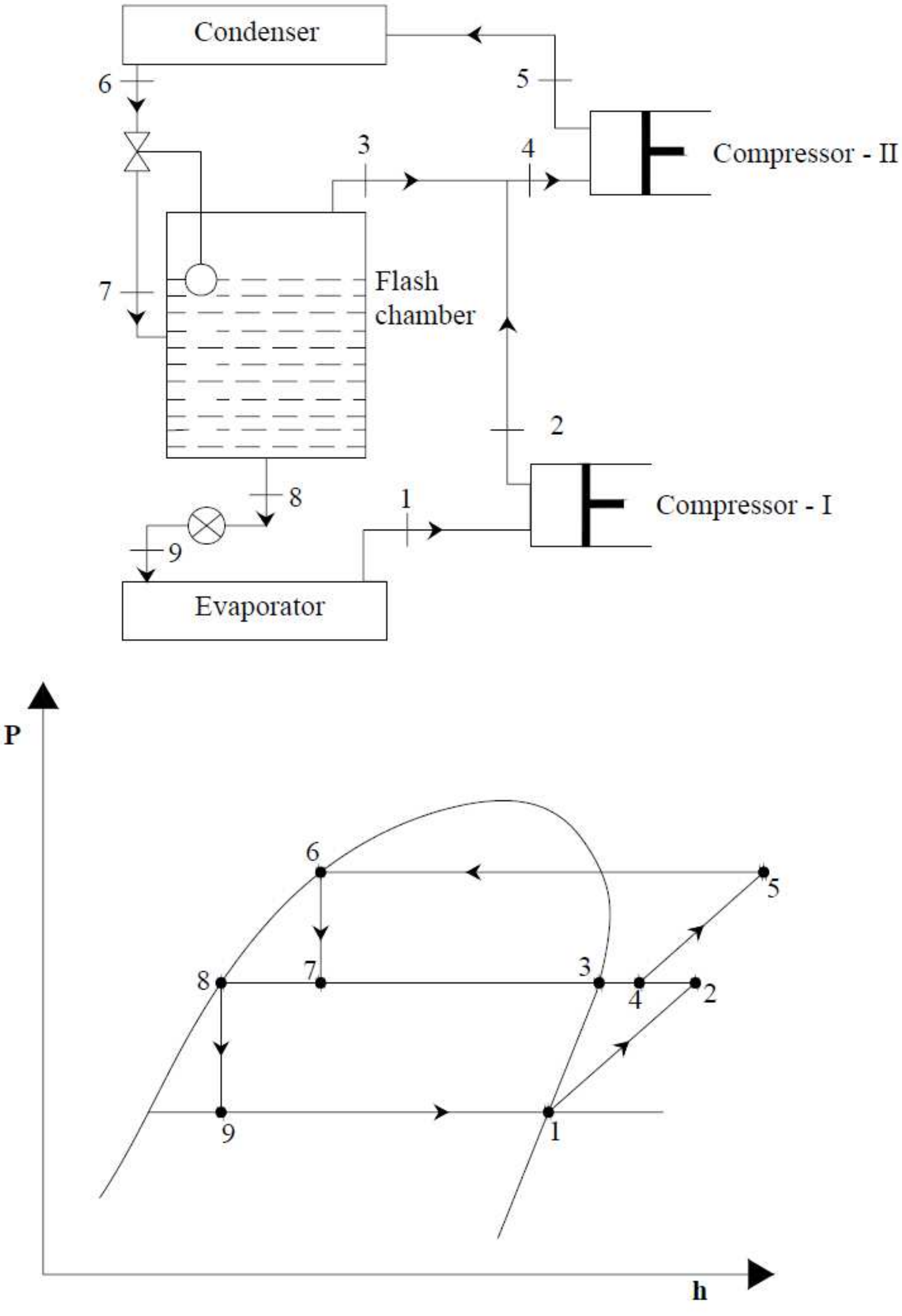
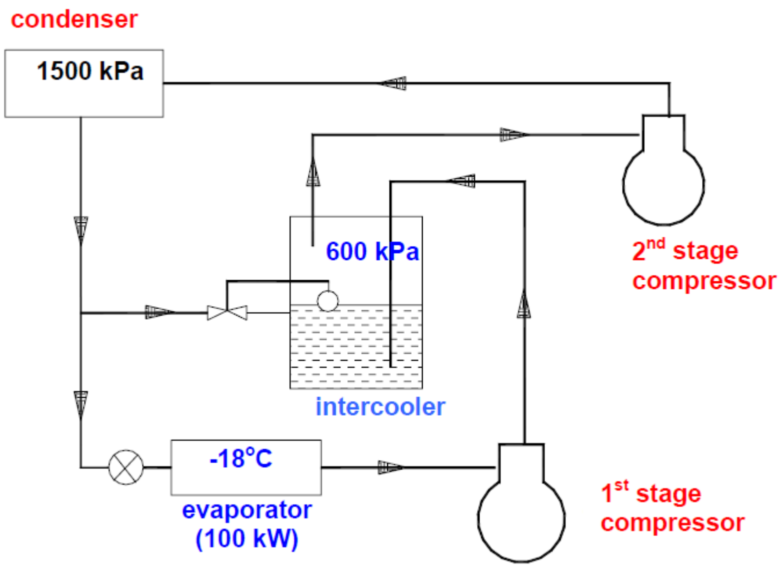


***Fig.12.8**: A two-stage compression system with flash tank for flash gas removal only*
(a) System schematic; (b) Cycle on P-h diagram

10. The required refrigeration capacity of a vapour compression refrigeration system (with R-22 as refrigerant) is 100 kW at –30°C evaporator temperature. Initially the system was single-stage with a single compressor compressing the refrigerant vapour from evaporator to a condenser operating at 1500 kPa pressure. Later the system was modified to a two-stage system operating on the cycle shown below. At the intermediate pressure of 600 kPa there is intercooling but no removal of flash gas. Find a) Power requirement of the original single-stage system; b) Total power requirement of the two compressors in the revised two-stage system. Assume that the state of refrigerant at the exit of evaporator, condenser and intercooler is saturated, and the compression processes are isentropic.



Ans.:

From refrigerant property data, the following values are obtained for R 22:

Point	Temp.,°C	Pressure,kPa	Dryness fraction	Density,kg/m^3	Enthalpy, kJ/kg	Entropy, kJ/kg.K
1	-30	163.9	1.0	7.379	392.7	1.802
3	39.1	1500	0.0	-	248.4	-
2	76.93	1500	-	-	449.9	1.802
2’’’	53.55	1500	-	-	429.6	1.742
2’’	5.86	600	1.0	-	407.2	1.742
2’	28.94	600	-	-	424.4	1.802

Power input to 1st stage compressor, $\mathbf{W_{c,1}}$ is given by:

$$W_{c,1} = m_{r,1}(h_{2'} - h_1) = 0.693(424.4 - 392.7) = \mathbf{21.97\ kW}$$

The mass flow rate of refrigerant vapour through 2nd stage compressor ($\mathbf{m_{r,2}}$) is obtained from energy balance across intercooler:

$$m_{r,2}.h_{2''} = m_{r,1}.h_{2'} + (m_{r,2} - m_{r,1}).h_{4'}$$

Substituting the values of enthalpy and mass flow rate through 1st stage compressor:

$$m_{r,2} = \mathbf{0.768\ kg/s}$$

Power input to 2nd stage compressor, $\mathbf{W_{c,2}}$ is given by:

$$W_{c,2} = m_{r,2}(h_{2'''} - h_{2''}) = 0.768(429.6 - 407.2) = \mathbf{17.2\ kW}$$

Therefore, total power input, Wc is given by:

$$W_c = W_{c,1}+W_{c,2} = 21.97+17.2 = \mathbf{39.17\ kW}$$

COP of the two-stage system is given by:

$$COP = Q_e/(W_{c,1}+W_{c,2}) = 100/39.17 = \mathbf{2.553}$$

From property data, the discharge temperatures at the exit of 1st and 2nd stage compressors are given, respectively by:

$$\mathbf{T2' = 28.94^{o}C}$$
$$\mathbf{T2''' = 53.55^{o}C}$$

Comments:

It is observed from the above example that for the given input data, though the use of a two-stage system with intercooling in place of a single stage system does not increase the COP significantly (**≈ 1.2 %**), there is a significant reduction in the maximum compressor discharge temperature (**≈ 24°C).** The results would be different if the operating conditions and/or the refrigerant used is different.

Vapour Absorption Refrigeration Systems

Introduction

Vapour Absorption Refrigeration Systems (VARS) belong to the class of vapour cycles similar to vapour compression refrigeration systems. However, unlike vapour compression refrigeration systems, the required input to absorption systems is in the form of heat. Hence these systems are also called as heat operated or thermal energy driven systems. Since conventional absorption systems use liquids for absorption of refrigerant, these are also sometimes called as wet absorption systems.

Similar to vapour compression refrigeration systems, vapour absorption refrigeration systems have also been commercialized and are widely used in various refrigeration and air conditioning applications. Since these systems run on low-grade thermal energy, they are preferred when low-grade energy such as waste heat or solar energy is available. Since conventional absorption systems use natural refrigerants such as water or ammonia they are environment friendly.

Basic principle

In this lesson, the basic working principle of absorption systems, the maximum COP of ideal absorption refrigeration systems, basics of properties of mixtures and simple absorption refrigeration systems will be discussed.

When a solute such as lithium bromide salt is dissolved in a solvent such as water, the boiling point of the solvent (water) is elevated. On the other hand, if the temperature of the solution (solvent + solute) is held constant, then the effect of dissolving the solute is to reduce the vapour pressure of the solvent below that of the saturation pressure of pure solvent at that temperature. If the solute itself has some vapour pressure (i.e., volatile solute) then the total pressure exerted over the solution is the sum total of the partial pressures of solute and solvent. If the solute is non-volatile (e.g. lithium bromide salt) or if the boiling point difference between the solution and solvent is large ($\geq 300_{o}$C), then the total pressure exerted over the solution will be almost equal to the vapour pressure of the solvent only. In the simplest absorption refrigeration system, refrigeration is obtained by connecting two vessels, with one vessel containing pure solvent and the other containing a solution.

Since the pressure is almost equal in both the vessels at equilibrium, the temperature of the solution will be higher than that of the pure solvent. This means that if the solution is at ambient temperature, then the pure solvent will be at a temperature lower than the ambient. Hence refrigeration effect is produced at the vessel containing pure solvent due to this temperature difference. The solvent evaporates due to heat transfer from the surroundings, flows to the vessel containing solution and is absorbed by the solution. This process is continued as long as the composition and temperature of the solution are maintained and liquid solvent is available in the container.

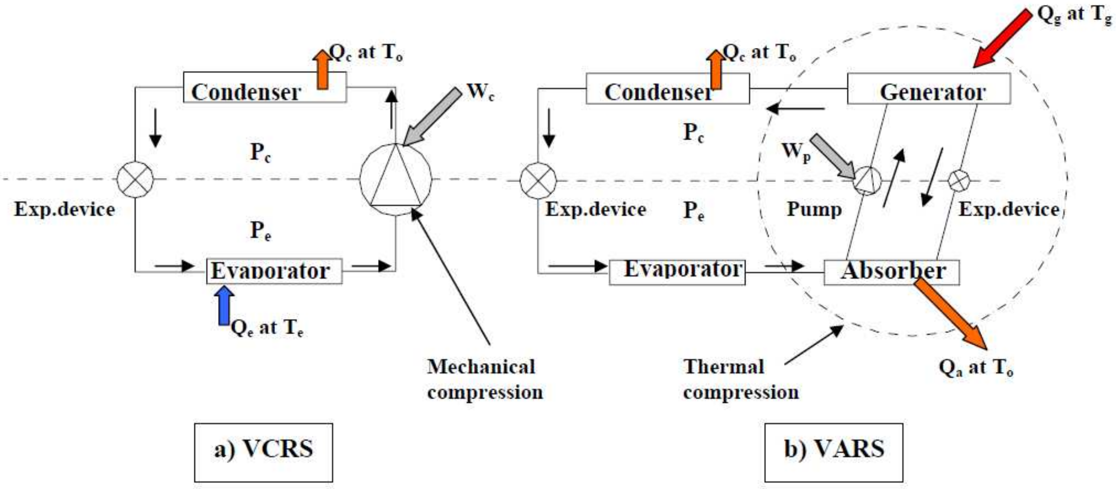
For example, Fig.14.1 shows an arrangement, which consists of two vessels A and B connected to each other through a connecting pipe and a valve. Vessel A is filled with pure water, while vessel B is filled with a solution containing on mass basis 50 percent of water and 50 percent lithium bromide (LiBr salt). Initially the valve connecting these two vessels is closed, and both vessels are at thermal equilibrium with the surroundings, which is at 30$_o$C. At 30$_o$C, the saturation pressure of water is 4.24 kPa, and the equilibrium vapour pressure of water-lithium bromide solution (50 : 50 by mass) at 30$_o$C is 1.22 kPa.

Thus at initial equilibrium condition, the pressure in vessel A is 4.24 kPa, while it is 1.22 kPa in vessel B. Now the valve between vessels A and B is opened. Initially due to pressure difference water vapour will flow from vessel A to vessel B, and this vapour will be absorbed by the solution in vessel B. Since absorption in this case is exothermic, heat will be released in vessel B. Now suppose by some means the concentration and temperature of vessel B are maintained constant at 50 % and 30$_o$C, respectively. Then at equilibrium, the pressure in the entire system (vessels A and B) will be 1.22 kPa (equilibrium pressure of 50 % LiBr solution at 30$_o$C). The temperature of water in vessel A will be the saturation temperature corresponding to 1.22 kPa, which is equal to about 10$_o$C, as shown in the figure. Since the water temperature in A is lower than the surroundings, a refrigeration effect (Q_e) can produced by transferring heat from the surroundings to water at 10$_o$C. Due to this heat transfer, water vaporizes in A, flows to B and is absorbed by the solution in B. The exothermic heat of absorption (Q_a) is rejected to the surroundings.

Now for the above process to continue, there should always be pure water in vessel A, and vessel B must be maintained always at 50 percent concentration and 30$_o$C. This is not possible in a closed system such as the one shown in Fig.14.1. In a closed system with finite sized reservoirs, gradually the amount of water in A decreases and the solution in B becomes diluted with water. As a result, the system pressure and temperature of water in A increase with time. Hence the refrigeration effect at A reduces gradually due to the reduced temperature difference between the surroundings and water. Thus refrigeration produced by systems using only two vessels is intermittent in nature. In these systems, after a period, the refrigeration process has to be stopped and both the vessels A and B have to be brought back to their original condition. This requires removal of water absorbed in B and adding it back to vessel A in liquid form, i.e., a process of regeneration as shown in Fig.14.1(c). Assume that before regeneration is carried out, the valve between A and B is closed and both A and B are brought in thermal equilibrium with the surroundings (30$_o$C), then during the regeneration process, heat at high temperature T_g is supplied to the dilute LiBr solution in B, as a result water vapour is generated in B. The vapour generated in B is condensed into pure water in A by rejecting heat of condensation to the surroundings. This process has to be continued till all the water absorbed during the refrigeration process (14.1(b)) is transferred back to A. Then to bring the system back to its original condition, the valve has to be closed and solution in vessel B has to be cooled to 30$_o$C. If we assume a steady-flow process of regeneration and neglect temperature difference for heat transfer, then the temperature of water in A will be 30$_o$C and pressure inside the system will be 4.24 kPa. Then the temperature in vessel

B, T_g depends on the concentration of solution in B. The amount of heat transferred during refrigeration and regeneration depends on the properties of solution and the operating conditions. It can be seen that the output from this system is the refrigeration obtained Q_e and the input is heat supplied to vessel B during vapour regeneration process, Q_g.

The system described may be called as an Intermittent Absorption Refrigeration System. The solvent is the refrigerant and the solute is called as absorbent. These simple systems can be used to provide refrigeration using renewable energy such as solar energy in remote and rural areas. As already explained, these systems provided refrigeration intermittently, if solar energy is used for regenerating the refrigerant, then regeneration process can be carried out during the day and refrigeration can be produced during the night.
Though the intermittent absorption refrigeration systems discussed above are simple in design and inexpensive, they are not useful in applications that require continuous refrigeration. Continuous refrigeration can be obtained by having a modified system with two pairs of vessels A and B and additional expansion valves and a solution pump.



a) VCRS

b) VARS

Figs.14.2: a) Vapour compression refrigeration system (VCRS)
b) Vapour Absorption Refrigeration System (VARS)

Figure 14.2(a) and (b) show a continuous output vapour compression refrigeration system and a continuous output vapour absorption refrigeration system. As shown in the figure in a continuous absorption system, low temperature and low pressure refrigerant with low quality enters the evaporator and vaporizes by producing useful refrigeration Q_e. From the evaporator, the low temperature, low pressure refrigerant vapour enters the absorber where it comes in contact with a solution that is weak in refrigerant. The weak solution absorbs the refrigerant and becomes strong in refrigerant. The heat of absorption is rejected to the external heat sink at T_o. The solution that is now rich in refrigerant is pumped to high pressure using a solution pump and fed to the generator. In the generator heat at high temperature T_g is supplied, as a result refrigerant vapour is generated at high pressure. This high pressure vapour is then condensed in the condenser by rejecting heat of condensation to the external heat sink at T_o. The condensed refrigerant liquid is then throttled in the expansion device and is then fed to the evaporator to complete the refrigerant cycle. On the solution side, the hot, high-pressure solution that is weak in refrigerant is throttled to the absorber pressure in the solution expansion valve and fed to the absorber where it comes in contact with the refrigerant vapour from evaporator. Thus continuous refrigeration is produced at evaporator, while heat at high temperature is continuously supplied to the generator. Heat rejection to the external heat sink takes place at absorber and condenser. A small amount of mechanical energy is required to run the solution pump. If we neglect pressure drops, then the absorption system operates between the condenser and evaporator pressures. Pressure in absorber is same as the pressure in evaporator and pressure in generator is same as the pressure in condenser.

It can be seen from Fig.14.2, that as far as the condenser, expansion valve and evaporators are concerned both compression and absorption systems are identical. However, the difference lies in the way the refrigerant is compressed to condenser pressure. In vapour compression refrigeration systems the vapour is compressed mechanically using the compressor, where as in absorption system the vapour is first converted into a liquid and then the liquid is pumped to condenser pressure using the solution pump. Since for the same pressure difference, work input required to pump a liquid (solution) is much less than the work required for compressing a vapour due to very small specific volume of liquid ($w = -\int_{Pe}^{Pc} v.dP$), the mechanical energy required to operate vapour absorption refrigeration system is much less than that required to operate a compression system. However, the absorption system requires a relatively large amount of low-grade thermal energy at generator temperature to generate refrigerant vapour from the solution in generator. Thus while the energy input is in the form of mechanical energy in vapour compression refrigeration systems, it is mainly in the form of thermal energy in case of absorption systems. The solution pump work is often negligible compared to the generator heat input. Thus the COPs for compression and absorption systems are given by:

$$COP_{VCRS} = \frac{Q_e}{W_c} \tag{14.1}$$

$$COP_{VARS} = \frac{Q_e}{Q_g + W_p} \approx \frac{Q_e}{Q_g} \tag{14.2}$$

Thus absorption systems are advantageous where a large quantity of low-grade thermal energy is available freely at required temperature. However, it will be seen that for the refrigeration and heat rejection temperatures, the COP of vapour compression refrigeration system will be much higher than the COP of an absorption system as a high grade mechanical energy is used in the former, while a low-grade thermal energy is used in the latter. However, comparing these systems based on COPs is not fully justified, as mechanical energy is more expensive than thermal energy. Hence, sometimes the second law (or exergetic) efficiency is used to compare different refrigeration systems. It is seen that the second law (or exergetic) efficiency of absorption system is of the same order as that of a compression system.

14.3. Maximum COP of ideal absorption refrigeration system

In case of a single stage compression refrigeration system operating between constant evaporator and condenser temperatures, the maximum possible COP is given by Carnot COP:

$$COP_{Carnot} = \frac{T_e}{T_c - T_e} \tag{14.3}$$

If we assume that heat rejection at the absorber and condenser takes place at same external heat sink temperature T_o, then a vapour absorption refrigeration system operates between three temperature levels, T_g, T_o and T_e. The maximum possible COP of a refrigeration system operating between three temperature levels can be obtained by applying first and second laws of thermodynamics to the system. Figure 14.3 shows the various energy transfers and the corresponding temperatures in an absorption refrigeration system.

From first law of thermodynamics,

$$Q_e + Q_g - Q_{c+a} + W_p = 0 \tag{14.4}$$

where Q_e is the heat transferred to the absorption system at evaporator temperature T_e, Q_g is the heat transferred to the generator of the absorption system at temperature T_g, Q_{a+c} is the heat transferred from the absorber and condenser of the absorption system at temperature T_o and W_p is the work input to the solution pump.

From second law of thermodynamics,

$$\Delta S_{total} = \Delta S_{sys} + \Delta S_{surr} \geq 0 \tag{14.5}$$

where ΔS_{total} is the total entropy change which is equal to the sum of entropy change of the system ΔS_{sys} and entropy change of the surroundings ΔS_{surr}. Since the refrigeration system operates in a closed cycle, the entropy change of the working fluid of the system undergoing the cycle is zero, i.e., $\Delta S_{sys} = 0$. The entropy change of the surroundings is given by:

$$\Delta S_{surr} = -\frac{Q_e}{T_e} - \frac{Q_g}{T_g} + \frac{Q_{a+c}}{T_o} \geq 0 \tag{14.6}$$

Substituting the expression for first law of thermodynamics in the above equation

$$Q_g\left(\frac{T_g - T_o}{T_g}\right) \geq Q_e\left(\frac{T_o - T_e}{T_e}\right) - W_p \tag{14.7}$$

Neglecting solution pump work, W_p; the COP of VARS is given by:

$$COP_{VARS} = \frac{Q_e}{Q_g} \leq \left(\frac{T_e}{T_o - T_e}\right)\left(\frac{T_g - T_o}{T_g}\right) \tag{14.8}$$

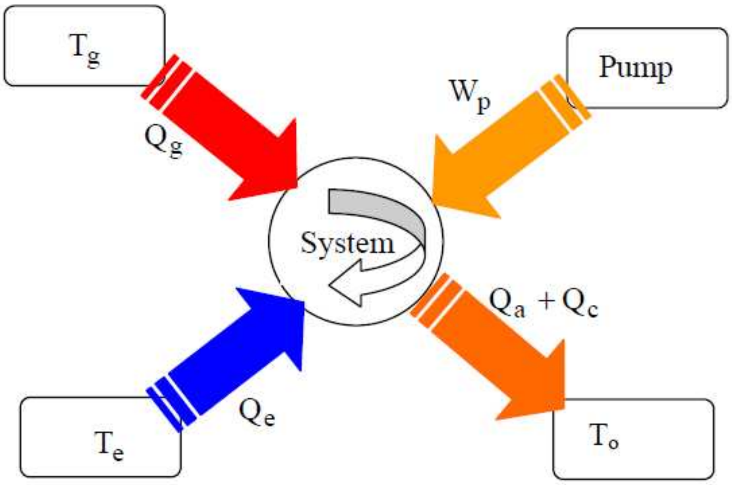


***Fig.14.3**: Various energy transfers in a vapour absorption refrigeration system*

14.4. Properties of refrigerant-absorbent mixtures

The solution used in absorption refrigeration systems may be considered as a homogeneous binary mixture of refrigerant and absorbent. Depending upon the boiling point difference between refrigerant and absorbent and the operating temperatures, one may encounter a pure refrigerant vapour or a mixture of refrigerant and absorbent vapour in generator of the absorption system. Unlike pure substances, the thermodynamic state of a binary mixture (in liquid or vapour phase) cannot be fixed by pressure and temperature alone. According to Gibbs' phase rule, one more parameter in addition to temperature and pressure is required to completely fix the thermodynamic state. Generally, the composition of the mixture is taken as the third independent parameter. The composition of a mixture can be expressed either in mass fraction or in mole fraction. The mass fraction of components 1 and 2 in a binary mixture are given by:

$$\xi_1 = \frac{m_1}{m_1 + m_2}; \quad \xi_2 = \frac{m_2}{m_1 + m_2} \tag{14.12}$$

where m_1 and m_2 are the mass of components 1 and 2, respectively

The mole fraction of components 1 and 2 in a binary mixture are given by:

$$x_1 = \frac{n_1}{n_1 + n_2}; \quad x_2 = \frac{n_2}{n_1 + n_2} \tag{14.13}$$

where n_1 and n_2 are the number of moles of components 1 and 2, respectively

An important property of a mixture is its miscibility. A mixture is said to be completely miscible if a homogeneous mixture can be formed through any arbitrary range of concentration values. Miscibility of mixtures is influenced by the temperature at which they are mixed. Some mixtures are miscible under certain conditions and immiscible at other conditions. The refrigerant-absorbent mixtures used in absorption refrigeration systems must be completely miscible under all conditions both in liquid and vapour phases.

14.4.1. Ideal, homogeneous binary mixtures

A binary mixture of components 1 and 2 is called as an ideal mixture, when it satisfies the following conditions.

Condition 1: The volume of the mixture is equal to the sum of the volumes of its constituents, i.e., upon mixing there is neither contraction nor expansion. Thus the specific volume of the mixture, v is given by:

$$v = \xi_1 . v_1 + \xi_2 . v_2 \tag{14.14}$$

where ξ_1 and ξ_2 are the mass fractions of components 1 and 2. For a binary mixture, ξ_1 and ξ_2 are related by:

$$\xi_1 + \xi_2 = 1 \Rightarrow \xi_2 = 1 - \xi_1 \tag{14.15}$$

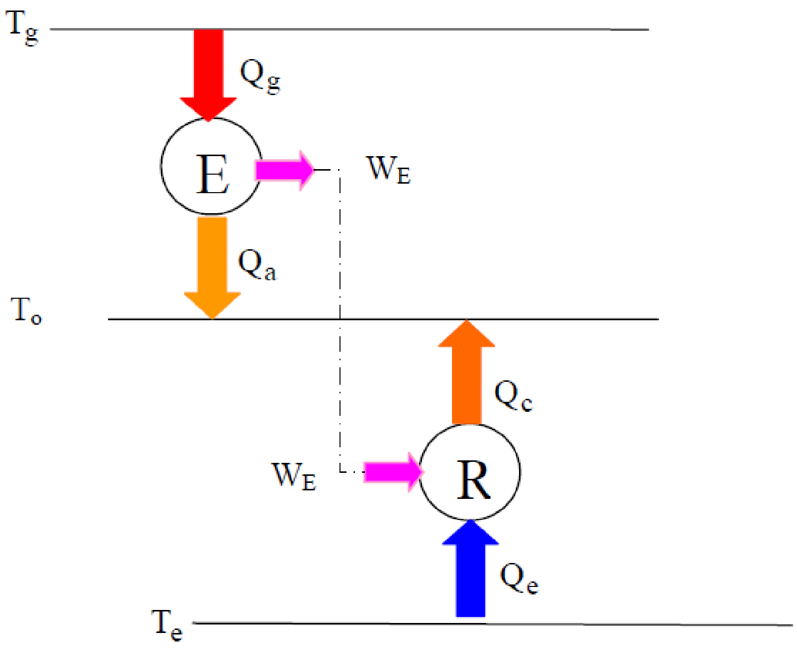


Fig.14.4: *Vapour absorption refrigeration system as a combination of a heat engine and a refrigerator*

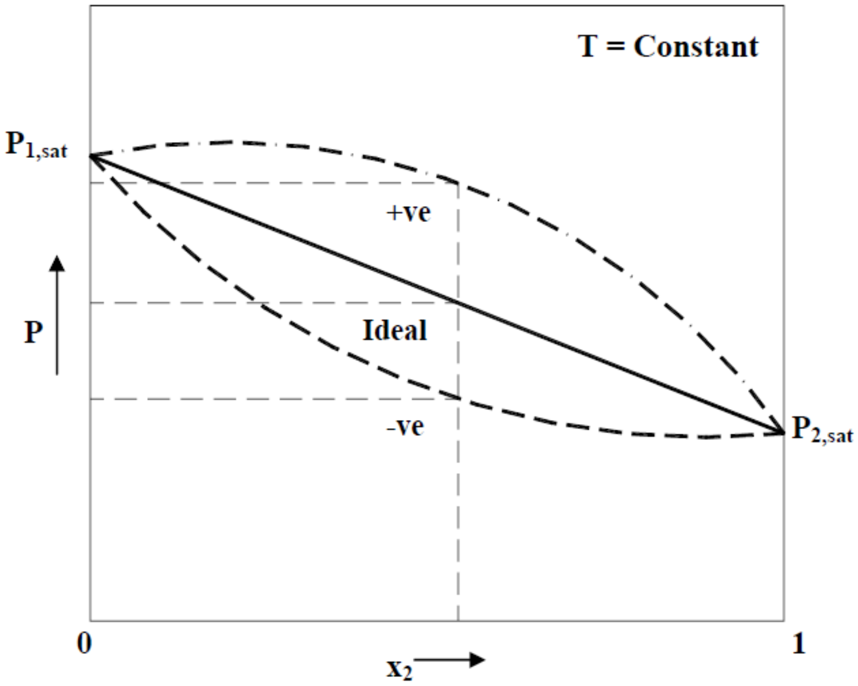


Fig.14.5: *Pressure-concentration behaviour of ideal and real mixtures at a constant temperature*

Condition 2: Neither heat is generated nor absorbed upon mixing, i.e., the heat of solution is zero. Then the specific enthalpy of the mixture, h is given by:

$$h = \xi_1 . h_1 + \xi_2 . h_2 = \xi_1 . h_1 + (1 - \xi_1) h_2 \tag{14.16}$$

Condition 3: The mixture obeys Raoult's law in liquid phase, i.e., the vapour pressure exerted by components 1 and 2 ($P_{v,1}$ and $P_{v,2}$) at a temperature T are given by:

$$P_{v,1} = x_1 . P_{1,sat} \tag{14.17}$$

$$P_{v,2} = x_2 . P_{2,sat} \tag{14.18}$$

where x_1 and x_2 are the mole fractions of components 1 and 2 in solution, and $P_{1,sat}$ and $P_{2,\ sat}$ are the saturation pressures of pure components 1 and 2 at temperature T. The mole fractions x_1 and x_2 are related by:

$$x_1 + x_2 = 1 \Rightarrow x_2 = 1 - x_1 \tag{14.19}$$

Condition 4: The mixture obeys Dalton's law in vapour phase; i.e., the vapour pressure exerted by components 1 and 2 ($P_{v,1}$ and $P_{v,2}$) in vapour phase at a temperature T are given by:

$$P_{v,1} = y_1 . P_{total} \tag{14.20}$$

$$P_{v,2} = y_2 . P_{total} \tag{14.21}$$

where y_1 and y_2 are the vapour phase mole fractions of components 1 and 2 and P_{total} is the total pressure exerted at temperature T. The vapour phase mole fractions y_1 and y_2 are related by:

$$y_1 + y_2 = 1 \Rightarrow y_2 = 1 - y_1 \tag{14.22}$$

and the total pressure P_{total} is given by:

$$P_{total} = P_{v,1} + P_{v,2} \tag{14.23}$$

If one of the components, say component 2 is non-volatile compared to component 1(e.g. component 1 is water and component 2 is lithium bromide salt), then $y_1 \approx 1$ and $y_2 \approx 0$, $P_{v,2} \approx 0$, then from Raoult's and to Dalton's laws:

$$P_{total} \approx P_{v,1} = x_1 . P_{1,sat} \tag{14.24}$$

14.4.2. Real mixtures

Real mixtures deviate from ideal mixtures since:

1. A real solution either contracts or expands upon mixing, i.e.,

$$v \neq \xi_1 . v_1 + \xi_2 . v_2 \tag{14.25}$$

2. Either heat is evolved (exothermic) or heat is absorbed upon mixing;

$$h = \xi_1 . h_1 + (1 - \xi_1) h_2 + \Delta h_{mix} \tag{14.26}$$

where Δh_{mix} is the heat of mixing, which is taken as negative when heat is evolved and positive when heat is absorbed.

The above two differences between ideal and real mixtures can be attributed to the deviation of real mixtures from Raoult's law. Real mixtures approach ideal mixtures as the mole fraction of the component contributing to vapour pressure approaches unity, i.e., for very dilute solutions. Figure 14.5 shows the equilibrium pressure variation with liquid phase mole fraction (x) of ideal and real binary mixtures with positive (+ve) and negative deviations (-ve) from Raoult's law at a constant temperature. It can be seen that when the deviation from Raoult's law is positive (+ve), the equilibrium vapour pressure will be higher than that predicted by Raoult's law, consequently at a given pressure and composition, the equilibrium temperature of solution will be lower than that predicted by Raoult's law. The converse is true for solutions with –ve deviation from Raoult's law, i.e., the equilibrium temperature at a given pressure and composition will be higher than that predicted by Raoult's law for solution with negative deviation. This behaviour can also be shown on specific enthalpy-composition diagram as shown in Fig. 14.6 for a solution with negative deviation from Raoult's law. Refrigerant-absorbent mixtures used in vapour absorption refrigeration systems exhibit a negative deviation from Raoult's law, i.e., the process of absorption is exothermic with a negative heat of mixing.

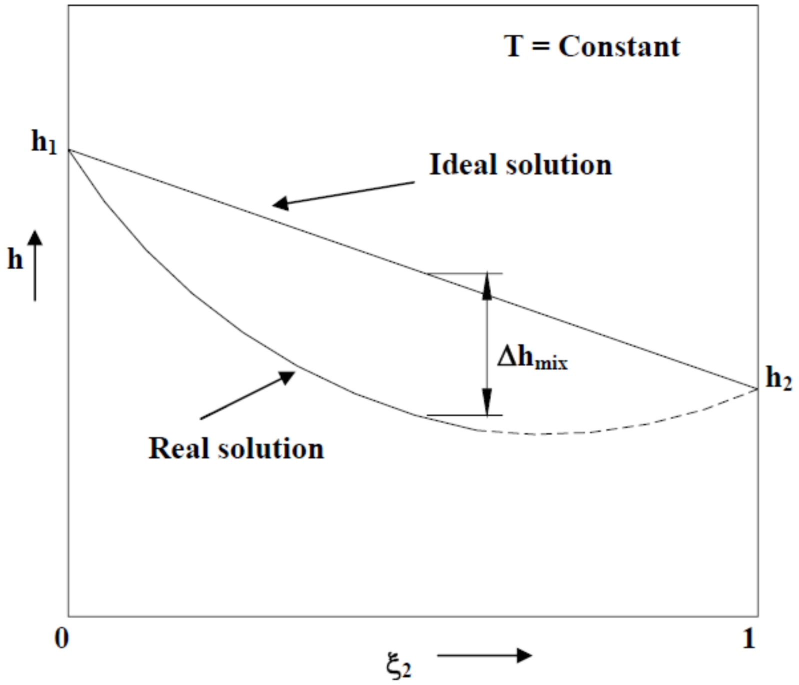


Fig.14.6: *Enthalpy-concentration behaviour of an ideal mixture and a real mixture with negative deviation from Raoult's law*

result of this heat exchange, less heat input is required in the generator and less heat is rejected in the absorber, thus improving the system performance significantly.

Vapour Absorption Refrigeration Systems Based On Water-Lithium Bromide Pair

15.1. Introduction

Vapour absorption refrigeration systems using water-lithium bromide pair are extensively used in large capacity air conditioning systems. In these systems water is used as refrigerant and a solution of lithium bromide in water is used as absorbent. Since water is used as refrigerant, using these systems it is not possible to provide refrigeration at sub-zero temperatures. Hence it is used only in applications requiring refrigeration at temperatures above 0°C. Hence these systems are used for air conditioning applications. The analysis of this system is relatively easy as the vapour generated in the generator is almost pure refrigerant (water), unlike ammonia-water systems where both ammonia and water vapour are generated in the generator.

The vapour pressure data of water-lithium bromide solutions can be very conveniently represented in a Dühring plot. In a Dühring plot, the temperature of the solution is plotted as abscissa on a linear scale, the saturation temperature of pure water is plotted as ordinate on the right hand side (linear scale) and the pressure on a logarithmic scale is plotted as ordinate on the left hand side. The plot shows the pressure-temperature values for various constant concentration lines (isosters), which are linear on Dühring plot. Figures 15.1 shows the Dühring plot. The Dühring plot can be used for finding the vapour pressure data and also for plotting the operating cycle. Figure 15.2 shows the water-lithium bromide based absorption refrigeration system on Dühring plot. Other types of charts showing vapour pressure data for water-lithum bromide systems are also available in literature. Figure 15.3 shows another chart wherein the mass fraction of lithium bromide is plotted on abscissa, while saturation temperature of pure water and vapour pressure are plotted as ordinates. Also shown are lines of constant solution temperature on the chart. Pressure-temperature-composition data are also available in the form of empirical equations.

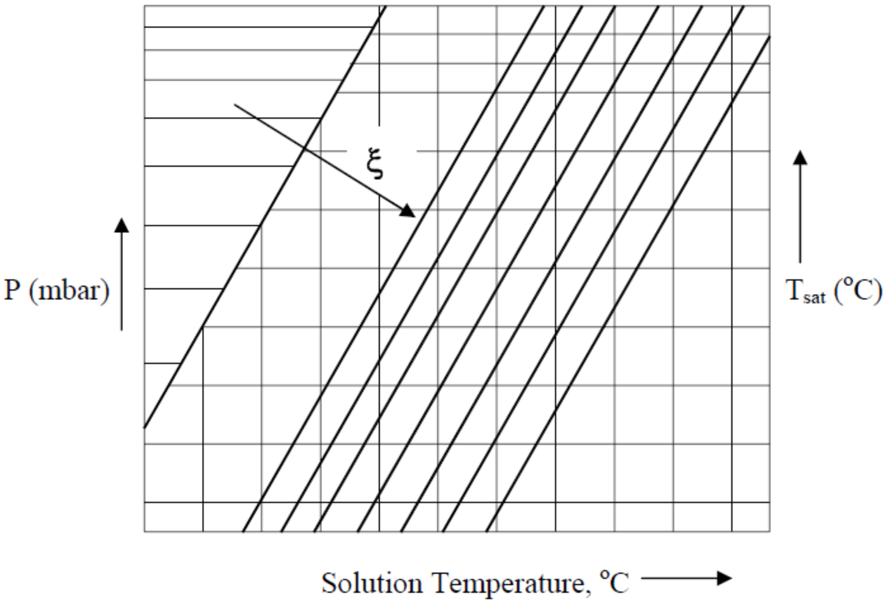


***Fig.15.1**.: A typical Dühring plot*

1. Regulating the flow rate of weak solution pumped to the generator through the solution pump
2. Reducing the generator temperature by throttling the supply steam, or by reducing the flow rate of hot water
3. Increasing the condenser temperature by bypassing some of the cooling water supplied to the condenser

Method 1 does not affect the COP significantly as the required heat input reduces with reduction in weak solution flow rate, however, since this may lead to the problem of crystallization, many a time a combination of the above three methods are used in commercial systems to control the capacity.

Generator: From energy balance:

$Q_g = mh_1+m_{SS}h_8-m_{WS}h_7 = 380.54$ kW **(Ans.)**

Condenser: From energy balance:

$Q_c = m(h_1-h_2) = 325.9$ kW **(Ans.)**

Solution heat exchanger: From energy balance:

$Q_{SHX} = m\lambda(h_8-h_9) = m(\lambda+1)(h_7-h_6) = 122.3$ kW **(Ans.)**

c) System COP (neglecting pump work) = **$Q_e/Q_g = 0.7884$** **(Ans.)**

Second law efficiency = COP/COP_{Carnot}

$COP_{Carnot} = [T_e/(T_c-T_e)][(T_g-T_a)/T_g] = \mathbf{1.129}$

$\therefore$ Second law efficiency = 0.6983 **(Ans.)**

d) Solution pump work (assuming the solution to be incompressible)

$W_P = v_{sol}(P_6-P_5) = (P_6-P_5)/\rho_{sol} = (123.3-8.72)*10^{-1}/1200 = 0.0095$ kW **(Ans.)**

iii. It should exhibit small heat of mixing so that a high COP can be achieved. However, this requirement contradicts the first requirement. Hence, in practice a trade-off is required between solubility and heat of mixing.
iv. The refrigerant-absorbent mixture should have high thermal conductivity and low viscosity for high performance.
v. It should not undergo crystallization or solidification inside the system.
vi. The mixture should be safe, chemically stable, non-corrosive, inexpensive and should be available easily.

The most commonly used refrigerant-absorbent pairs in commercial systems are:

1. Water-Lithium Bromide (H_2O-LiBr) system for above 0°C applications such as air conditioning. Here water is the refrigerant and lithium bromide is the absorbent.
2. Ammonia-Water (NH_3-H_2O) system for refrigeration applications with ammonia as refrigerant and water as absorbent.

Of late efforts are being made to develop other refrigerant-absorbent systems using both natural and synthetic refrigerants to overcome some of the limitations of (H_2O-LiBr) and (NH_3-H_2O) systems.

Currently, large water-lithium bromide (H_2O-LiBr) systems are extensively used in air conditioning applications, where as large ammonia-water (NH_3-H_2O) systems are used in refrigeration applications, while small ammonia-water systems with a third inert gas are used in a pumpless form in small domestic refrigerators (triple fluid vapour absorption systems).

9. The operating temperatures of a single stage vapour absorption refrigeration system are: generator: **90°C**; condenser and absorber: **40°C**; evaporator: **0°C**. The system has a refrigeration capacity of **100 kW** and the heat input to the system is **160 kW**. The solution pump work is negligible.

a) Find the COP of the system and the total heat rejection rate from the system.

b) An inventor claims that by improving the design of all the components of the system he could reduce the heat input to the system to **80 kW** while keeping the refrigeration capacity and operating temperatures same as before. Examine the validity of the claim.

Ans.:

a) $\mathrm{COP} = Q_e/Q_g = 100/160 = \mathbf{0.625}$ **(Ans.)**

Total heat rejection rate = $Q_a + Q_c = Q_e + Q_g = 100 + 160 =$ **260 kW** **(Ans.)**

b) According to the inventor's claim, the $\mathrm{COP_{claim}}$ is given by:

$$\mathrm{COP_{claim}} = Q_e/Q_g = 100/80 = \mathbf{1.25}$$

However, for the given temperatures, the maximum possible COP is given by:

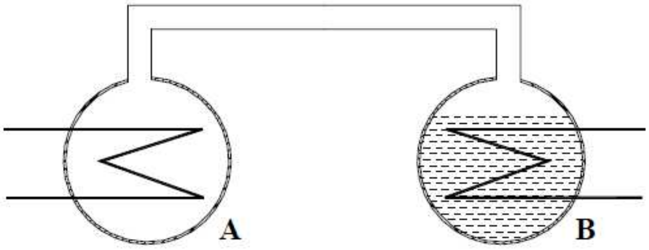
$$\mathrm{COP_{ideal\ VARS}} = \left(\frac{Q_e}{Q_g}\right)_{max} = \left(\frac{T_e}{T_o - T_e}\right)\left(\frac{T_g - T_o}{T_g}\right)$$

Substituting the values of operating temperatures, we find that:

$$\mathrm{COP_{max}} = \left(\frac{T_e}{T_o - T_e}\right)\left(\frac{T_g - T_o}{T_g}\right) = \left(\frac{273}{313 - 273}\right)\left(\frac{50}{363}\right) = 0.94$$

Since $\mathrm{COP_{claim}} > \mathrm{COP_{max}} \Rightarrow$ Inventor's claim is FALSE **(Ans.)**

1. The following figure shows a pair of containers A & B. Container B contains an aqueous solution of (LiBr+H_2O) at a mass fraction (x_i) of 0.6. Container A and connecting pipe are filled with pure water vapor. Initially the system (A+B) is at an equilibrium temperature of 90°C, at which the pressure is found to be 9.0 kPa. Now water vapour starts condensing in A as cooling water starts flowing through the coil kept in A.



a) What is the temperature of the coil at which steam starts condensing in A?
b) Does the System pressure remain constant during condensation? If not, how to maintain the pressure constant at 9.0 kPa? What happens to the temperature of solution in B?
c) As water vapour condenses in A there will be transfer of water vapour from B to A resulting in change of mass fraction of solution (Δx) in B. Find a relation between Δx and *f*, where *f* is the ratio of initial mass of solution in B to the mass of water vapour transferred from B to A.
d) What is the amount of solution required initially in B so that a mass of 1 kg of water is transferred from B to A with a corresponding change of mass fraction(Δx) by 0.05?
e) Neglecting the contribution of temperature changes, what is the amount of heat transferred at A and B during the transfer of 1 kg of water from B to A? Is energy balanced?
f) What is required to reverse the process so that initial conditions are restored?
g) Show the forward and reverse process on D ring plot.

Use the following data:
Initial enthalpy of solution = 220 kJ/kg; Final enthalpy of solution = 270 kJ/kg
Assume that the average latent heat of vaporization of water and enthalpy of water vapour = 2500 kJ/kg
Saturation pressure of water vapour (in kPa) is given by the Antoine's equation:

$$\ln(p_{sat}) = c_o - \frac{c_1}{T + c_2}$$; where T is temperature in K, c_o=16.54, c_1=3985, c_2=-39.0

Ans.:

a) Steam in vessel A starts condensing when the surface temperature of the coil falls below the saturation temperature of water at 9.0 kPa. Using Antoine's equation:

$$\ln(9) = 16.54 - \frac{3985}{T - 39} \Rightarrow T = 316.84\,K = 43.7^{o}\,C \qquad \textbf{(Ans.)}$$

b) System **pressure falls** as condensation of water vapour takes place in A. To keep the system pressure constant, vapour has to be generated in B by **supplying heat to solution in B**. Since the solution in B becomes richer in LiBr (i.e., concentration increases), at the same pressure of 9.0 kPa, the **solution temperature in B increases**. **(Ans.)**

c) From the definition of concentration for H_2O-LiBr solution;

$$\Delta x = x_f - x_i = \left(\frac{M_L}{M_L + M_{W,f}}\right) - \left(\frac{M_L}{M_L + M_{W,i}}\right) = M_L\left[\frac{(M_{W,i} - M_{W,f})}{(M_L + M_{W,i}).(M_L + M_{W,i})}\right]$$

Amount of water transferred from B to A = $\mathbf{(M_{W,i} - M_{W,f})}$

The factor f is defined as:

$$f = \left(\frac{M_L + M_{W,i}}{M_{W,i} - M_{W,f}}\right)$$

Substituting the above in the expression for Δx and using the definition of concentration, we find that:

$$\Delta x = x_f - x_i = \left(\frac{x_f}{f}\right) \qquad \textbf{(Ans.)}$$

d) Mass of water transferred is 1.0 kg and change in concentration is 0.05. Hence the final concentration is:

$$\mathbf{x_f = x_i + 0.05 = 0.60 + 0.05 = 0.65}$$

Substituting this value in the expression for Δx, we find that

$$f = \left(\frac{x_f}{\Delta x}\right) = \left(\frac{0.65}{0.05}\right) = 13$$

Hence the initial mass of solution is given by:

$$(M_L + M_{W,i}) = f.(\text{mass of water transferred}) = 13\,X\,1.0 = 13\,\text{kgs} \qquad \textbf{(Ans.)}$$

e) From energy balance of vessel B, the amount of energy transferred to B is given by:

$$Q_{B,in} = (M_{B,f}.h_f - M_{B,i}.h_i) + (M_{W,i} - M_{W,f})h_W$$

Substituting the values of enthalpies and initial and final mass of solution (13 kg and 12 kg, respectively), we find that the heat transferred to B is:

$Q_{B,in}$ = 2880 kJ **(Ans.)**

Neglecting the heat transferred during initial sensible cooling of vapour, the total heat transferred at Vessel A is:

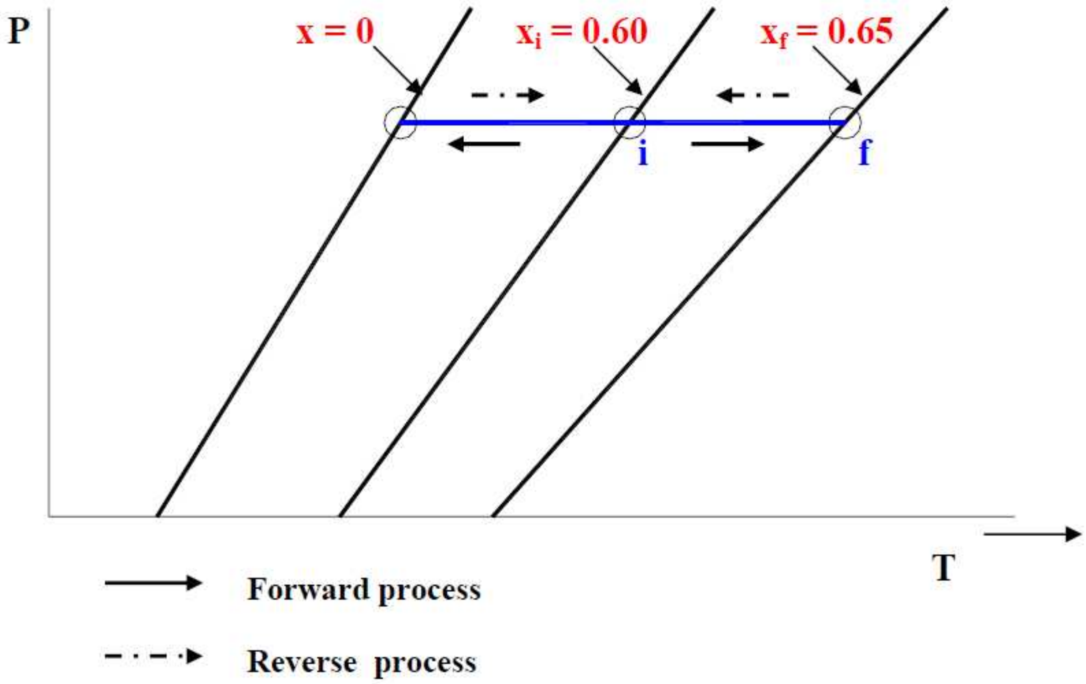
$Q_{A,out}$ = Amount of water vapour condensed X latent heat of vapourization = **2500 kJ**

(Ans.)

The difference in energy transferred at A and B is stored in the form of heat of solution. **(Ans.)**

f) To reverse the process and arrive at initial condition, the condensed water in vessel A has to be vapourized by supplying heat to vessel A. The vapour generated is absorbed by strong solution in B. Since this is an exothermic process, heat has to be rejected from B. **(Ans.)**

g) Dühring plot of forward and reverse processese is shown below:



MODULE-III
REFRIGERANTS

Introduction:

The thermodynamic efficiency of a refrigeration system depends mainly on its operating temperatures. However, important practical issues such as the system design, size, initial and operating costs, safety, reliability, and serviceability etc. depend very much on the type of refrigerant selected for a given application. Due to several environmental issues such as ozone layer depletion and global warming and their relation to the various refrigerants used, the selection of suitable refrigerant has become one of the most important issues in recent times. Replacement of an existing refrigerant by a completely new refrigerant, for whatever reason, is an expensive proposition as it may call for several changes in the design and manufacturing of refrigeration systems. Hence it is very important to understand the issues related to the selection and use of refrigerants. In principle, any fluid can be used as a refrigerant. Air used in an air cycle refrigeration system can also be considered as a refrigerant. However, in this lecture the attention is mainly focused on those fluids that can be used as refrigerants in vapour compression refrigeration systems only.

Primary and secondary refrigerants:

Fluids suitable for refrigeration purposes can be classified into primary and secondary refrigerants. Primary refrigerants are those fluids, which are used directly as working fluids, for example in vapour compression and vapour absorption refrigeration systems. When used in compression or absorption systems, these fluids provide refrigeration by undergoing a phase change process in the evaporator. As the name implies, secondary refrigerants are those liquids, which are used for transporting thermal energy from one location to other. Secondary refrigerants are also known under the name brines or antifreezes. Of course, if the operating temperatures are above 0_{o}C, then pure water can also be used as secondary refrigerant, for example in large air conditioning systems. Antifreezes or brines are used when refrigeration is required at sub-zero temperatures. Unlike primary refrigerants, the secondary refrigerants do not undergo phase change as they transport energy from one location to other. An important property of a secondary refrigerant is its freezing point. Generally, the freezing point of a brine will be lower than the freezing point of its constituents. The temperature at which freezing of a brine takes place its depends on its concentration. The concentration at which a lowest temperature can be reached without solidification is called as eutectic point. The commonly used secondary refrigerants are the solutions of water and ethylene glycol, propylene glycol or calcium chloride. These solutions are known under the general name of brines. In this lecture attention is focused on primary refrigerants used mainly in vapour compression refrigeration systems. As discussed earlier, in an absorption refrigeration system, a refrigerant and absorbent combination is used as the working fluid.

Refrigerant selection criteria:

Selection of refrigerant for a particular application is based on the following requirements:

i. Thermodynamic and thermo-physical properties
ii. Environmental and safety properties, and
iii. Economics

Thermodynamic and thermo-physical properties:

The requirements are:

a) Suction pressure: At a given evaporator temperature, the saturation pressure should be above atmospheric for prevention of air or moisture ingress into the system and ease of leak detection. Higher suction pressure is better as it leads to smaller compressor displacement

b) Discharge pressure: At a given condenser temperature, the discharge pressure should be as small as possible to allow light-weight construction of compressor, condenser etc.

c) Pressure ratio: Should be as small as possible for high volumetric efficiency and low power consumption

d) Latent heat of vaporization: Should be as large as possible so that the required mass flow rate per unit cooling capacity will be small

The above requirements are somewhat contradictory, as the operating pressures, temperatures and latent heat of vaporization are related by Clausius-Clapeyron Equation:

In the above equation, P_{sat} is the saturation pressure (in atm.) at a temperature T(in Kelvin), h_{fg} and s_{fg} are enthalpy and entropy of vaporization and R is the gas constant. Since the change in entropy of vaporization is relatively small, from the above equation it can be shown that:

In the above equation, P_c and P_e are the condenser and evaporator pressures, T_c and T_e are condenser and evaporator temperatures. From the above equation, it can be seen that for given condenser and evaporator temperatures as the latent heat of vaporization increases, the pressure ratio also increases. Hence a trade-off is required between the latent heat of vaporization and pressure ratio.

In addition to the above properties; the following properties are also important:

e) Isentropic index of compression: Should be as small as possible so that the temperature rise during compression will be small

f) Liquid specific heat: Should be small so that degree of subcooling will be large leading to smaller amount of flash gas at evaporator inlet

g) Vapour specific heat: Should be large so that the degree of superheating will be small

h) Thermal conductivity: Thermal conductivity in both liquid as well as vapour phase should be high for higher heat transfer coefficients

i) Viscosity: Viscosity should be small in both liquid and vapour phases for smaller frictional pressure drops

The thermodynamic properties are interrelated and mainly depend on normal boiling point, critical temperature, molecular weight and structure. The normal boiling point indicates the useful temperature levels as it is directly related to the operating pressures. A high critical temperature yields higher COP due to smaller compressor superheat and smaller flash gas losses. On the other hand since the vapour pressure will be low when critical temperature is high, the volumetric capacity will be lower for refrigerants with high critical temperatures. This once again shows a need for trade-off between high COP and high volumetric capacity. It is observed that for most of the refrigerants the ratio of normal boiling point to critical temperature is in the range of 0.6 to 0.7. Thus the normal boiling point is a good indicator of the critical temperature of the refrigerant.

The important properties such as latent heat of vaporization and specific heat depend on the molecular weight and structure of the molecule. Trouton's rule shows that the latent heat of vaporization will be high for refrigerants having lower molecular weight. The specific heat of refrigerant is related to the structure of the molecule. If specific heat of refrigerant vapour is low then the shape of the vapour dome will be such that the compression process starting with a saturated point terminates in the superheated zone (**i.e, compression process will be dry**). However, a small value of vapour specific heat indicates higher degree of superheat. Since vapour and liquid specific heats are also related, a large value of vapour specific heat results in a higher value of liquid specific heat, leading to higher flash gas losses. Studies show that in general the optimum value of molar vapour specific heat lies in the range of **40 to 100 kJ/kmol.K**.

The freezing point of the refrigerant should be lower than the lowest operating temperature of the cycle to prevent blockage of refrigerant pipelines.

Environmental and safety properties:

Next to thermodynamic and thermophysical properties, the environmental and safety properties are very important. In fact, at present the environment friendliness of the refrigerant is a major factor in deciding the usefulness of a particular refrigerant. The important environmental and safety properties are:

a) Ozone Depletion Potential (ODP): According to the Montreal protocol, the ODP of refrigerants should be zero, i.e., they should be non-ozone depleting substances. Refrigerants having non-zero ODP have either already been phased-out (e.g. R 11, R 12) or will be phased-out in near-future(e.g. R22). Since ODP depends mainly on the presence of chlorine or bromine in the molecules, refrigerants having either chlorine (i.e., CFCs and HCFCs) or bromine cannot be used under the new regulations

b) Global Warming Potential (GWP): Refrigerants should have as low a GWP value as possible to minimize the problem of global warming. Refrigerants with zero ODP but a high value of GWP (e.g. R134a) are likely to be regulated in future.

c) Total Equivalent Warming Index (TEWI): The factor TEWI considers both direct (due to release into atmosphere) and indirect (through energy consumption) contributions of refrigerants to global warming. Naturally, refrigerants with as a low a value of TEWI are preferable from global warming point of view.

d) Toxicity: Ideally, refrigerants used in a refrigeration system should be nontoxic. However, all fluids other than air can be called as toxic as they will cause suffocation when their concentration is large enough. Thus toxicity is a relative term, which becomes meaningful only when the degree of concentration and time of exposure required to produce harmful effects are specified. Some fluids are toxic even in small concentrations. Some fluids are mildly toxic, i.e., they are dangerous only when the concentration is large and duration of exposure is long. Some refrigerants such as CFCs and HCFCs are non-toxic when mixed with air in normal condition. However, when they come in contact with an open flame or an electrical heating element, they decompose forming highly toxic elements (e.g. phosgene-$COCl_2$). In general the degree of hazard depends on:

- Amount of refrigerant used vs total space
- Type of occupancy
- Presence of open flames
- Odor of refrigerant, and
- Maintenance condition

Thus from toxicity point-of-view, the usefulness of a particular refrigerant depends on the specific application.

e) Flammability: The refrigerants should preferably be non-flammable and nonexplosive. For flammable refrigerants special precautions should be taken to avoid accidents.

Based on the above criteria, ASHRAE has divided refrigerants into six safety groups (A1 to A3 and B1 to B3). Refrigerants belonging to Group A1 (e.g. R11, R12, R22, R134a, R744, R718) are least hazardous, while refrigerants belonging to Group B3 (e.g. R1140) are most hazardous.

Other important properties are:

f) Chemical stability: The refrigerants should be chemically stable as long as they are inside the refrigeration system.

g) Compatibility with common materials of construction (both metals and nonmetals)

h) Miscibility with lubricating oils: Oil separators have to be used if the refrigerant is not miscible with lubricating oil (e.g. ammonia). Refrigerants that are completely miscible with oils are easier to handle (e.g. R12). However, for refrigerants with limited solubility (e.g. R 22) special precautions should be taken while designing the system to ensure oil return to the compressor

i) Dilelectric strength: This is an important property for systems using hermetic compressors. For these systems the refrigerants should have as high a dielectric strength as possible

j) Ease of leak detection: In the event of leakage of refrigerant from the system, it should be easy to detect the leaks.

Economic properties:

The refrigerant used should preferably be inexpensive and easily available.

Designation of refrigerants:

Figure 26.1 shows the classification of fluids used as refrigerants in vapour compression refrigeration systems. Since a large number of refrigerants have been developed over the years for a wide variety of applications, a numbering system has been adopted to designate various refrigerants. From the number one can get some useful information about the type of refrigerant, its chemical composition, molecular weight etc. All the refrigerants are designated by R followed by a unique number.

i) Fully saturated, halogenated compounds: These refrigerants are derivatives of alkanes (C_nH_{2n+2}) such as methane (CH_4), ethane (C_2H_6). These refrigerants are designated by R XYZ, where:
X+1 indicates the number of Carbon (C) atoms
Y-1 indicates number of Hydrogen (H) atoms, and

Z indicates number of Fluorine (F) atoms
The balance indicates the number of Chlorine atoms. Only 2 digits indicates that the value of X is zero.

Ex: R 22

X = 0 □ No. of Carbon atoms = 0+1 = 1 □ derivative of methane (CH_4)
Y = 2 □ No. of Hydrogen atoms = 2-1 = 1
Z = 2 □ No. of Fluorine atoms = 2
The balance = 4 – no. of (H+F) atoms = 4-1-2 = 1 □ No. of Chlorine atoms = 1
□The chemical formula of R 22 = $CHClF_2$
Similarly it can be shown that the chemical formula of:
R12 = CCl_2F_2
R134a = $C_2H_2F_4$ (derivative of ethane)
(letter a stands for isomer, e.g. molecules having same chemical composition but different atomic arrangement, e.g. R134 and R134a)

ii) Inorganic refrigerants: These are designated by number 7 followed by the molecular weight of the refrigerant (rounded-off).
Ex.: Ammonia: Molecular weight is 17, □ the designation is R 717
Carbon dioxide: Molecular weight is 44, □ the designation is R 744
Water: Molecular weight is 18, □ the designation is R 718

Refrigerants

Mixtures

- Azeotropic

- Zeotropic

Pure fluids

Synthetic

iii) Mixtures: Azeotropic mixtures are designated by 500 series, where as zeotropic refrigerants (e.g. non-azeotropic mixtures) are designated by 400 series.
Azeotropic mixtures:
R 500: Mixture of R 12 (73.8 %) and R 152a (26.2%)
R 502: Mixture of R 22 (48.8 %) and R 115 (51.2%)
R503: Mixture of R 23 (40.1 %) and R 13 (59.9%)
R507A: Mixture of R 125 (50%) and R 143a (50%)
Zeotropic mixtures:
R404A : Mixture of R 125 (44%), R 143a (52%) and R 134a (4%)
R407A : Mixture of R 32 (20%), R 125 (40%) and R 134a (40%)
R407B : Mixture of R 32 (10%), R 125 (70%) and R 134a (20%)
R410A : Mixture of R 32 (50%) and R 125 (50%)

- **CFCs**
- **HCFCs**
- **HFCs**

Natural

- **Organic (HCs)**
- **Inorganic**
  - o **NH_3**
  - o **CO_2**
  - o **H_2O**

Fig.26.1*: Classification of fluids used as refrigerants*

iv) Hydrocarbons:
Propane (C_3H_8) : R 290
n-butane (C_4H_{10}) : R 600
iso-butane (C_4H_{10}) : R 600a
Unsaturated Hydrocarbons: R1150 (C_2H_4)
R1270 (C_3H_6)

UNIT-II
MODULE-IV
AIR – CONDITIONING

PSYCHOMETRY

27.1. Introduction:

Atmospheric air makes up the environment in almost every type of air conditioning system. Hence a thorough understanding of the properties of atmospheric air and the ability to analyze various processes involving air is fundamental to air conditioning design.

Psychrometry is the study of the properties of mixtures of air and water vapour.

Atmospheric air is a mixture of many gases plus water vapour and a number of pollutants (Fig.27.1). The amount of water vapour and pollutants vary from place to place. The concentration of water vapour and pollutants decrease with altitude, and above an altitude of about 10 km, atmospheric air consists of only dry air. The pollutants have to be filtered out before processing the air. Hence, what we process is essentially a mixture of various gases that constitute air and water vapour. This mixture is known as *moist air*.

other pressures also. However, since in most cases the pressures involved are low, one can apply the perfect gas model to estimate psychrometric properties.

27.2.1. Basic gas laws for moist air:

According to the *Gibbs-Dalton law* for a mixture of perfect gases, the total pressure exerted by the mixture is equal to the sum of partial pressures of the constituent gases. According to this law, for a homogeneous perfect gas mixture occupying a volume V and at temperature T, each constituent gas behaves as though the other gases are not present (i.e., there is no interaction between the gases). Each gas obeys perfect gas equation. Hence, the partial pressures exerted by each gas, p_1, p_2, p_3 … and the total pressure p_t are given by:

$$\mathbf{p_1 = \frac{n_1 R_u T}{V};\ p_2 = \frac{n_2 R_u T}{V};\ p_3 = \frac{n_3 R_u T}{V}\ \ldots\ldots} \qquad (27.1)$$
$$\mathbf{p_t = p_1 + p_2 + p_3 + \ldots\ldots}$$

where $n_1, n_2, n_3, \ldots$ are the number of moles of gases 1,2,3,…

Applying this equation to moist air.

$$\mathbf{p = p_t = p_a + p_v} \qquad (27.2)$$

where $p = p_t$ = total barometric pressure
p_a = partial pressure of dry air
p_v = partial pressure of water vapour

27.2.2. Important psychrometric properties:

Dry bulb temperature (DBT) is the temperature of the moist air as measured by a standard thermometer or other temperature measuring instruments.

Saturated vapour pressure (p_{sat}) is the saturated partial pressure of water vapour at the dry bulb temperature. This is readily available in thermodynamic tables and charts. ASHRAE suggests the following regression equation for saturated vapour pressure of water, which is valid for 0 to 100°C.

$$\mathbf{\ln(p_{sat}) = \frac{c_1}{T} + c_2 + c_3 T + c_4 T^2 + c_5 T^3 + c_6 \ln(T)} \qquad (27.3)$$

where p_{sat} = saturated vapor pressure of water in kiloPascals
T = temperature in K
The regression coefficients c_1 to c_6 are given by:

c_1 = -5.80022006E+03, c_2 = -5.516256E+00, c_3 = -4.8640239E-02
c_4 = 4.1764768E-05, c_5 = -1.4452093E-08, c_6 = 6.5459673E+00

Relative humidity (Φ) is defined as the ratio of the mole fraction of water vapour in moist air to mole fraction of water vapour in saturated air at the same temperature and pressure. Using perfect gas equation we can show that:

The *sling psychrometer* is widely used for measurements involving room air or other applications where the air velocity inside the room is small. The sling psychrometer consists of two thermometers mounted side by side and fitted in a frame with a handle for whirling the device through air. The required air circulation (≈ 3 to 5 m/s) over the sensing bulbs is obtained by whirling the psychrometer (≈ 300 RPM). Readings are taken when both the thermometers show steady-state readings.

In the *aspirated psychrometer*, the thermometers remain stationary, and a small fan, blower or syringe moves the air across the thermometer bulbs.

The function of the wick on the wet-bulb thermometer is to provide a thin film of water on the sensing bulb. To prevent errors, there should be a continuous film of water on the wick. The wicks made of cotton or cloth should be replaced frequently, and only distilled water should be used for wetting it. The wick should extend beyond the bulb by 1 or 2 cms to minimize the heat conduction effects along the stem.

Other types of psychrometric instruments:

1. Dunmore Electric Hygrometer
2. DPT meter
3. Hygrometer (Using horse's or human hair)

From steam tables, the saturation temperature of water at 3.13975 Kpa is 24.8°C, hence moisture in air will condense when it comes in contact with the cold surface whose temperature is lower than the dew point temperature. **(Ans.)**

7. Moist air at 1 atm. pressure has a dry bulb temperature of 32°C and a wet bulb temperature of 26°C. Calculate a) the partial pressure of water vapour, b) humidity ratio, c) relative humidity, d) dew point temperature, e) density of dry air in the mixture, f) density of water vapour in the mixture and g) enthalpy of moist air using perfect gas law model and psychrometric equations.

Ans.:

a) Using modified Apjohn equation and the values of DBT, WBT and barometric pressure, the vapour pressure is found to be:

$$\mathbf{p_v = 2.956\ kPa} \qquad \textbf{(Ans.)}$$

b) The humidity ratio W is given by:

$$\mathbf{W = 0.622 \times 2.956/(101.325-2.956) = 0.0187\ kgw/kgda} \qquad \textbf{(Ans.)}$$

c) Relative humidity RH is given by:

$RH = (p_v/p_s) \times 100 = (p_v/\text{saturation pressure at } 32^oC) \times 100$

From steam tables, the saturation pressure of water at 32°C is **4.7552 kPa**, hence,

$$\mathbf{RH = (2.956/4.7552) \times 100 = 62.16\%} \qquad \textbf{(Ans.)}$$

d) Dew point temperature is the saturation temperature of steam at 2.956 kPa. Hence using steam tables we find that:

$$\mathbf{DPT = T_{sat}(2.956\ kPa) = 23.8^oC} \qquad \textbf{(Ans.)}$$

e) Density of dry air and water vapour

Applying perfect gas law to dry air:

$$\textbf{Density of dry air } \rho_a = (p_a/R_aT) = (p_t - p_v)/R_aT = (101.325 - 2.956)/(287.035 \times 305) \times 10^3$$

$$= \mathbf{1.1236\ kg/m^3} \textbf{ of dry air} \qquad \textbf{(Ans.)}$$

f) Similarly the density of water vapour in air is obtained using perfect gas law as:

$$\textbf{Density of water vapour } \rho v = (p_v/R_vT) = 2.956 \times 10^3/(461.52 \times 305) = 0.021\ kg/m^3$$

(Ans.)

g) Enthalpy of moist air is found from the equation:

$$\mathbf{h = 1.005 \times t + W(2501 + 1.88 \times t) = 1.005 \times 32 + 0.0187(2501 + 1.88 \times 32)}$$

$$\mathbf{h = 80.05\ kJ/kg} \textbf{ of dry air} \qquad \textbf{(Ans.)}$$

PSYCHOMETRIC PROCESSES

b) Sensible heating (Process O-B):

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil. The heat transfer rate during this process is given by:

$$Q_h = m_a(h_B - h_O) = m_a c_{pm}(T_B - T_O) \qquad (28.2)$$

where m_a is the mass flow rate of dry air.

From energy balance:

$$Q_h = m_a(h_D - h_O) - m_w h_w \qquad (28.15)$$

where Q_h is the heat supplied through the heating coil and h_w is the enthalpy of steam.

Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a coolind and dehumidification process.

e) Cooling & humidification (Process O-E):

As the name implies, during this process, the air temperature drops and its humidity increases. This process is shown in Fig.28.6. As shown in the figure, this can be achieved by spraying cool water in the air stream. The temperature of water should be lower than the dry-bulb temperature of air but higher than its dew-point temperature to avoid condensation ($T_{DPT} < T_w < T_O$).

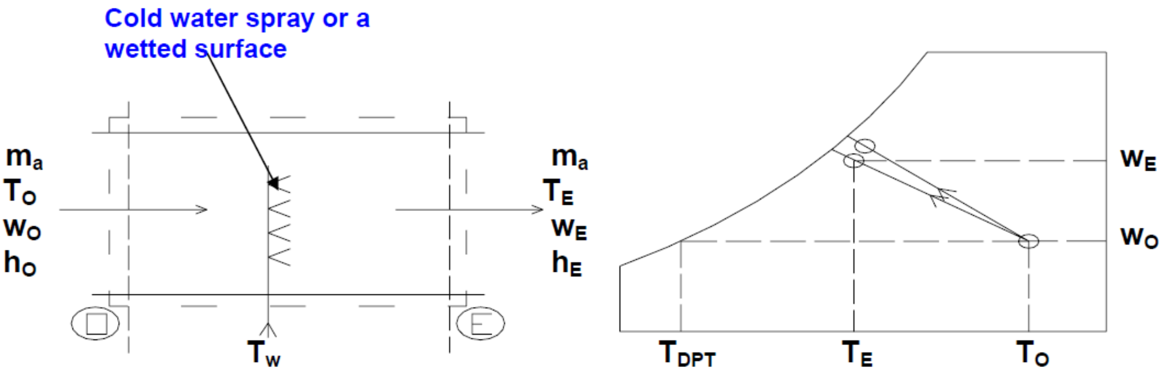


***Fig.28.6**: Cooling and humdification process*

It can be seen that during this process there is sensible heat transfer from air to water and latent heat transfer from water to air. Hence, the total heat transfer depends upon the water temperature. If the temperature of the water sprayed is equal to the wet-bulb temperature of air, then the net transfer rate will be zero as the sensible heat transfer from air to water will be equal to latent heat transfer from water to air. If the water temperature is greater than WBT, then there will be a net heat transfer from water to air. If the water temperature is less than WBT, then the net heat transfer will be from air to water. Under a special case when the spray water is entirely recirculated and is neither heated nor cooled, the system is perfectly insulated and the make-up water is supplied at WBT, then at steady-state, the air undergoes an adiabatic saturation process, during which its WBT remains constant. This is the process of adiabatic saturation discussed in Chapter 27. The process of cooling and humidification is encountered in a wide variety of devices such as evaporative coolers, cooling towers etc.

by manipulating the term in the parenthesis of RHS, it can be shown that:

$$Q_T = Q_S + Q_L = \frac{h_C A_S}{c_{pm}}[(h_i - h_a)] \qquad (28.22)$$

thus the total heat transfer and its direction depends upon the enthalpy difference (or potential) between water and air (h_i-h_a).

if $h_i > h_a$; then the total heat transfer is from water to air and water gets cooled

if $h_i < h_a$; then the total heat transfer is from air to water and water gets heated

if $h_i = h_a$; then the net heat transfer is zero, i.e., the sensible heat transfer rate is equal to but in the opposite direction of latent heat transfer. Temperature of water remains at its wet bulb temperature value

The concept of enthalpy potential is very useful in psychrometric calculations and is frequently used in the design and analysis of evaporative coolers, cooling towers, air washers etc.

At the outlet of the cooling coil; $T_o = 11^oC$ and RH = 90%

From psychrometric chart; W_o = 0.00734 kgw/kgda and h_o = 29.496 kJ/kgda

a) From mass balance across the cooling coil, the condesate rate, m_w is:

$$\mathbf{m_w = m_a(W_i - W_o) = 2.0(0.00988 - 0.00734) = 0.00508\ kg/s}$$

From energy balance across the cooling tower, the required capacity of the cooling coil, Q_c is given by:;

$$\mathbf{Q_c = m_a(h_i - h_o) - m_w.h_w = 2.0(50.155 - 29.496) - 0.00508 \times 29.26 = 41.17\ kW} \quad \textbf{(ans.)}$$

b) The sensible heat transfer rate, Q_s is given by:

$$\mathbf{Q_s = m_a c_{pm}(T_i - T_o) = 2.0 \times 1.0216 \times (25 - 11) = 28.605\ kW}$$

The latent heat transfer rate, Q_l is given by:

$$\mathbf{Q_s = m_a h_{fg}(W_i - W_o) = 2.0 \times 2501.0 \times (0.00988 - 0.00734) = 12.705\ kW^1}$$

The Sensible Heat Factor (SHF) is given by:

$$\mathbf{SHF = Q_s/(Q_s + Q_l) = 28.605/(28.605 + 12.705) = 0.692} \qquad \textbf{(ans.)}$$

c) From its definition, the by-pass factor of the coil, BPF is given by:

$$\mathbf{BPF = (T_o - ADP)/(T_i - ADP) = (11 - 7)/(25 - 7) = 0.222} \qquad \textbf{(ans.)}$$

29.1. Introduction:

Design and analysis of air conditioning systems involves selection of suitable inside and outside design conditions, estimation of the required capacity of cooling or heating equipment, selection of suitable cooling/heating system, selecting supply conditions, design of air transmission and distribution systems etc. Generally, the inputs are the building specifications and its usage pattern and any other special requirements. Figure 29.1 shows the schematic of a basic summer air conditioning system. As shown in the figure, under a typical summer condition, the building gains sensible and latent heats from the surroundings and also due to internal heat sources (RSH and RLH). The supply air to the building extracts the building heat gains from the conditioned space. These heat gains along with other heat gains due to ventilation, return ducts etc. have to be extracted from the air stream by the cooling coil, so that air at required cold and dry condition can be supplied to the building to complete the cycle. In general, the sensible and latent heat transfer rates (GSH and GLH) on the cooling coil are larger than the building heat gains due to the need for ventilation and return duct losses. To estimate the required cooling capacity of the cooling coil (GTH), it is essential to estimate the building and other heat gains. The building heat gains depend on the type of the building, outside conditions and the required inside conditions. Hence selection of suitable inside and outside design conditions is an important step in the design and analysis of air conditioning systems.

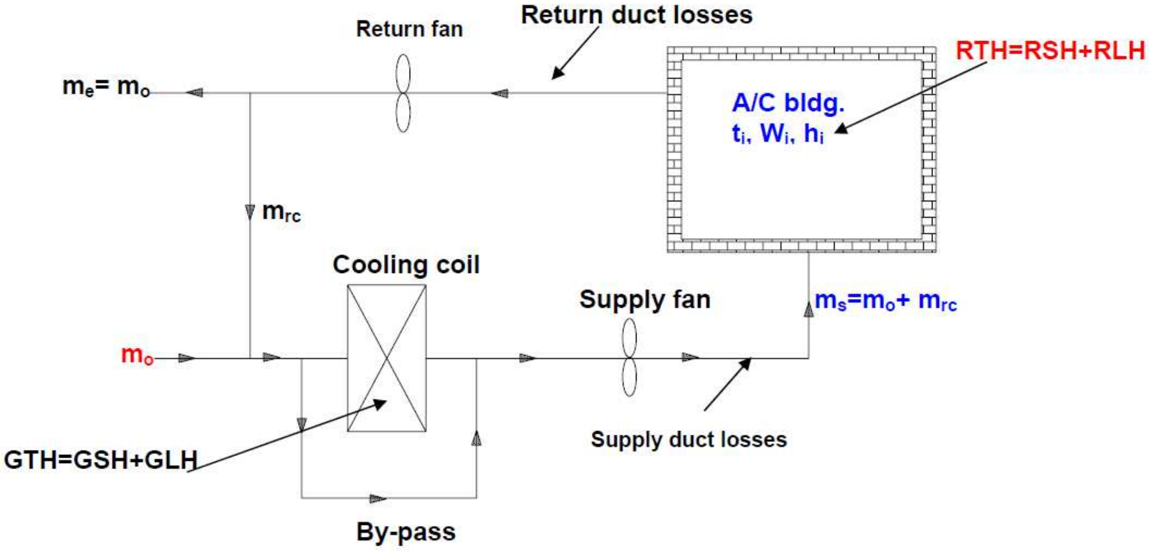


Fig.29.1*: Schematic of a basic summer air conditioning system*

10. A 1.8 meter tall human being with a body mass of 60 kg performs light work (activity = 1.2 met) in an indoor environment. The indoor conditions are: DBT of 30°C, mean radiant temperature of 32°C, air velocity of 0.2 m/s. Assuming an average surface temperature of 34°C for the surface of the human being and light clothing, find the amount of evaporative heat transfer required so that the human being is at neutral equilibrium.

Ans.: Using Du Bois equation, the surface area of the human being A_s is:

$$A_{Du} = 0.202\, m^{0.425} h^{0.725} = 0.202 \times 60^{0.425} \times 1.8^{0.725} = 1.7625\ m^2$$

Hence the total heat generation rate from the body, Q_g is:

$$Q_g = A_s \times (\text{Activity level in met}) \times 58.2 = 1.7625 \times 1.2 \times 58.2 = 123.1\ W$$

Using Belding & Hatch equations, the convective and radiative heat losses from the surface of the body are found as:

$$Q_c = 14.8\, V^{0.5}(t_b - t) = 14.8 \times 0.2^{0.5}(34 - 30) = 26.48\ W$$
$$Q_r = 11.603\ (t_b - t_s) = 11.603(34 - 32) = 23.2\ W$$

For neutral equilibrium,

$$Q_g = Q_c + Q_r + Q_e = \Rightarrow Q_e = Q_g - (Q_c + Q_r)$$

Substituting the values of Qg, Qc and Qr in the above expression, we find that the required amount of evaporative heat transfer Qe is equal to:

$$Q_e = 123.1 - (26.48 + 23.2) = 73.42\ W \qquad \textbf{(Ans.)}$$

30.1. Introduction:

Generally from the building specifications, inside and outside design conditions; the latent and sensible cooling or heating loads on a building can be estimated. Normally, depending on the ventilation requirements of the building, the required outdoor air (fresh air) is specified. The topic of load estimation will be discussed in a later chapter. From known loads on the building and design inside and outside conditions, psychrometric calculations are performed to find:

1. Supply air conditions (air flow rate, DBT, humidity ratio & enthalpy)
2. Coil specifications (Latent and sensible loads on coil, coil ADP & BPF)

In this chapter fixing of supply air conditions and coil specifications for summer air conditioning systems are discussed. Since the procedure is similar for winter air conditioning system, the winter air conditioning systems are not discussed here.

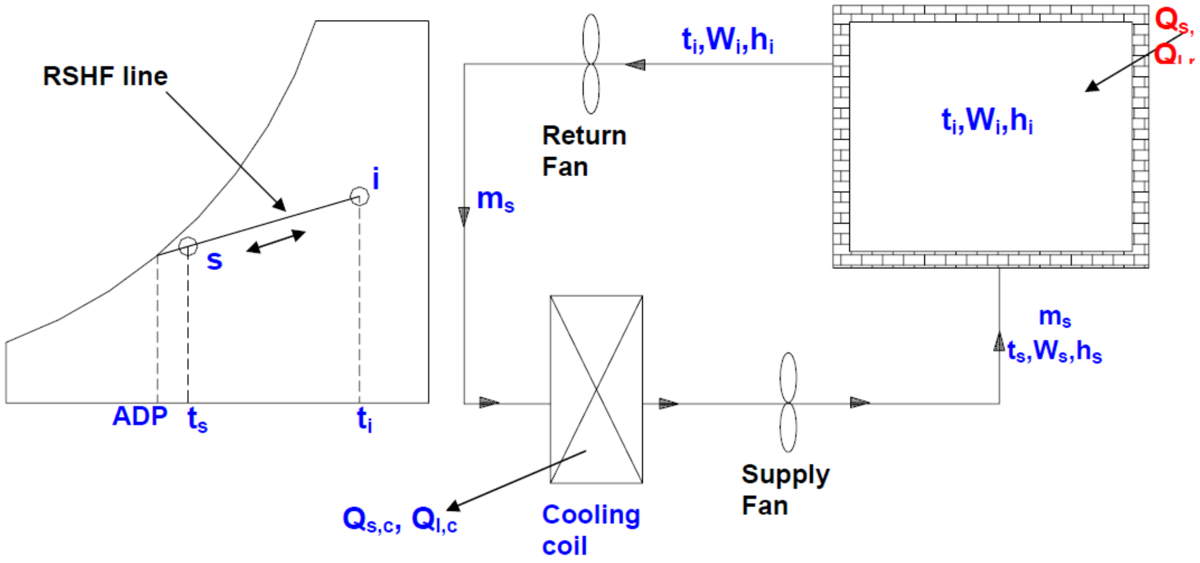


Fig.30.1: *A simple, 100% re-circulation type air conditioning system*

From the RSHF value one can calculate the slope of the process undergone by the air as it flows through the conditioned space (process s-i) as:

$$\textbf{slope of process line } s-i,\ \tan\theta = \frac{1}{2451}\left(\frac{1-RSHF}{RSHF}\right) \qquad (30.5)$$

Since the condition i is known say, from thermal comfort criteria, knowing the slope, one can draw the process line s-i through i. The intersection of this line with the saturation curve gives the ADP of the cooling coil as shown in Fig.30.1. It should be noted that for the given room sensible and latent cooling loads, **the supply condition must always lie on this line so that the it can extract the sensible and latent loads on the conditioned space in the required proportions**.

Since the case being considered is one of 100 % re-circulation, the process that the air undergoes as it flows through the cooling coil (i.e. process i-s) will be exactly opposite to the process undergone by air as it flows through the room (process s-i). Thus, the temperature and humidity ratio of air decrease as it flows through the cooling coil and temperature and humidity ratio increase as air flows through the conditioned space. Assuming no heat transfer due to the ducts and fans, the sensible and latent heat transfer rates at the cooling coil are exactly equal to the sensible and latent heat transfer rates to the conditioned space; i.e.,

$$Q_{s,r} = Q_{s,c} \ \&\ Q_{l,r} = Q_{l,c} \qquad (30.6)$$

Fixing of supply condition:

Case i) By-pass factor of the cooling coil is zero:

Figure 30.2 shows the schematic of the summer air conditioning system with outdoor air and the corresponding process on psychrometric chart, when the by-pass factor X is zero. Since the sensible and latent cooling loads on the conditioned space are assumed to be known from cooling load calculations, similar to the earlier case, one can draw the process line s-i, from the RSHF and state i. The intersection of this line with the saturation curve gives the room ADP. As shown on the psychrometric chart, when the by-pass factor is zero, the room ADP is equal to coil ADP, which in turn is equal to the temperature of the supply air. Hence from the supply temperature one can calculate the required supply air mass flow rate (which is the minimum required as X is zero) using the equation:

$$\mathbf{m_s = \frac{Q_{s,r}}{C_{pm}(t_i - t_s)} = \frac{Q_{s,r}}{C_{pm}(t_i - t_{ADP})}} \qquad (30.12)$$

From the supply mass flow rate, one can find the supply air humidity ratio and enthalpy using Eqns.(30.9) and (30.10).

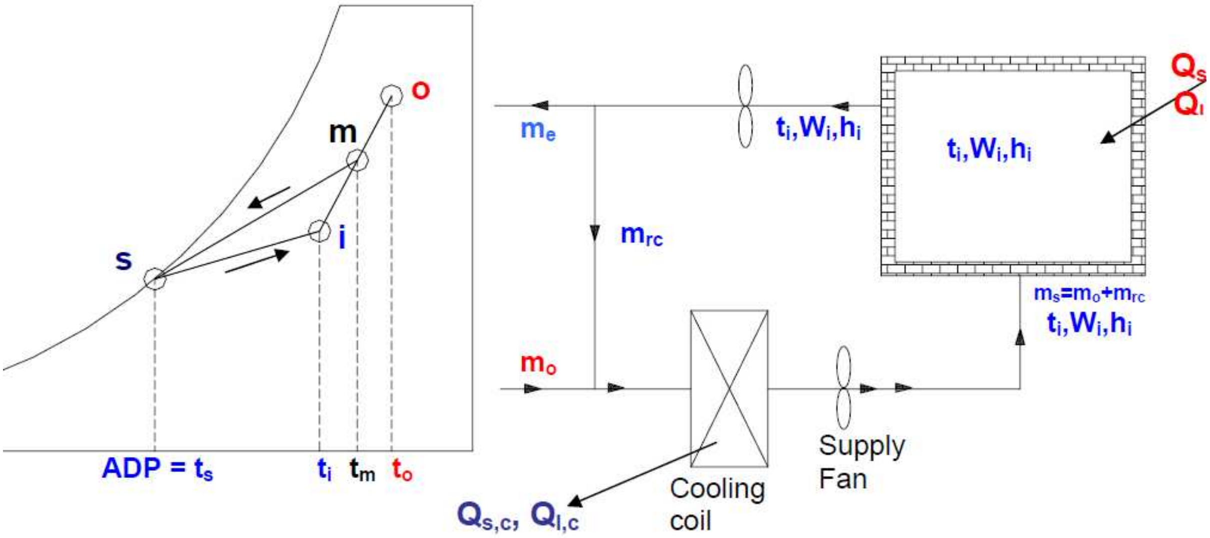


Fig.30.2: A summer air conditioning system with outdoor air for ventilation and a zero by-pass factor

From mass balance of air;

$$\mathbf{m_s = m_{rc} + m_o} \qquad (30.13)$$

Where m_{rc} is the re-circulated air flow rate and m_o is the outdoor air flow rate. Since either m_o or the ratio $m_o : m_{rc}$ are specified, one can calculate the amount of re-circulated air from Eqn.(30.13).

Calculation of coil loads:

From energy balance across the cooling coil; the sensible, latent and total heat transfer rates, $Q_{s,c}$, $Q_{l,c}$ and $Q_{t,c}$ at the cooling coil are given by:

$$\begin{aligned} Q_{s,c} &= m_s C_{pm}(t_m - t_s) \\ Q_{l,c} &= m_s h_{fg}(W_m - W_s) \\ Q_{t,c} &= Q_{s,c} + Q_{l,c} = m_s(h_m - h_s) \end{aligned} \tag{30.14}$$

Where 'm' refers to the mixing condition which is a result of mixing of the recirculated air with outdoor air. Applying mass and energy balance to the mixing process one can obtain the state of the mixed air from the equation:

$$\frac{m_o}{m_s} = \frac{W_m - W_i}{W_o - W_i} = \frac{h_m - h_i}{h_o - h_i} \approx \frac{t_m - t_i}{t_o - t_i} \tag{30.15}$$

Since $(m_o/m_s) > 0$, from the above equation it is clear that $W_m > W_i$, $h_m > h_i$ and $t_m > t_i$. This implies that $\mathbf{m_s(h_m - h_s) > m_s(h_i - h_s)}$, or the load on the cooling coil is greater than the load on the conditioned space. This is of course due to the fact that during mixing, some amount of hot and humid air is added and the same amount of relative cool and dry air is exhausted ($m_o = m_e$).

From Eqn.(30.1) to (30.3) and (30.14), the difference between the cooling load on the coil and cooling load on the conditioned space can be shown to be equal to:

$$\begin{aligned} Q_{s,c} - Q_{s,r} &= m_o C_{pm}(t_o - t_i) \\ Q_{l,c} - Q_{l,r} &= m_o h_{fg}(W_o - W_i) \\ Q_{t,c} - Q_{t,r} &= m_o(h_o - h_i) \end{aligned} \tag{30.16}$$

From the above equation it is clear that the difference between cooling coil and conditioned space increases as the amount of outdoor air (m_o) increases and/or the outdoor air becomes hotter and more humid.

The line joining the mixed condition 'm' with the coil ADP is the process line undergone by the air as it flows through the cooling coil. The slope of this line depends on the Coil Sensible Heat Factor (CSHF) given by:

$$\text{CSHF} = \frac{Q_{s,c}}{Q_{s,c} + Q_{l,c}} = \frac{Q_{s,c}}{Q_{t,c}} \tag{30.17}$$

Case ii: Coil by-pass factor, $X > 0$:

For actual cooling coils, the by-pass factor will be greater than zero, as a result the air temperature at the exit of the cooling coil will be higher than the coil ADP. This is shown in Fig.30.3 along with the process on psychrometric chart. It can be seen from the figure that when X > 0, the room ADP will be different from the coil ADP. The system shown in Fig.30.3 is adequate when the RSHF is high (> 0.75).

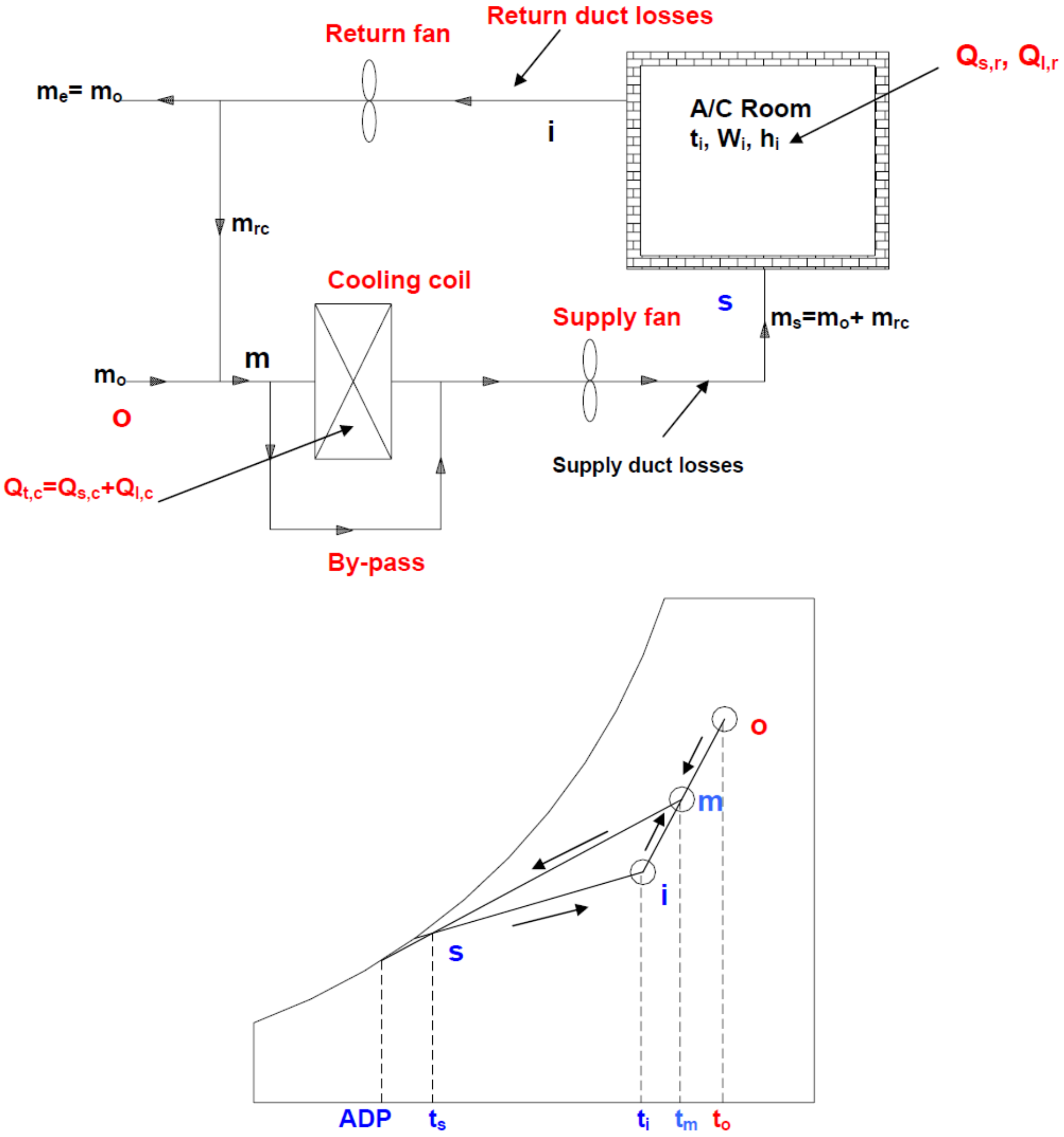


Fig.30.3: *A summer air conditioning system with outdoor air for ventilation and a non-zero by-pass factor*

Normally in actual systems, either the supply temperature (t_s) or the temperature rise of air as it flows through the conditioned space (t_i-t_s) will be specified. Then the step-wise procedure for finding the supply air conditions and the coil loads are as follows:

i. Since the supply temperature is specified one can calculate the required supply air flow rate and supply conditions using Eqns. (30.8) to (30.10).

ii. Since conditions 'i', supply air temperature t_s and RSHF are known, one can draw the line i-s. The intersection of this line with the saturation curve gives the room ADP.

iii. Condition of air after mixing (point 'm') is obtained from known values of m_s and m_o using Eqn.(30.15).

iv. Now joining points 'm' and 's' gives the process line of air as it flows through the cooling coil. The intersection of this line with the saturation curve gives the coil ADP. It can be seen that the coil ADP is lower than the room ADP.

v. The capacity of the cooling coil is obtained from Eqn.(30.14).

vi. From points 'm', 's' and coil ADP, the by-pass factor of the cooling coil can be calculated.

If the coil ADP and coil by-pass factor are given instead of the supply air temperature, then a trial-and-error method has to be employed to obtain the supply air condition.

30.2.3. High latent cooling load applications (low RSHF):

When the latent load on the building is high due either to high outside humidity or due to large ventilation requirements (e.g. hospitals) or due to high internal latent loads (e.g. presence of kitchen or laundry), then the simple system discussed above leads to very low coil ADP. A low coil ADP indicates operation of the refrigeration system at low evaporator temperatures. Operating the system at low evaporator temperatures decreases the COP of the refrigeration system leading to higher costs. Hence a reheat coil is sometimes used so that the cooling coil can be operated at relatively high ADP, and at the same time the high latent load can also be taken care of. Figure 30.4 shows an air conditioning system with reheat coil along with the psychrometric representation of the process. As shown in the figure, in a system with reheat coil, air is first cooled and dehumidified from point 'm' to point 'c' in the cooling coil and is then reheated sensibly to the required supply temperature t_s using the reheat coil. If the supply temperature is specified, then the mass flow rate and state of the supply air and condition of the air after mixing can be obtained using equations given above. Since the heating process in the reheat coil is sensible, the process line c-s will be horizontal. Thus if the coil ADP is known, then one can draw the coil condition line and the intersection of this line with the horizontal line drawn from supply state 's' gives the condition of the air at the exit of the cooling coil. From this condition, one can calculate the load on the cooling coil using the supply mass flow rate and state of air after mixing. The capacity of the reheat coil is then obtained from energy balance across it, i.e.,

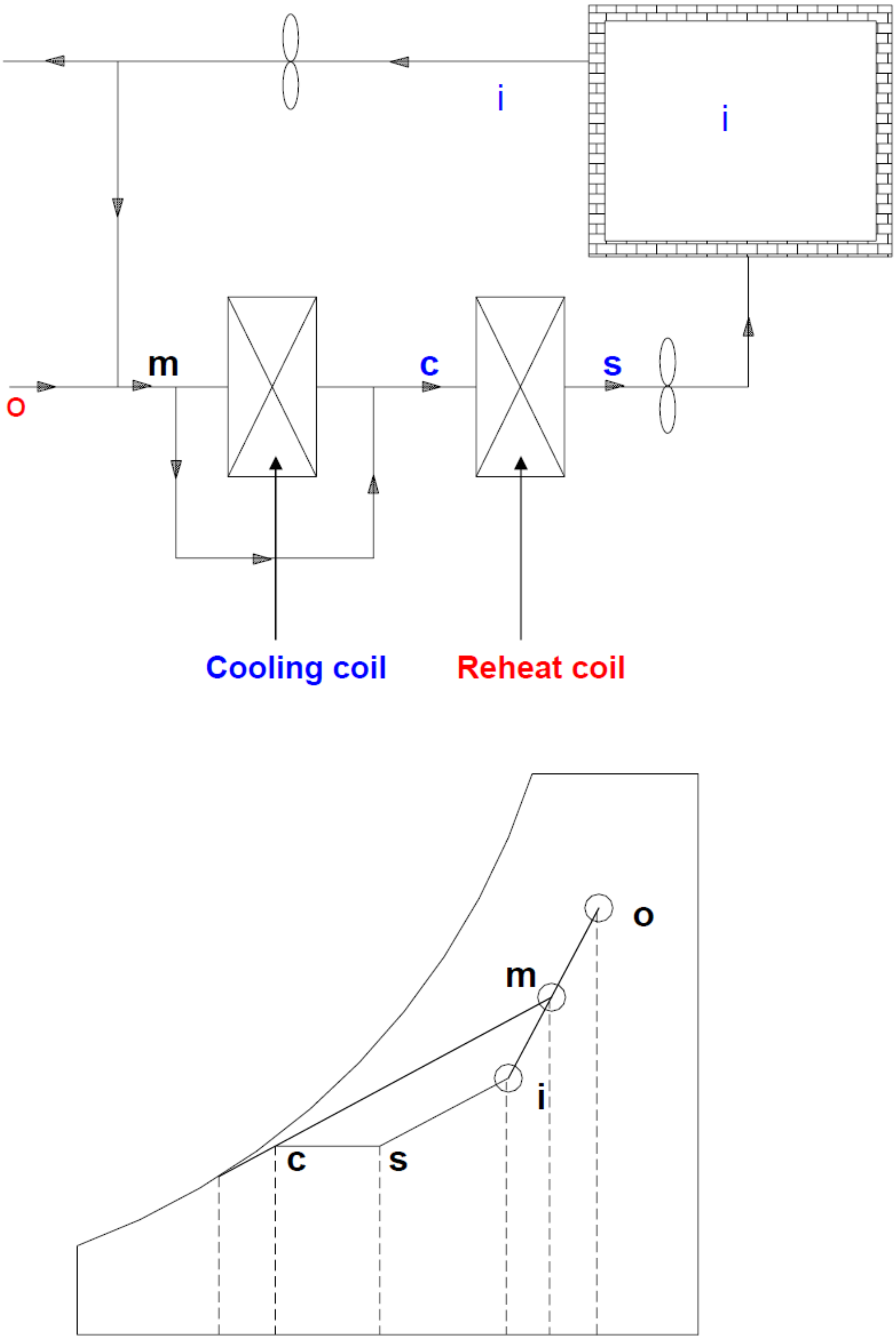


Fig.30.4: *A summer air conditioning system with reheat coil for high latent cooling load applications*

Advantages and disadvantages of reheat coil:

a) Refrigeration system can be operated at reasonably high evaporator temperatures leading to high COP and low running cost.
b) However, mass flow rate of supply air increases due to reduced temperature rise (t_i-t_s) across the conditioned space
c) Wasteful use of energy as air is first cooled to a lower temperature and then heated. Energy is required for both cooling as well as reheat coils. However, this can be partially offset by using waste heat such as heat rejected at the condenser for reheating of air.

Thus the actual benefit of reheat coil depends may vary from system.

i. As much as possible the supply air quantity should be minimized so that smaller ducts and fans can be used leading savings in cost of space, material and power. However, the minimum amount should be sufficient to prevent the feeling of stagnation. If the required air flow rate through the cooling coil is insufficient, then it is possible to mix some amount of re-circulated air with this air so that amount of air supplied to the conditioned space increases. This merely increases the supply air flow rate, but does not affect sensible and cooling loads on the conditioned space. Generally, the temperature rise (t_i-t_s) will be in the range of 8 to 15^{o}C.

ii. The cooling coil should have 2 to 6 rows for moderate climate and 6 to 8 rows in hot and humid climate. The by-pass factor of the coil varies from 0.05 to 0.2. The by-pass factor decreases as the number of rows increases and vice versa. The fin pitch and air velocity should be suitable.

iii. If chilled water is used for cooling and dehumidification, then the coil ADP will be higher than about 4^{o}C.

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