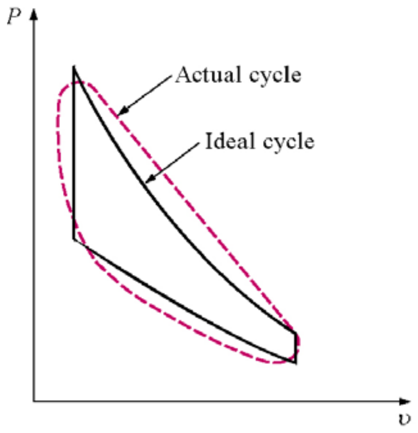
Chapter 8: Gas Power Cycles

Our study of gas power cycles will involve the study of those heat engines in which the working fluid remains in the gaseous state throughout the cycle. We often study the ideal cycle in which internal irreversibilities and complexities (the actual intake of air and fuel, the actual combustion process, and the exhaust of products of combustion among others) are removed.



We will be concerned with how the major parameters of the cycle affect the performance of heat engines. The performance is often measured in terms of the cycle efficiency.

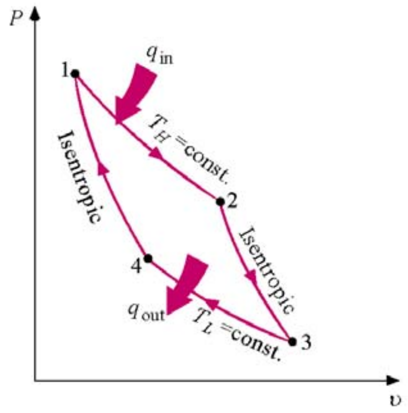
$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

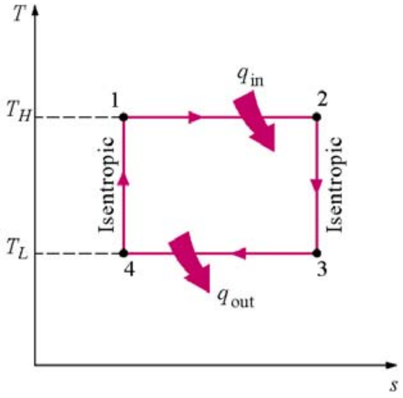
Carnot Cycle

The Carnot cycle was introduced in Chapter 5 as the most efficient heat engine that can operate between two fixed temperatures T_H and T_L. The Carnot cycle is described by the following four processes.

Carnot Cycle

Process	Description
1-2	Isothermal heat addition
2-3	Isentropic expansion
3-4	Isothermal heat rejection
4-1	Isentropic compression





Note the processes on both the *P*-*v* and *T*-*s* diagrams. The areas under the process curves on the *P*-*v* diagram represent the work done for closed systems. The net cycle work done is the area enclosed by the cycle on the *P*-*v* diagram. The areas under the process curves on the *T*-*s* diagram represent the heat transfer for the processes. The net heat added to the cycle is the area that is enclosed by the cycle on the *T*-*s* diagram. For a cycle we know $W_{\text{net}} = Q_{\text{net}}$; therefore, the areas enclosed on the *P*-*v* and *T*-*s* diagrams are equal.

$$\eta_{th,\,Carnot} = 1 - \frac{T_L}{T_H}$$

We often use the Carnot efficiency as a means to think about ways to improve the cycle efficiency of other cycles. One of the observations about the efficiency of both ideal and actual cycles comes from the Carnot efficiency: Thermal efficiency increases with an increase in the average temperature at which heat is supplied to the system or with a decrease in the average temperature at which heat is rejected from the system.

Air-Standard Assumptions

In our study of gas power cycles, we assume that the working fluid is air, and the air undergoes a thermodynamic cycle even though the working fluid in the actual power system does not undergo a cycle.

To simplify the analysis, we approximate the cycles with the following assumptions:

- The air continuously circulates in a closed loop and always behaves as an ideal gas.

- All the processes that make up the cycle are internally reversible.

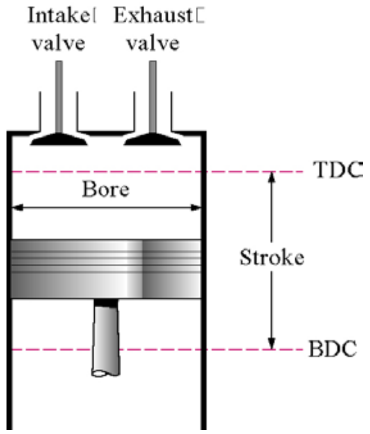
- The combustion process is replaced by a heat-addition process from an external source.

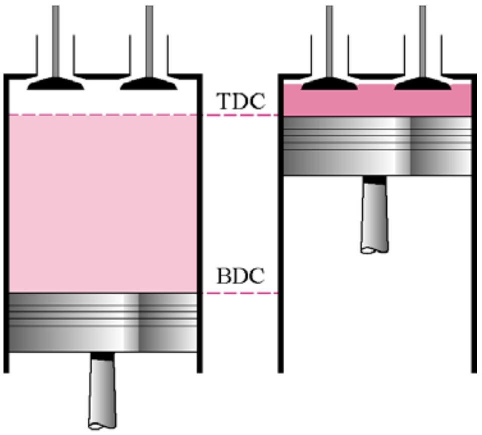
- A heat rejection process that restores the working fluid to its initial state replaces the exhaust process.

- The cold-air-standard assumptions apply when the working fluid is air and has constant specific heat evaluated at room temperature (25^{o}C or 77^{o}F).

Terminology for Reciprocating Devices

The following is some terminology we need to understand for reciprocating engines—typically piston-cylinder devices. Let's look at the following figures for the definitions of top dead center (TDC), bottom dead center (BDC), stroke, bore, intake valve, exhaust valve, clearance volume, displacement volume, compression ratio, and mean effective pressure.





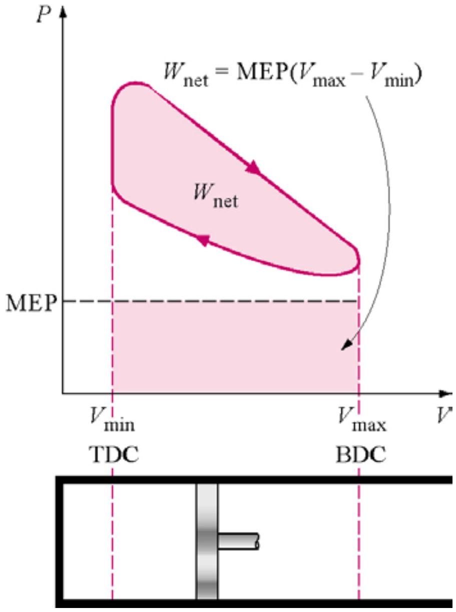
(*a*) Displacement volume

(*b*) Clearance volume

The compression ratio *r* of an engine is the ratio of the maximum volume to the minimum volume formed in the cylinder.

$$r = \frac{V\max}{V\min} = \frac{V_{BDC}}{V_{TDC}}$$

The mean effective pressure **(MEP)** is a fictitious pressure that, if it operated on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.



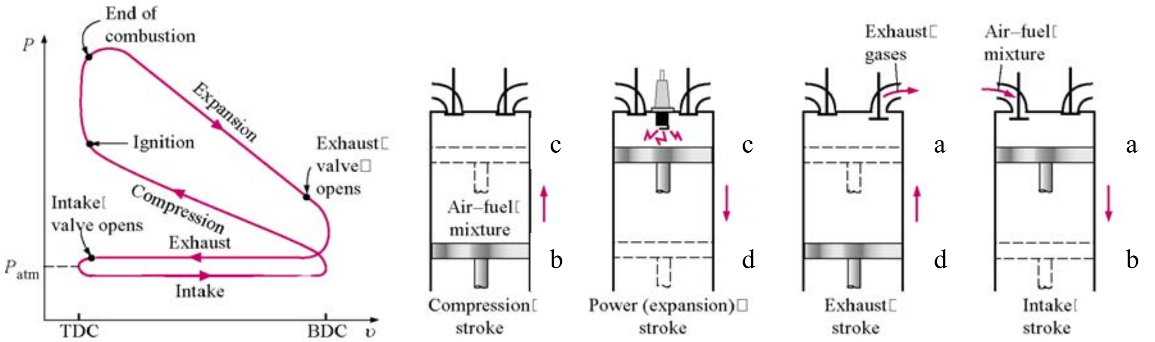
$$MEP = \frac{W_{net}}{V_{\max} - V_{\min}} = \frac{w_{net}}{v_{\max} - v_{\min}}$$

Otto Cycle: The Ideal Cycle for Spark-Ignition Engines

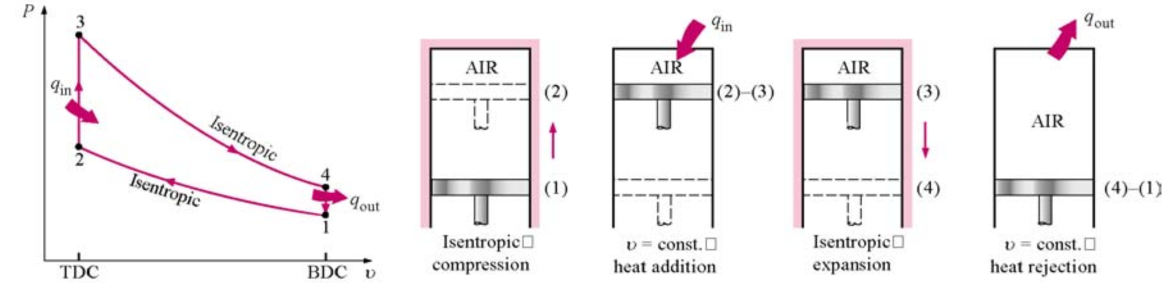
Consider the automotive spark-ignition power cycle.

Processes
Intake stroke
Compression stroke
Power (expansion) stroke
Exhaust stroke

Often the ignition and combustion process begins before the completion of the compression stroke. The number of crank angle degrees before the piston reaches TDC on the number one piston at which the spark occurs is called the engine timing. What are the compression ratio and timing of your engine in your car, truck, or motorcycle?



(a) Actual four-stroke spark-ignition engine

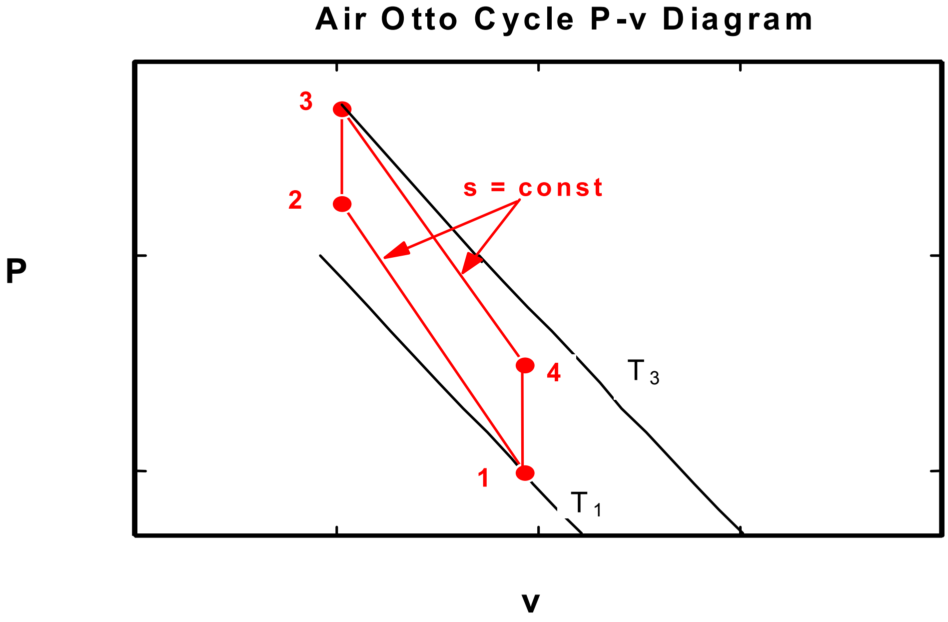


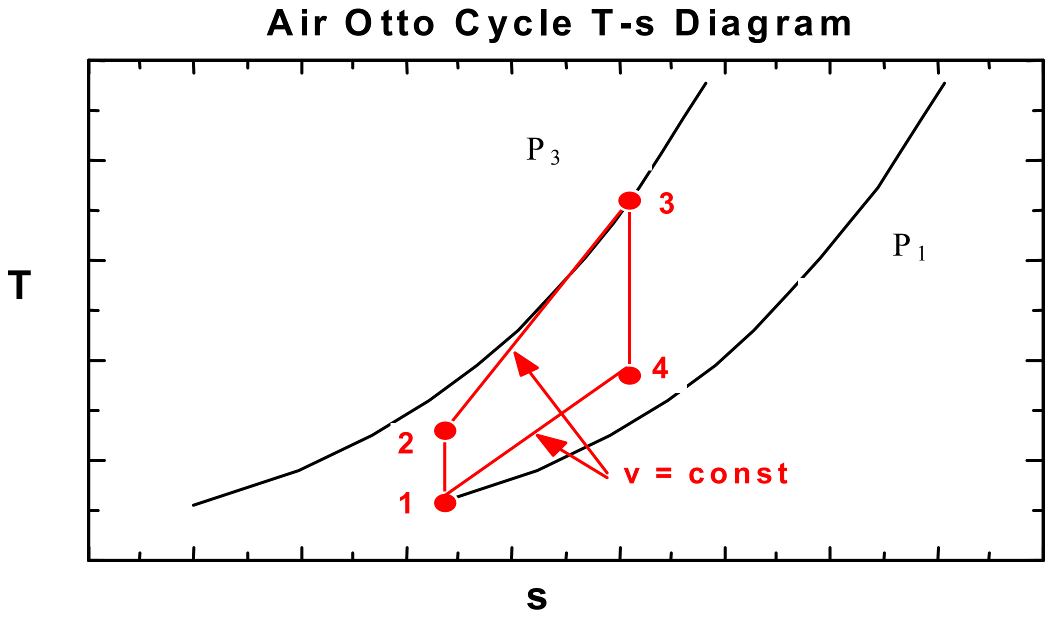
(b) Ideal Otto cycle

The air-standard Otto cycle is the ideal cycle that approximates the spark-ignition combustion engine.

Process	Description
1-2	Isentropic compression
2-3	Constant volume heat addition
3-4	Isentropic expansion
4-1	Constant volume heat rejection

The *P-v* and *T-s* diagrams are





Thermal Efficiency of the Otto cycle:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{Q_{net}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

Now to find Q_{in} and Q_{out}.

Apply first law closed system to process 2-3, V = constant.

$$Q_{net,\,23} - W_{net,\,23} = \Delta U_{23}$$
$$W_{net,\,23} = W_{other,\,23} + W_{b,\,23} = 0 + \int_2^3 P\,dV = 0$$

Thus, for constant specific heats,

$$Q_{net,\,23} = \Delta U_{23}$$
$$Q_{net,\,23} = Q_{in} = mC_v(T_3 - T_2)$$

Apply first law closed system to process 4-1, V = constant.

$$Q_{net,\,41} - W_{net,\,41} = \Delta U_{41}$$
$$W_{net,\,41} = W_{other,\,41} + W_{b,\,41} = 0 + \int_4^1 P\,dV = 0$$

Thus, for constant specific heats,

$$Q_{net,\,41} = \Delta U_{41}$$
$$Q_{net,\,41} = -Q_{out} = mC_v(T_1 - T_4)$$
$$Q_{out} = -mC_v(T_1 - T_4) = mC_v(T_4 - T_1)$$

The thermal efficiency becomes

$$\eta_{th,\,Otto} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$= 1 - \frac{mC_v(T_4 - T_1)}{mC_v(T_3 - T_2)}$$

$$\eta_{th,\,Otto} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$= 1 - \frac{T_1(T_4 / T_1 - 1)}{T_2(T_3 / T_2 - 1)}$$

Recall processes 1-2 and 3-4 are isentropic, so

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} \quad and \quad \frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1}$$

Since $V_3 = V_2$ and $V_4 = V_1$, we see that

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

$$or$$

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

The Otto cycle efficiency becomes

$$\eta_{th,\,Otto} = 1 - \frac{T_1}{T_2}$$

Is this the same as the Carnot cycle efficiency?

Since process 1-2 is isentropic,

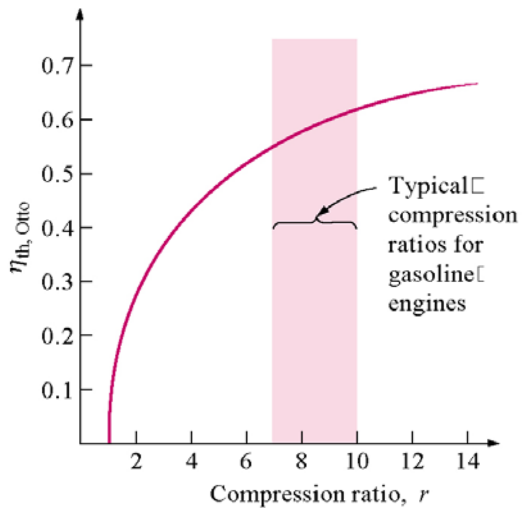
$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

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

where the compression ratio is $r = V_1/V_2$ and

$$\eta_{th,\,Otto} = 1 - \frac{1}{r^{k-1}}$$

We see that increasing the compression ratio increases the thermal efficiency. However, there is a limit on r depending upon the fuel. Fuels under high temperature resulting from high compression ratios will prematurely ignite, causing knock.



Example 8-1

An Otto cycle having a compression ratio of 9:1 uses air as the working fluid. Initially $P_1 = 95$ kPa, $T_1 = 17^{o}$C, and $V_1 = 3.8$ liters. During the heat addition process, 7.5 kJ of heat are added. Determine all T's, P's, η_{th}, the back work ratio, and the mean effective pressure.

Process Diagrams: Review the P-v and T-s diagrams given above for the Otto cycle.

Assume constant specific heats with $C_v = 0.718$ kJ/kg ·K, $k = 1.4$. (Use the 300 K data from Table A-2)

Process 1-2 is isentropic; therefore, recalling that $r = V_1/V_2 = 9$,

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

$$= (17+273)K(9)^{1.4-1}$$

$$= 698.4K$$

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^k = P_1(r)^k$$

$$= 95kPa(9)^{1.4}$$

$$= 2059\,kPa$$

The first law closed system for process 2-3 was shown to reduce to (your homework solutions must be complete; that is, develop your equations from the application of the first law for each process as we did in obtaining the Otto cycle efficiency equation)

$$Q_{in} = mC_v(T_3 - T_2)$$

Let $q_{\text{in}} = Q_{\text{in}} / m$ and $m = V_1 / v_1$

$$v_1 = \frac{RT_1}{P_1}$$

$$= \frac{0.287\dfrac{kJ}{kg \cdot K}(290\,K)}{95\,kPa}\frac{m^3kPa}{kJ}$$

$$= 0.875\frac{m^3}{kg}$$

$$q_{in} = \frac{Q_{in}}{m} = Q_{in}\frac{v_1}{V_1}$$

$$= 7.5kJ\frac{0.875\frac{m^3}{kg}}{3.8\cdot 10^{-3}m^3}$$

$$= 1727\frac{kJ}{kg}$$

Then,

$$T_3 = T_2 + \frac{q_{in}}{C_v}$$

$$= 698.4K + \frac{1727\frac{kJ}{kg}}{0.718\frac{kJ}{kg\cdot K}}$$

$$= 3103.7\,K$$

Using the combined gas law

$$P_3 = P_2\frac{T_3}{T_2} = 9.15\,MPa$$

Process 3-4 is isentropic; therefore,

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1}$$

$$= (3103.7)K\left(\frac{1}{9}\right)^{1.4-1}$$

$$= 1288.8\,K$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = P_3\left(\frac{1}{r}\right)^{k}$$

$$= 9.15\ MPa\left(\frac{1}{9}\right)^{1.4}$$

$$= 422\,kPa$$

Process 4-1 is constant volume. So the first law for the closed system gives, on a mass basis,

$$Q_{out} = mC_v(T_4 - T_1)$$

$$q_{out} = \frac{Q_{out}}{m} = C_v(T_4 - T_1)$$

$$= 0.718\frac{kJ}{kg \cdot K}(1288.8 - 290)K$$

$$= 717.1\frac{kJ}{kg}$$

For the cycle, $\Delta u = 0$, and the first law gives

$$\begin{aligned} w_{net} &= q_{net} = q_{in} - q_{out} \\ &= (1727 - 717.4)\frac{kJ}{kg} \\ &= 1009.6\frac{kJ}{kg} \end{aligned}$$

The thermal efficiency is

$$\begin{aligned} \eta_{th,\,Otto} &= \frac{w_{net}}{q_{in}} = \frac{1009.6\dfrac{kJ}{kg}}{1727\dfrac{kJ}{kg}} \\ &= 0.585 \;\; or \;\; 58.5\% \end{aligned}$$

The mean effective pressure is

$$\begin{aligned} MEP &= \frac{W_{net}}{V_{\max} - V_{\min}} = \frac{w_{net}}{v_{\max} - v_{\min}} \\ &= \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1(1 - v_2/v_1)} = \frac{w_{net}}{v_1(1 - 1/r)} \\ &= \frac{1009.6\dfrac{kJ}{kg}}{0.875\dfrac{m^3}{kg}(1 - \dfrac{1}{9})}\frac{m^3 kPa}{kJ} = 1298 \; kPa \end{aligned}$$

The back work ratio is (can you show that this is true?)

$$BWR = \frac{w_{comp}}{w_{\exp}} = \frac{\Delta u_{12}}{-\Delta u_{34}}$$

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

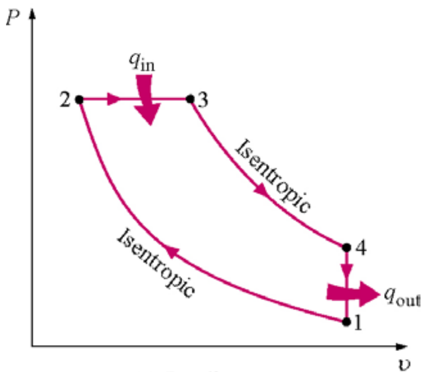
$$= 0.225 \ \ or \ \ 22.5\%$$

Air-Standard Diesel Cycle

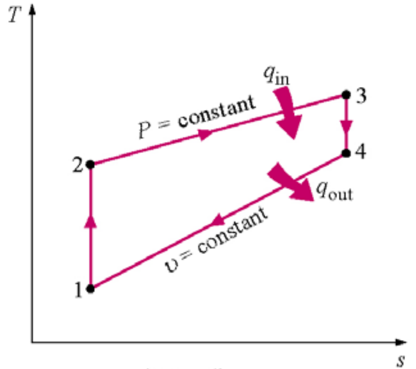
The air-standard Diesel cycle is the ideal cycle that approximates the Diesel combustion engine

Process	Description
1-2	Isentropic compression
2-3	Constant pressure heat addition
3-4	Isentropic expansion
4-1	Constant volume heat rejection

The *P-v* and *T-s* diagrams are



(a) P-υdiagram



(b) T-s diagram

Thermal efficiency of the Diesel cycle

$$\eta_{th,\ Diesel} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

Now to find Q_{in} and Q_{out}.

Apply the first law closed system to process 2-3, P = constant.

$$\begin{aligned} E_{in} - E_{out} &= \Delta E \\ Q_{net,\,23} - W_{net,\,23} &= \Delta U_{23} \\ W_{net,\,23} &= W_{other,\,23} + W_{b,\,23} = 0 + \int_2^3 PdV \\ &= P_2(V_3 - V_2) \end{aligned}$$

Thus, for constant specific heats

$$\begin{aligned} Q_{net,\,23} &= \Delta U_{23} + P_2(V_3 - V_2) \\ Q_{net,\,23} &= Q_{in} = mC_v(T_3 - T_2) + mR(T_3 - T_2) \\ Q_{in} &= mC_p(T_3 - T_2) \end{aligned}$$

Apply the first law closed system to process 4-1, V = constant (just like the Otto cycle)

$$\begin{aligned} E_{in} - E_{out} &= \Delta E \\ Q_{net,\,41} - W_{net,\,41} &= \Delta U_{41} \\ W_{net,\,41} &= W_{other,\,41} + W_{b,\,41} = 0 + \int_4^1 PdV = 0 \end{aligned}$$

Thus, for constant specific heats

$$\begin{aligned} Q_{net,\,41} &= \Delta U_{41} \\ Q_{net,\,41} &= -Q_{out} = mC_v(T_1 - T_4) \\ Q_{out} &= -mC_v(T_1 - T_4) = mC_v(T_4 - T_1) \end{aligned}$$

The thermal efficiency becomes

$$\eta_{th,\,Diesel} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$= 1 - \frac{mC_v(T_4 - T_1)}{mC_p(T_3 - T_2)}$$

$$\eta_{th,\,Diesel} = 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$= 1 - \frac{1}{k}\frac{T_1(T_4 / T_1 - 1)}{T_2(T_3 / T_2 - 1)}$$

What is T_3/T_2 ?

$$\frac{P_3V_3}{T_3} = \frac{P_2V_2}{T_2} \quad \text{where } P_3 = P_2$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$$

where r_c is called the cutoff ratio, defined as V_3/V_2, and is a measure of the duration of the heat addition at constant pressure. Since the fuel is injected directly into the cylinder, the cutoff ratio can be related to the number of degrees that the crank rotated during the fuel injection into the cylinder.

What is T_4/T_1 ?

$$\frac{P_4V_4}{T_4} = \frac{P_1V_1}{T_1} \quad \text{where } V_4 = V_1$$

$$\frac{T_4}{T_1} = \frac{P_4}{P_1}$$

Recall processes 1-2 and 3-4 are isentropic, so

$$P_1V_1^k = P_2V_2^k \quad \text{and} \quad P_4V_4^k = P_3V_3^k$$

Since $V_4 = V_1$ and $P_3 = P_2$, we divide the second equation by the first equation and obtain

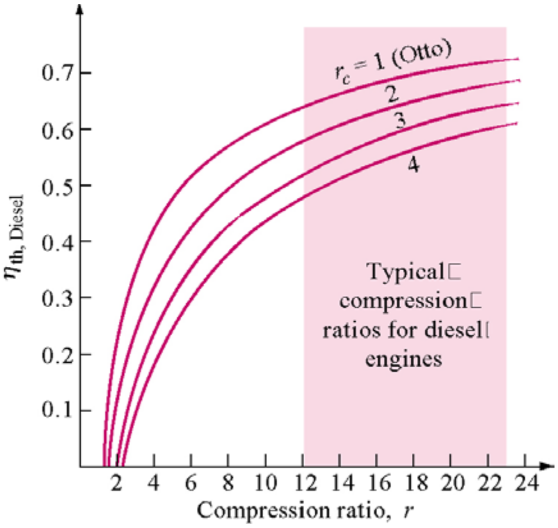
$$\frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^k = r_c^k$$

Therefore,

$$\begin{aligned}\eta_{th,\,Diesel} &= 1 - \frac{1}{k}\frac{T_1(T_4 / T_1 - 1)}{T_2(T_3 / T_2 - 1)} \\ &= 1 - \frac{1}{k}\frac{T_1}{T_2}\frac{r_c^k - 1}{(r_c - 1)} \\ &= 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)}\end{aligned}$$

What happens as r_c goes to 1? Sketch the P-v diagram for the Diesel cycle and show r_c approaching1 in the limit.

When $r_c > 1$ for a fixed r, $\eta_{th,\,Diesel} < \eta_{th,\,Otto}$. But, since $r_{Diesel} > r_{Otto}$, $\eta_{th,\,Diesel} > \eta_{th,\,Otto}$.

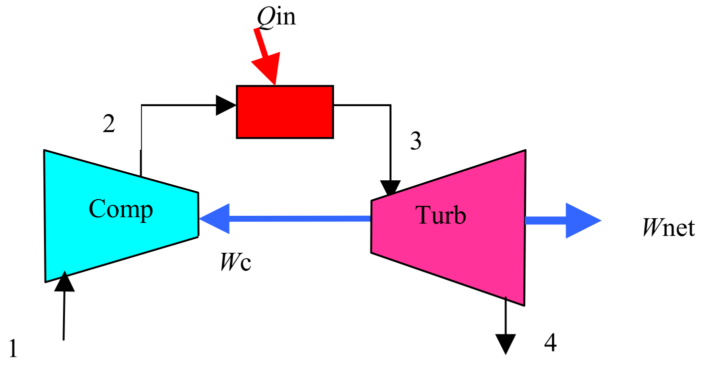


Brayton Cycle

The Brayton cycle is the air-standard ideal cycle approximation for the gas-turbine engine. This cycle differs from the Otto and Diesel cycles in that the processes making the cycle occur in open systems or control volumes. Therefore, an open system, steady-flow analysis is used to determine the heat transfer and work for the cycle.

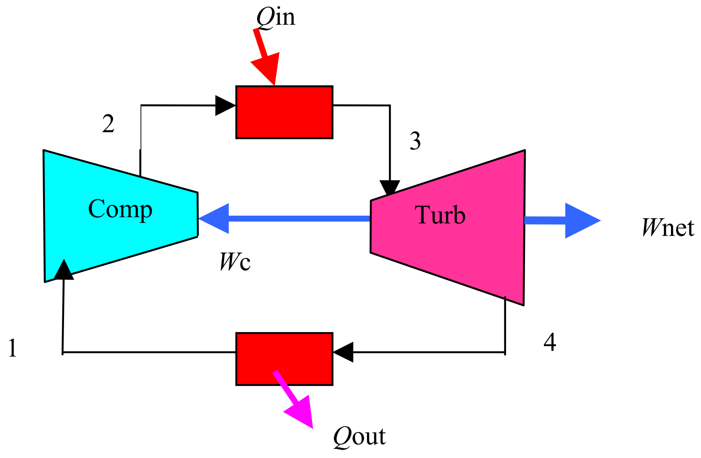
We assume the working fluid is air and the specific heats are constant and will consider the cold-air-standard cycle.

Brayton cycle



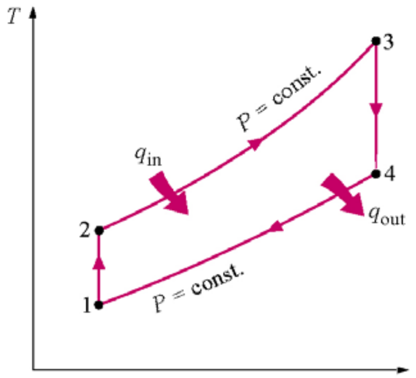
The closed cycle gas-turbine engine

Closed Brayton cycle

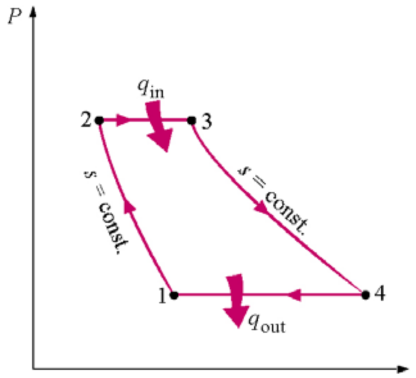


Process	Description
1-2	Isentropic compression (in a compressor)
2-3	Constant pressure heat addition
3-4	Isentropic expansion (in a turbine)
4-1	Constant pressure heat rejection

The *T-s* and *P-v* diagrams are



(a) *T-s* diagram



(b) *P-υ* diagram

Thermal efficiency of the Brayton cycle

$$\eta_{th,\ Brayton} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

Now to find Q_{in} and Q_{out}.

Apply the conservation of energy to process 2-3 for P = constant (no work), steady-flow, and neglect changes in kinetic and potential energies.

$$\begin{aligned} \dot{E}_{in} &= \dot{E}_{out} \\ \dot{m}_2 h_2 + \dot{Q}_{in} &= \dot{m}_3 h_3 \end{aligned}$$

The conservation of mass gives

$$\begin{aligned} \dot{m}_{in} &= \dot{m}_{out} \\ \dot{m}_2 &= \dot{m}_3 = \dot{m} \end{aligned}$$

For constant specific heats, the heat added per unit mass flow is

$$\begin{aligned} \dot{Q}_{in} &= \dot{m}(h_3 - h_2) \\ \dot{Q}_{in} &= \dot{m}C_p(T_3 - T_2) \\ q_{in} &= \frac{\dot{Q}_{in}}{\dot{m}} = C_p(T_3 - T_2) \end{aligned}$$

The conservation of energy for process 4-1 yields for constant specific heats (let's take a minute for you to get the following result)

$$\begin{aligned} \dot{Q}_{out} &= \dot{m}(h_4 - h_1) \\ \dot{Q}_{out} &= \dot{m}C_p(T_4 - T_1) \\ q_{out} &= \frac{\dot{Q}_{out}}{\dot{m}} = C_p(T_4 - T_1) \end{aligned}$$

The thermal efficiency becomes

$$\eta_{th,\ Brayton} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

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

$$\eta_{th,\ Brayton} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$= 1 - \frac{T_1(T_4 / T_1 - 1)}{T_2(T_3 / T_2 - 1)}$$

Recall processes 1-2 and 3-4 are isentropic, so

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \quad and \quad \frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k}$$

Since $P_3 = P_2$ and $P_4 = P_1$, we see that

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

The Brayton cycle efficiency becomes

$$\eta_{th,\ Brayton} = 1 - \frac{T_1}{T_2}$$

Is this the same as the Carnot cycle efficiency?

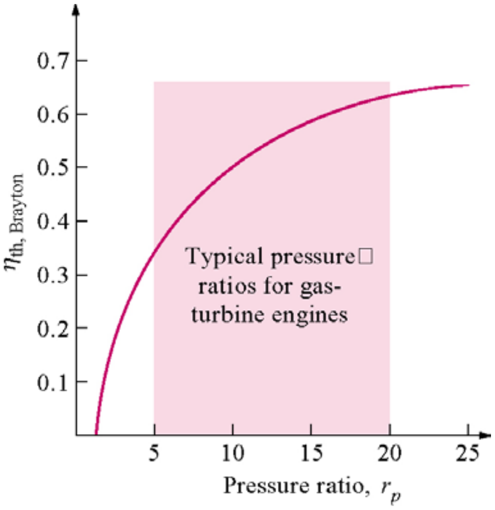
Since process 1-2 is isentropic,

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

$$\frac{T_1}{T_2} = \frac{1}{r_p^{(k-1)/k}}$$

where the pressure ratio is $r_p = P_2/P_1$ and

$$\eta_{th,\ Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}}$$



Extra Assignment

Evaluate the Brayton cycle efficiency by determining the net work directly from the turbine work and the compressor work. Compare your result with the above expression. Note that this approach does not require the closed cycle assumption.

Example 8-2

The ideal air-standard Brayton cycle operates with air entering the compressor at 95 kPa, 22°C. The pressure ratio r_p is 6:1 and the air leaves the heat addition process at 1100 K. Determine the compressor work and the turbine work per unit mass flow, the cycle efficiency, the back work ratio, and compare the compressor exit temperature to the turbine exit temperature. Assume constant properties.

Apply the conservation of energy for steady-flow and neglect changes in kinetic and potential energies to process 1-2 for the compressor. Note that the compressor is isentropic.

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}_1 h_1 + \dot{W}_{comp} = \dot{m}_2 h_2$$

The conservation of mass gives

$$\dot{m}_{in} = \dot{m}_{out}$$
$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

For constant specific heats, the compressor work per unit mass flow is

$$\dot{W}_{comp} = \dot{m}(h_2 - h_1)$$
$$\dot{W}_{comp} = \dot{m}C_p(T_2 - T_1)$$
$$w_{comp} = \frac{\dot{W}_{comp}}{\dot{m}} = C_p(T_2 - T_1)$$

Since the compressor is isentropic

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = r_p^{(k-1)/k}$$
$$\begin{aligned} T_2 &= T_1 r_p^{(k-1)/k} \\ &= (22+273)K(6)^{(1.4-1)/1.4} \\ &= 492.5\,K \end{aligned}$$

$$\begin{aligned} w_{comp} &= C_p(T_2 - T_1) \\ &= 1.005\frac{kJ}{kg \cdot K}(492.5 - 295)K \\ &= 198.15\frac{kJ}{kg} \end{aligned}$$

The conservation of energy for the turbine, process 3-4, yields for constant specific heats (let’s take a minute for you to get the following result)

$$\dot{W}_{turb} = \dot{m}(h_3 - h_4)$$
$$\dot{W}_{turb} = \dot{m}C_p(T_3 - T_4)$$
$$w_{turb} = \frac{\dot{W}_{turb}}{\dot{m}} = C_p(T_3 - T_4)$$

Since process 3-4 is isentropic

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{(k-1)/k}$$

Since $P_3 = P_2$ and $P_4 = P_1$, we see that

$$\frac{T_4}{T_3} = \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$= 1100K\left(\frac{1}{6}\right)^{(1.4-1)/1.4}$$

$$= 659.1\,K$$

$$w_{turb} = C_p(T_3 - T_4) = 1.005\frac{kJ}{kg \cdot K}(1100 - 659.1)K$$

$$= 442.5\frac{kJ}{kg}$$

We have already shown the heat supplied to the cycle per unit mass flow in process 2-3 is

$$\dot{m}_2 = \dot{m}_3 = \dot{m}$$

$$\dot{m}_2 h_2 + \dot{Q}_{in} = \dot{m}_3 h_3$$

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = h_3 - h_2$$

$$= C_p (T_3 - T_2) = 1.005 \frac{kJ}{kg \cdot K} (1100 - 492.5) K$$

$$= 609.6 \frac{kJ}{kg}$$

The net work done by the cycle is

$$w_{net} = w_{turb} - w_{comp}$$

$$= (442.5 - 198.15) \frac{kJ}{kg}$$

$$= 244.3 \frac{kJ}{kg}$$

The cycle efficiency becomes

$$\eta_{th,\, Brayton} = \frac{w_{net}}{q_{in}}$$

$$= \frac{244.3 \frac{kJ}{kg}}{609.6 \frac{kJ}{kg}} = 0.40 \quad or \quad 40\%$$

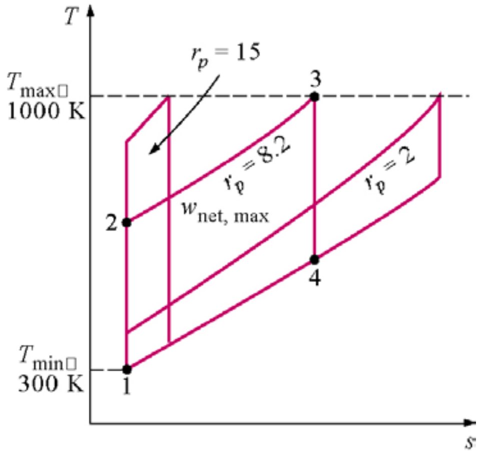
The back work ratio is defined as

$$BWR = \frac{w_{in}}{w_{out}} = \frac{w_{comp}}{w_{turb}}$$

$$= \frac{198.15\frac{kJ}{kg}}{442.5\frac{kJ}{kg}} = 0.448$$

Note that $T_4 = 659.1$ K > $T_2 = 492.5$ K, or the turbine outlet temperature is greater than the compressor exit temperature. Can this result be used to improve the cycle efficiency?

What happens to η_{th}, w_{in}/w_{out}, and w_{net} as the pressure ratio r_p is increased?



Let's take a closer look at the effect of the pressure ratio on the net work done.

$$\begin{aligned} w_{net} &= w_{turb} - w_{comp} \\ &= C_p(T_3 - T_4) - C_p(T_2 - T_1) \\ &= C_p T_3(1 - T_4 / T_3) - C_p T_1(T_2 / T_1 - 1) \\ &= C_p T_3(1 - \frac{1}{r_p{}^{(k-1)/k}}) - C_p T_1(r_p{}^{(k-1)/k} - 1) \end{aligned}$$

Note that the net work is **zero** when

$$r_p = 1 \quad and \quad r_p = \left(\frac{T_3}{T_1}\right)^{(k-1)/k}$$

For fixed T_3 and T_1, the pressure ratio that makes the work a maximum is obtained from:

$$\frac{dw_{net}}{dr_p} = 0$$

This is easier to do if we let $X = r_p{}^{(k\text{-}1)/k}$

$$w_{net} = C_p T_3(1 - \frac{1}{X}) - C_p T_1(X - 1)$$

$$\frac{dw_{net}}{dX} = C_p T_3[0 - (-1)X^{-2}] - C_p T_1[1 - 0] = 0$$

Solving for X

$$X^2 = \frac{T_3}{T_1} = \left(r_p\right)^{2(k-1)/k}$$

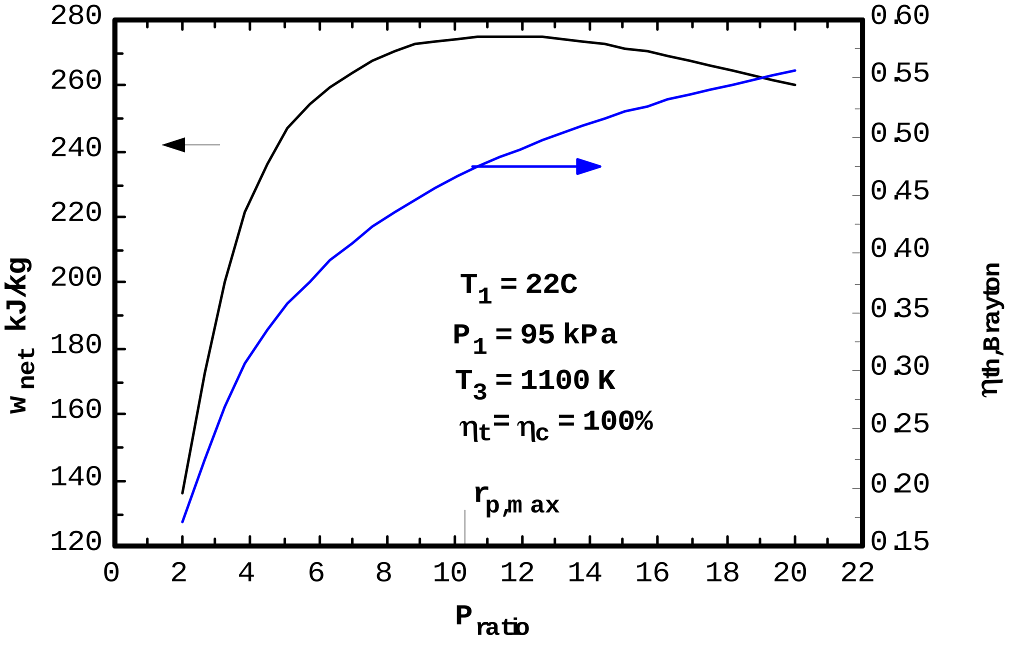
Then, the r_p that makes the work a maximum for the constant property case and fixed T_3 and T_1 is

$$r_{p,\,\text{max work}} = \left(\frac{T_3}{T_1}\right)^{k/[2(k-1)]}$$

For the ideal Brayton cycle, show that the following results are true.

- When $r_p = r_{p,\,\text{max work}}$, $T_4 = T_2$
- When $r_p < r_{p,\,\text{max work}}$, $T_4 > T_2$
- When $r_p > r_{p,\,\text{max work}}$, $T_4 < T_2$

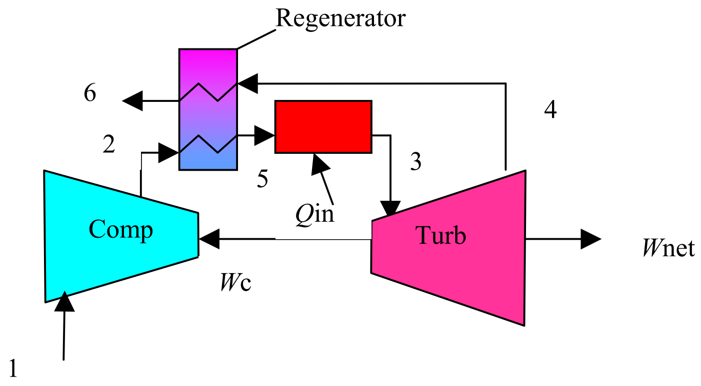
The following is a plot of net work per unit mass and the efficiency for the above example as a function of the pressure ratio.

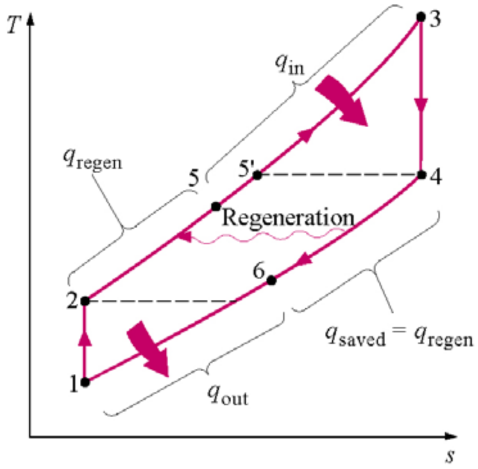


Regenerative Brayton Cycle

For the Brayton cycle, the turbine exhaust temperature is greater than the compressor exit temperature. Therefore, a heat exchanger can be placed between the hot gases leaving the turbine and the cooler gases leaving the compressor. This heat exchanger is called a regenerator or recuperator. The sketch of the regenerative Brayton cycle is shown below.

Regenerative Brayton cycle





We define the regenerator effectiveness ε_{regen} as the ratio of the heat transferred to the compressor gases in the regenerator to the maximum possible heat transfer to the compressor gases.

$$q_{regen,\,act} = h_5 - h_2$$

$$q_{regen,\,\max} = h_{5'} - h_2 = h_4 - h_2$$

$$\varepsilon_{regen} = \frac{q_{regen,\,act}}{q_{regen,\,\max}} = \frac{h_5 - h_2}{h_4 - h_2}$$

For ideal gases using the cold-air-standard assumption with constant specific heats, the regenerator effectiveness becomes

$$\varepsilon_{regen} \cong \frac{T_5 - T_2}{T_4 - T_2}$$

Using the closed cycle analysis and treating the heat addition and heat rejection as steady-flow processes, the regenerative cycle thermal efficiency is

$$\begin{aligned}\eta_{th,\,regen} &= 1 - \frac{q_{out}}{q_{in}}\\ &= 1 - \frac{h_6 - h_1}{h_3 - h_5}\end{aligned}$$

Notice that the heat transfer occurring within the regenerator is **not** included in the efficiency calculation because this energy is not a heat transfer across the cycle boundary.

Assuming an ideal regenerator $\varepsilon_{regen} = 1$ and constant specific heats, the thermal efficiency becomes (take the time to show this on your own)

$$\eta_{th,\,regen} = 1 - \frac{T_1}{T_3}\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$= 1 - \frac{T_1}{T_3}(r_p)^{(k-1)/k}$$

When does the efficiency of the air-standard Brayton cycle equal the efficiency of the air-standard regenerative Brayton cycle? If we set $\eta_{\text{th, Brayton}} = \eta_{\text{th, regen}}$ then

$$\eta_{th,\,Brayton} = \eta_{th,\,regen}$$

$$1 - \frac{1}{(r_p)^{(k-1)/k}} = 1 - \frac{T_1}{T_3}(r_p)^{(k-1)/k}$$

$$r_p = \left(\frac{T_3}{T_1}\right)^{k/[2(k-1)]}$$

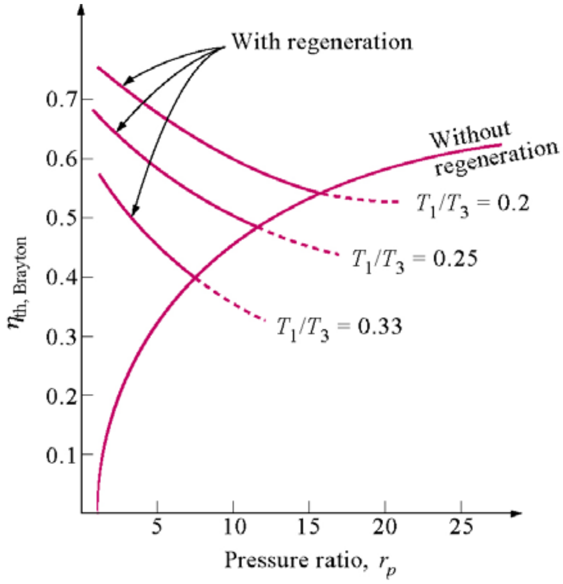
Recall that this is the pressure ratio that maximizes the net work for the simple Brayton cycle and makes $T_4 = T_2$. What happens if the regenerative Brayton cycle operates at a pressure ratio larger than this value?

For fixed T_3 and T_1, pressure ratios greater than this value cause T_4 to be less than T_2, and the regenerator is not effective.

What happens to the net work when a regenerator is added?

What happens to the heat supplied when a regenerator is added?

The following shows a plot of the regenerative Brayton cycle efficiency as a function of the pressure ratio and minimum to maximum temperature ratio, T_1/T_3.



Example 8-3: Regenerative Brayton Cycle

Air enters the compressor of a regenerative gas-turbine engine at 100 kPa and 300 K and is compressed to 800 kPa. The regenerator has an effectiveness of 65 percent, and the air enters the turbine at 1200 K. For a compressor efficiency of 75 percent and a turbine efficiency of 86 percent, determine

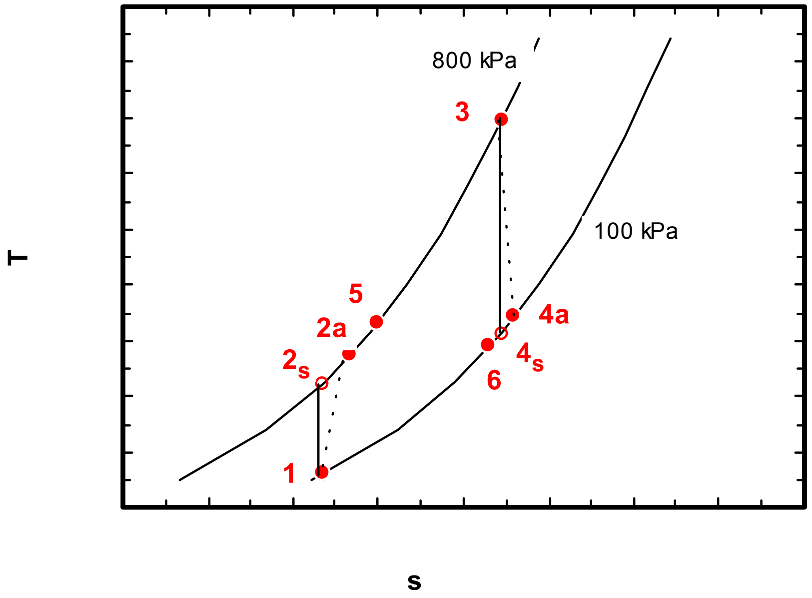
(a) The heat transfer in the regenerator.
(b) The back work ratio.
(c) The cycle thermal efficiency.

Compare the results for the above cycle with the ones listed below that have the same common data as required.

(a) The actual cycle with no regeneration, $\varepsilon = 0$.
(b) The actual cycle with ideal regeneration, $\varepsilon = 1.0$.
(b) The ideal cycle with regeneration, $\varepsilon = 0.65$.
(d) The ideal cycle with no regeneration, $\varepsilon = 0$.
(e) The ideal cycle with ideal regeneration, $\varepsilon = 1.0$.

We assume air is an ideal gas with constant specific heats, that is, we use the cold-air-standard assumption.

The cycle schematic is the same as above and the *T*-*s* diagram showing the effects of compressor and turbine efficiencies is below.



T-s Diagram for Gas Turbine with Regeneration

Summary of Results

Cycle type	Actual	Actual	Actual	Ideal	Ideal	Ideal
$\varepsilon_{\text{regen}}$	0.00	0.65	1.00	0.00	0.65	1.00
η_{comp}	0.75	0.75	0.75	1.00	1.00	1.00
η_{turb}	0.86	0.86	0.86	1.00	1.00	1.00
q_{in} kJ/kg	578.3	504.4	464.6	659.9	582.2	540.2
w_{comp} kJ/kg	326.2	326.2	326.2	244.6	244.6	244.6
w_{turb} kJ/kg	464.6	464.6	464.6	540.2	540.2	540.2
$w_{\text{comp}}/w_{\text{turb}}$	0.70	0.70	0.70	0.453	0.453	0.453
η_{th}	24.0%	27.5%	29.8%	44.8%	50.8%	54.7%

Compressor analysis

The isentropic temperature at compressor exit is

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$= 300K\left(\frac{800kPa}{100kPa}\right)^{(1.4-1)/1.4} = 543.4K$$

To find the actual temperature at compressor exit, T_{2a}, we apply the compressor efficiency

$$\eta_{comp} = \frac{w_{isen,comp}}{w_{act,comp}} = \frac{h_{2s}-h_1}{h_{2a}-h_1} \cong \frac{T_{2s}-T_1}{T_{2a}-T_1}$$

$$T_{2a} = T_1 + \frac{1}{\eta_{comp}}(T_{2s}-T_1)$$

$$= 300K + \frac{1}{0.75}(543.4-300)K$$

$$= 624.6K$$

Since the compressor is adiabatic and has steady-flow

$$w_{comp} = h_{2a} - h_1 = C_p(T_{2a} - T_1)$$

$$= 1.005\frac{kJ}{kg\cdot K}(624.6-300)K = 326.2\frac{kJ}{kg}$$

Turbine analysis

The conservation of energy for the turbine, process 3-4, yields for constant specific heats (let's take a minute for you to get the following result)

$$\dot{W}_{turb} = \dot{m}(h_3 - h_{4a})$$
$$\dot{W}_{turb} = \dot{m}C_p(T_3 - T_{4a})$$
$$w_{turb} = \frac{\dot{W}_{turb}}{\dot{m}} = C_p(T_3 - T_{4a})$$

Since $P_3 = P_2$ and $P_4 = P_1$, we can find the isentropic temperature at the turbine exit.

$$\frac{T_{4s}}{T_3} = \left(\frac{P_4}{P_3}\right)^{(k-1)/k}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k}$$

$$= 1200K\left(\frac{100kPa}{800kPa}\right)^{(1.4-1)/1.4} = 662.5K$$

To find the actual temperature at turbine exit, T_{4a}, we apply the turbine efficiency.

$$\eta_{turb} = \frac{w_{act,turb}}{w_{isen,turb}} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \cong \frac{T_3 - T_{4a}}{T_3 - T_{4s}}$$

$$\begin{aligned} T_{4a} &= T_3 - \eta_{turb}(T_3 - T_{4s}) \\ &= 1200K - 0.86(1200 - 662.5)K \\ &= 737.7K > T_{2a} \end{aligned}$$

The turbine work becomes

$$\begin{aligned} w_{turb} &= h_3 - h_{4a} = C_p(T_3 - T_{4a}) \\ &= 1.005\frac{kJ}{kg \cdot K}(1200 - 737.7)K \\ &= 464.6\frac{kJ}{kg} \end{aligned}$$

The back work ratio is defined as

$$\begin{aligned} BWR &= \frac{w_{in}}{w_{out}} = \frac{w_{comp}}{w_{turb}} \\ &= \frac{326.2\frac{kJ}{kg}}{464.6\frac{kJ}{kg}} = 0.70 \end{aligned}$$

Regenerator analysis

To find T_5, we apply the regenerator effectiveness.

$$\varepsilon_{regen} \cong \frac{T_5 - T_{2a}}{T_{4a} - T_{2a}}$$

$$\begin{aligned} T_5 &= T_{2a} + \varepsilon_{regen}(T_{4a} - T_{2a}) \\ &= 624.6K + 0.65(737.7 - 624.6)K \\ &= 698.1K \end{aligned}$$

To find the heat transferred from the turbine exhaust gas to the compressor exit gas, apply the steady-flow conservation of energy to the compressor gas side of the regenerator.

$$\begin{aligned} \dot{m}_{2a}h_{2a} + \dot{Q}_{regen} &= \dot{m}_5 h_5 \\ \dot{m}_{2a} &= \dot{m}_5 = \dot{m} \\ q_{regen} &= \frac{\dot{Q}_{regen}}{\dot{m}} = h_5 - h_{2a} \\ &= C_p(T_5 - T_{2a}) \\ &= 1.005\frac{kJ}{kg \cdot K}(698.1 - 624.6)K \\ &= 73.9\frac{kJ}{kg} \end{aligned}$$

Using q_{regen}, we can determine the turbine exhaust gas temperature at the regenerator exit.

$$\dot{m}_{4a}h_{4a} = \dot{Q}_{regen} + \dot{m}_6 h_6$$

$$\dot{m}_{4a} = \dot{m}_6 = \dot{m}$$

$$q_{regen} = \frac{\dot{Q}_{regen}}{\dot{m}} = h_{4a} - h_6 = C_p(T_{4a} - T_6)$$

$$T_6 = T_{4a} - \frac{q_{regen}}{C_p} = 737.7K - \frac{73.9\frac{kJ}{kg}}{1.005\frac{kJ}{kg \cdot K}}$$

$$= 664.2K$$

Heat supplied to cycle

Apply the steady-flow conservation of energy to the heat exchanger for process 5-3. We obtain a result similar to that for the simple Brayton cycle.

$$q_{in} = h_3 - h_5 = C_p(T_3 - T_5)$$

$$= 1.005\frac{kJ}{kg \cdot K}(1200 - 698.1)K$$

$$= 504.4\frac{kJ}{kg}$$

Cycle thermal efficiency

The net work done by the cycle is

$$w_{net} = w_{turb} - w_{comp}$$

$$= (464.6 - 326.2)\frac{kJ}{kg} = 138.4\frac{kJ}{kg}$$

The cycle efficiency becomes

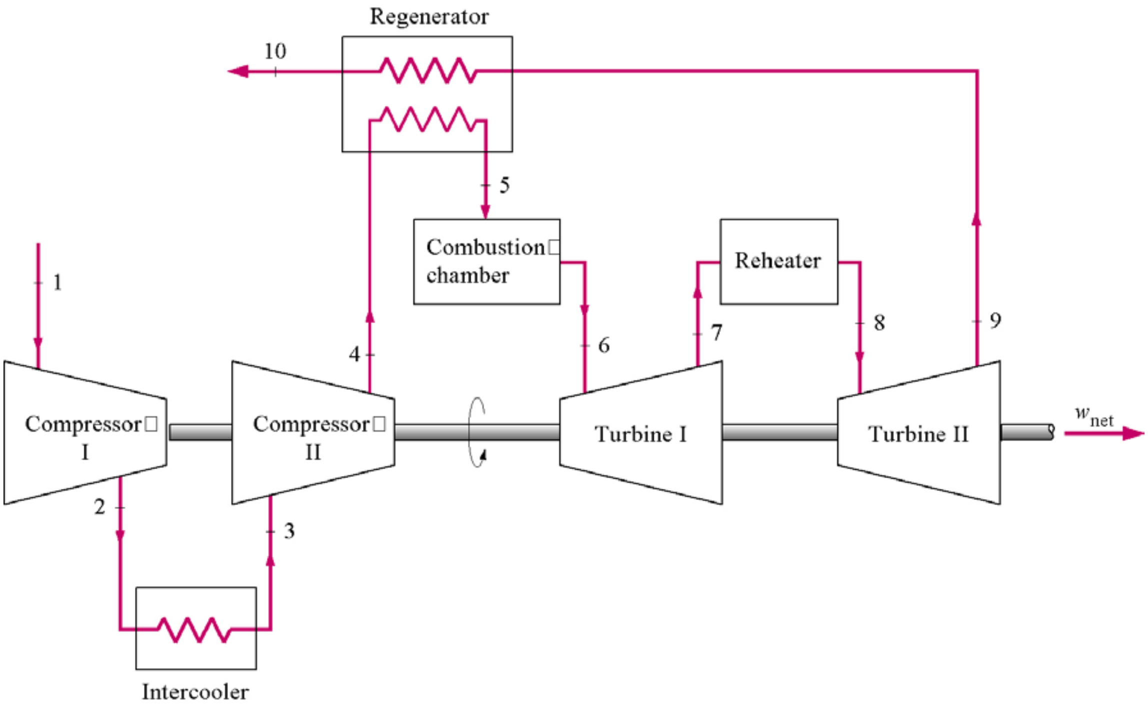
$$\eta_{th,\,Brayton} = \frac{w_{net}}{q_{in}}$$

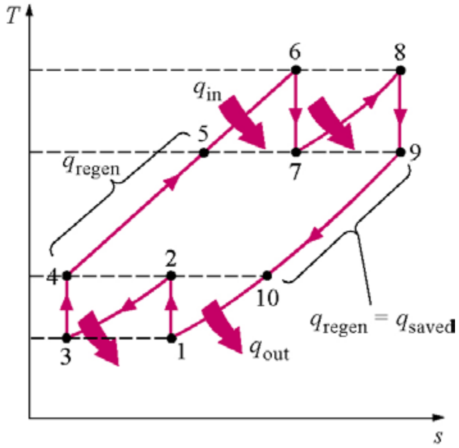
$$= \frac{138.4\,\frac{kJ}{kg}}{504.4\,\frac{kJ}{kg}} = 0.274 \quad or \quad 27.4\%$$

You are encouraged to complete the calculations for the other values found in the summary table.

Other Ways to Improve Brayton Cycle Performance

Intercooling and reheating are two important ways to improve the performance of the Brayton cycle with regeneration.





Intercooling

When using multistage compression, cooling the working fluid between the stages will reduce the amount of compressor work required. The compressor work is reduced because cooling the working fluid reduces the average specific volume of the fluid and thus reduces the amount of work on the fluid to achieve the given pressure rise.

To determine the intermediate pressure at which intercooling should take place to minimize the compressor work, we follow the approach shown in Chapter 6.

For the adiabatic, steady-flow compression process, the work input to the compressor per unit mass is

$$w_{comp} = \int_1^4 v\,dP = \int_1^2 v\,dP + \int_2^3 \cancelto{0}{v\,dP} + \int_3^4 v\,dP$$

For the isentropic compression process

$$
\begin{aligned}
w_{comp} &= \frac{k}{k\text{-}1}(P_2v_2 - P_1v_1) + \frac{k}{k\text{-}1}(P_4v_4 - P_3v_3) \\
&= \frac{k}{k\text{-}1}R(T_2 - T_1) + \frac{kR}{k\text{-}1}(T_4 - T_3) \\
&= \frac{k}{k\text{-}1}R\left[T_1(T_2/T_1 - 1) + T_3(T_4/T_3 - 1)\right] \\
&= \frac{k}{k\text{-}1}R\left[T_1\left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right) + T_3\left(\left(\frac{P_4}{P_3}\right)^{(k-1)/k} - 1\right)\right]
\end{aligned}
$$

Notice that the fraction $kR/(k\text{-}1) = C_p$.

$$
w_{comp} = C_p\left[T_1\left(\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right) + T_3\left(\left(\frac{P_4}{P_3}\right)^{(k-1)/k} - 1\right)\right]
$$

Can you obtain this relation another way? Hint: apply the first law to processes 1-4.

For two-stage compression, let's assume that intercooling takes place at constant pressure and the gases can be cooled to the inlet temperature for the compressor, such that $P_3 = P_2$ and $T_3 = T_1$.

The total work supplied to the compressor becomes

$$w_{comp} = C_p T_1 \left[\left(\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right) + \left(\left(\frac{P_4}{P_2} \right)^{(k-1)/k} - 1 \right) \right]$$

$$= C_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} + \left(\frac{P_4}{P_2} \right)^{(k-1)/k} - 2 \right]$$

To find the unknown pressure P_2 that gives the minimum work input for fixed compressor inlet conditions T_1, P_1, and exit pressure P_4, we set

$$\frac{dw_{comp}(P_2)}{dP_2} = 0$$

This yields

$$P_2 = \sqrt{P_1 P_4}$$

or, the pressure ratios across the two compressors are equal.

$$\frac{P_2}{P_1} = \frac{P_4}{P_2} = \frac{P_4}{P_3}$$

Intercooling is almost always used with regeneration. During intercooling the compressor exit temperature is reduced; therefore, more heat must be supplied in the heat addition process. Regeneration can make up part of the required heat transfer.

To supply only compressed air, using intercooling requires less work input. The next time you go to a home supply store where air compressors are sold, check the larger air compressors to see if intercooling is used. For the larger air compressors, the compressors are made of two piston-cylinder chambers. The intercooling heat exchanger may be only a pipe with a few attached fins that connects the large piston-cylinder chamber with the smaller piston-cylinder chamber.

Extra Assignment

Obtain the expression for the compressor total work by applying conservation of energy directly to the low- and high-pressure compressors.

Reheating

When using multistage expansion through two or more turbines, reheating between stages will increase the net work done (it also increases the required heat input). The regenerative Brayton cycle with reheating is shown above.

The optimum intermediate pressure for reheating is the one that maximizes the turbine work. Following the development given above for intercooling and assuming reheating to the high-pressure turbine inlet temperature in a constant pressure steady-flow process, we can show the optimum reheat pressure to be

$$P_7 = \sqrt{P_6 P_9}$$

or the pressure ratios across the two turbines are equal.

$$\frac{P_6}{P_7} = \frac{P_7}{P_9} = \frac{P_8}{P_9}$$

The specific objectives of the lesson:

This lesson discusses various gas cycle refrigeration systems based on air, namely:

1. Reverse Carnot cycle & its limitations (*Section 9.4*)
2. Reverse Brayton cycle – Ideal & Actual (*Section 9.5*)
3. Aircraft refrigeration cycles, namely Simple system, Bootstrap system, Regenerative system, etc. (*Section 9.6*)

At the end of the lesson the student should be able to:

1. Describe various air cycle refrigeration systems (*Section 9.1-9.6*)
2. State the assumptions made in the analyses of air cycle systems (*Section 9.2*)
3. Show the cycles on T-s diagrams (*Section 9.4-9.6*)
4. Perform various cycle calculations (*Section 9.3-9.6*)
5. State the significance of Dry Air Rated Temperature (*Section 9.6*)

9.1. 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.

9.2. 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.

9.3. 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.,

$$W = m(u_1 - u_2) = mc_v(T_1 - T_2) \tag{9.1}$$

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 $Pv = RT$ and the equation for isentropic process $P_1 v_1^{\gamma} = P_2 v_2^{\gamma}$ the final temperature (T_2) is related to the initial temperature (T_1) and initial and final pressures (P_1 and P_2) by the equation:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{9.2}$$

where γ is the coefficient of isentropic expansion given by:

$$\gamma = \left(\frac{c_p}{c_v}\right) \tag{9.3}$$

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:

$$W = \dot{m}(h_1 - h_2) = \dot{m} c_p(T_1 - T_2) \tag{9.4}$$

The final temperature is related to the initial temperature and initial and final pressures by Eq. (9.2).

9.4. 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-v coordinates. 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

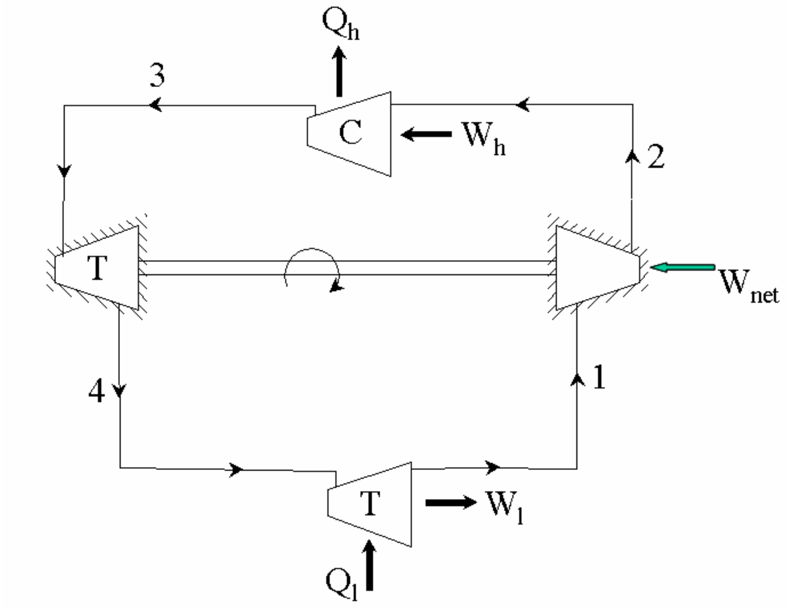


***Fig. 9.1(a)**. Schematic of a reverse Carnot refrigeration system*

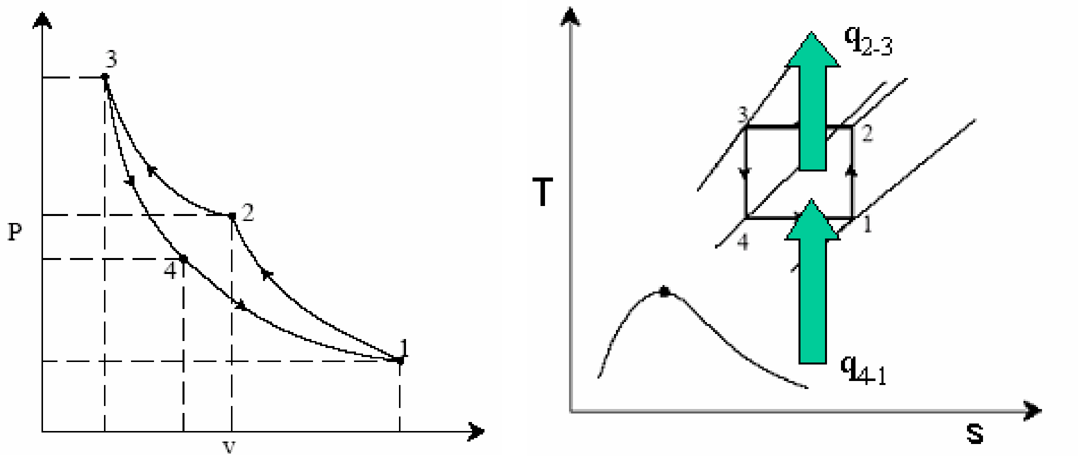


***Fig. 9.1(b)**. Reverse Carnot refrigeration system in P-v and T-s coordinates*

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

$$q_{2-3} = \int_2^3 T.ds = T_h(s_3 - s_2) \tag{9.5a}$$

$$q_{4-1} = \int_4^1 T.ds = T_l(s_1 - s_4) \tag{9.5b}$$

$$s_1 = s_2 \quad \text{and} \quad s_3 = s_4, \quad \text{hence } s_2 - s_3 = s_1 - s_4 \tag{9.6}$$

Applying first law of thermodynamics to the closed cycle,

$$\oint \delta q = (q_{4-1} + q_{2-3}) = \oint \delta w = (w_{2-3} - w_{4-1}) = -w_{net} \tag{9.7}$$

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.

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.

9.5. Ideal reverse Brayton cycle

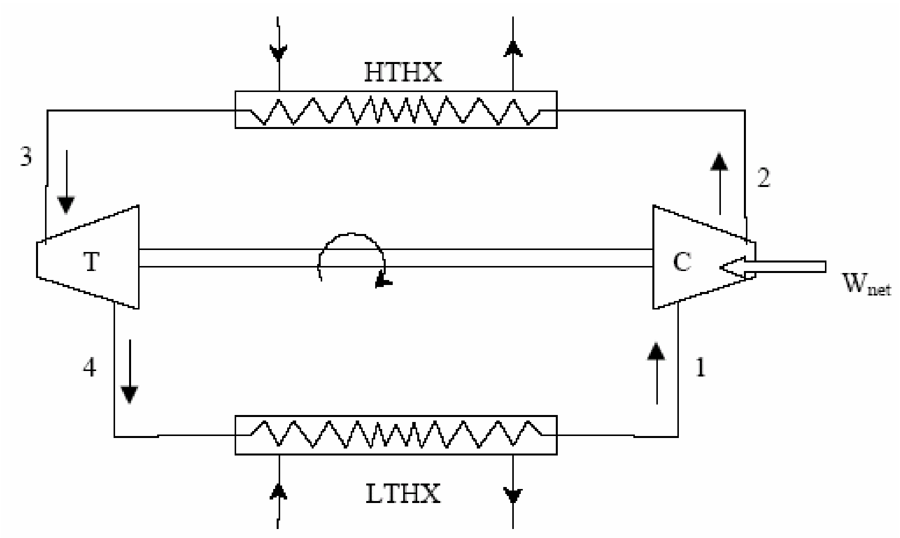


***Fig. 9.2(a)**. Schematic of a closed reverse Brayton cycle*

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 9.2(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

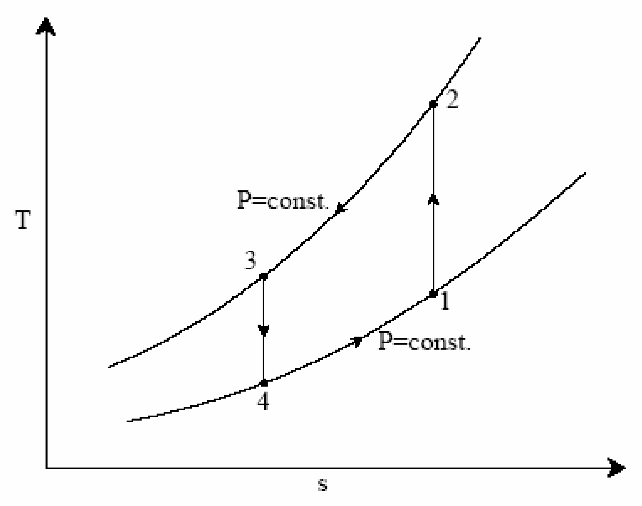


Fig. 9.2(b). *Reverse Brayton cycle in T-s plane*

Process 1-2: Gas at low pressure is compressed isentropically from state 1 to state 2. Applying steady flow energy equation and neglecting changes in kinetic and potential energy, we can write:

$$W_{1-2} = \dot{m}(h_2 - h_1) = \dot{m}\,c_p(T_2 - T_1)$$

$$s_2 = s_1 \tag{9.9}$$

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

where $r_p = (P_2/P_1)$ = pressure ratio

Process 2-3: Hot and high pressure gas flows through a heat exchanger and rejects heat sensibly and isobarically to a heat sink. The enthalpy and temperature of the gas drop during the process due to heat exchange, no work transfer takes place and the entropy of the gas decreases. Again applying steady flow energy equation and second T ds equation:

$$Q_{2-3} = \dot{m}(h_2 - h_3) = \dot{m}\,c_p(T_2 - T_3)$$

$$s_2 - s_3 = c_p \ln\frac{T_2}{T_3} \tag{9.10}$$

$$P_2 = P_3$$

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:

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

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:

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

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

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

Applying 1st law of thermodynamics to the entire cycle:

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

The COP of the reverse Brayton cycle is given by:

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

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

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

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

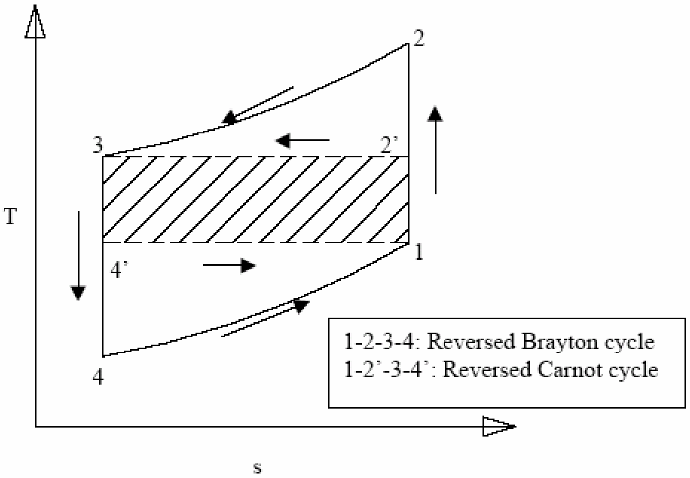


***Fig. 9.3**. Comparison of reverse Carnot and reverse Brayton cycle in T-s plane*

b) COP of Brayton cycle approaches COP of Carnot cycle as T_1 approaches T_4 (thin cycle), however, the specific refrigeration effect $[c_p(T_1\text{-}T_4)]$ 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

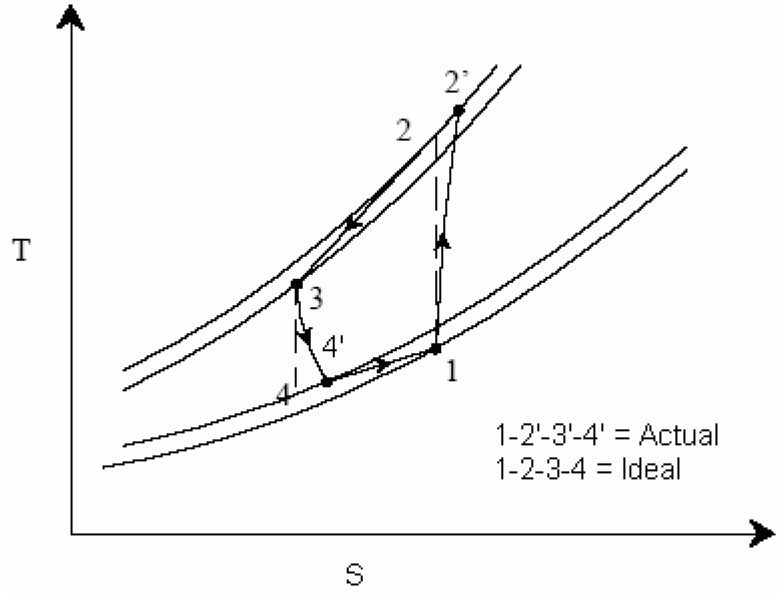


***Fig. 9.4**. Comparison of ideal and actual Brayton cycles T-s plane*

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:

$$W_{1-2,\mathrm{act}} = \frac{W_{1-2,\mathrm{isen}}}{\eta_{\mathrm{c,isen}}} \tag{9.17}$$

$$W_{3-4,\mathrm{act}} = \eta_{\mathrm{t,isen}} W_{3-4,\mathrm{isen}} \tag{9.18}$$

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

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

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

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

$$W_{\mathrm{net,act}} = W_{1-2,\mathrm{act}} - W_{3-4,\mathrm{act}} \tag{9.22}$$

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 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.

9.6. 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.

9.6.1. Simple aircraft refrigeration cycle:

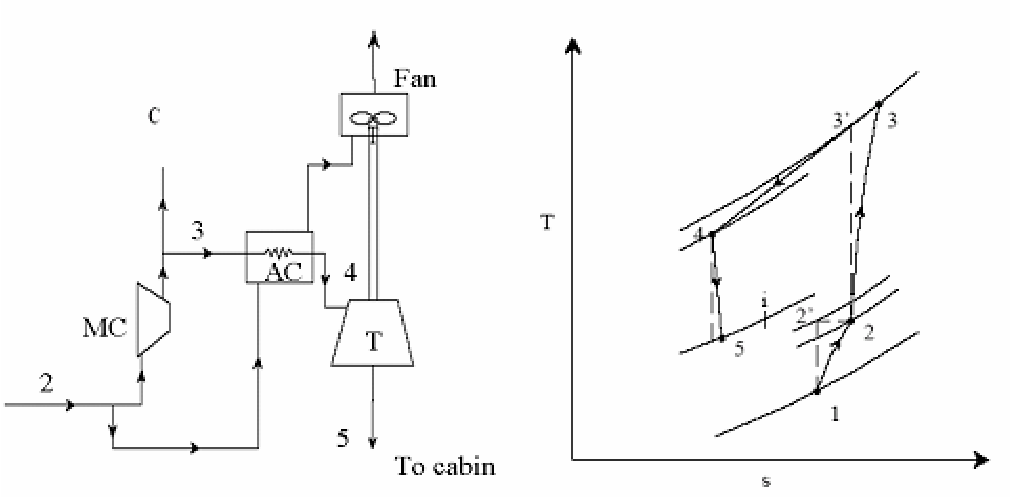


Fig. 9.5. Schematic of a simple aircraft refrigeration cycle

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:

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

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:

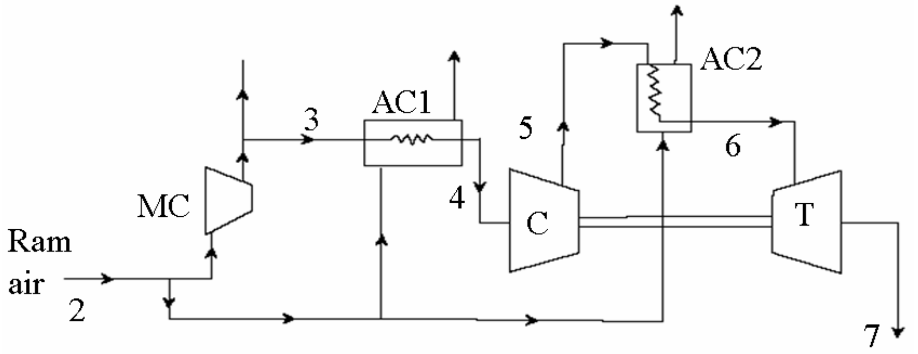
$$\dot{Q} = \dot{m}\, c_p (T_i - T_5) \tag{9.26}$$

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

9.6.2. 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), in stead 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.



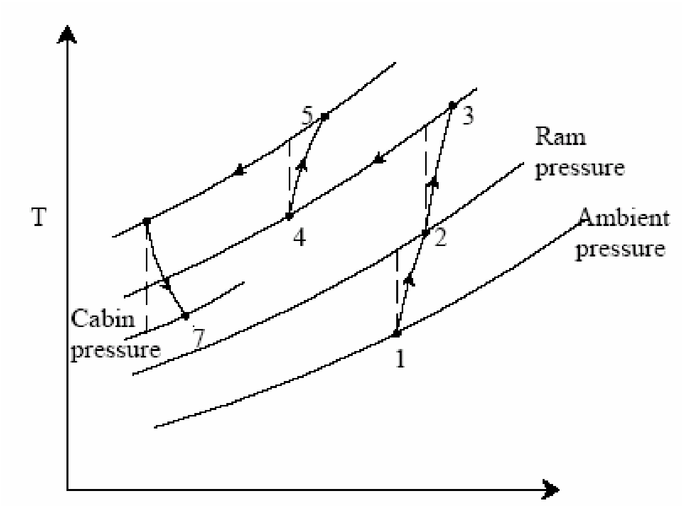


Fig. 9.6. *Schematic of a bootstrap system*

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:

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

where $\dot{m}$ 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

The specific objectives of the lesson:

This lesson discusses the most commonly used refrigeration system, i.e. Vapour compression refrigeration system. The following things are emphasized in detail:

1. The Carnot refrigeration cycle & its practical limitations (*Section 10.3*)
2. The Standard Vapour compression Refrigeration System (*Section 10.4*)
3. Analysis of Standard Vapour compression Refrigeration System (*Section 10.5*)

At the end of the lesson the student should be able to:

1. Analyze and perform cyclic calculations for Carnot refrigeration cycle (*Section 10.3*)
2. State the difficulties with Carnot refrigeration cycle (*Section 10.3*)
3. Analyze and perform cyclic calculations for standard vapour compression refrigeration systems (*Section 10.4*)
4. Perform various cycle calculations for different types of refrigerants (*Section 10.4*)

10.1. Comparison between gas cycles and vapor 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.

10.2. 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.

10.3. 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.

As shown in Fig.10.1(a), 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,

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

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

$$\oint \delta w = w_{3-4} - w_{1-2} = w_T - w_C = -w_{net} \tag{10.1}$$

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

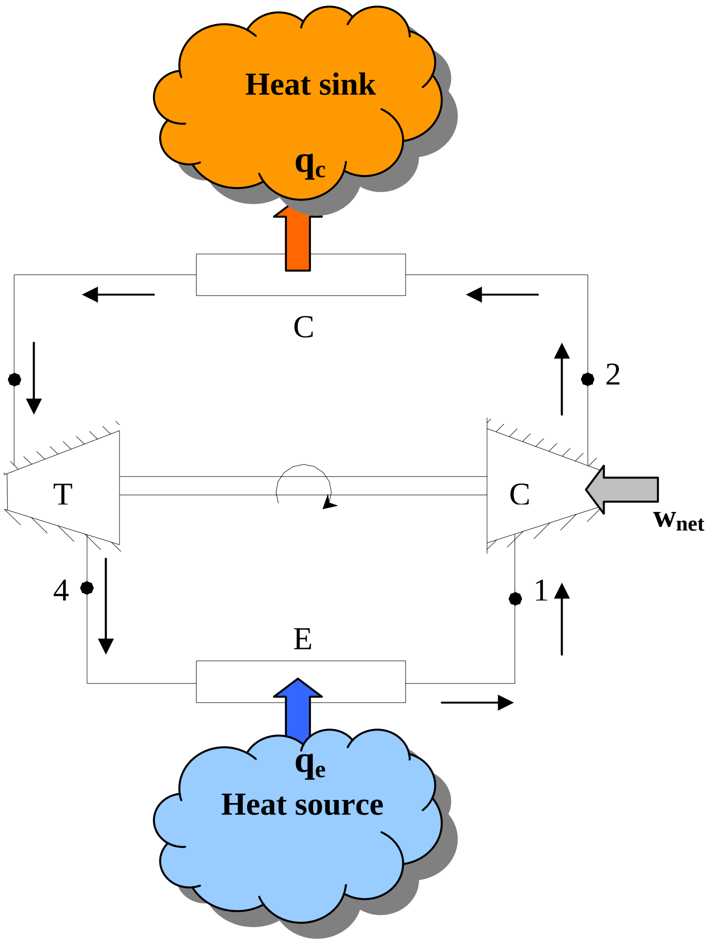


***Fig.10.1(a)**: Schematic of a Carnot refrigeration system*

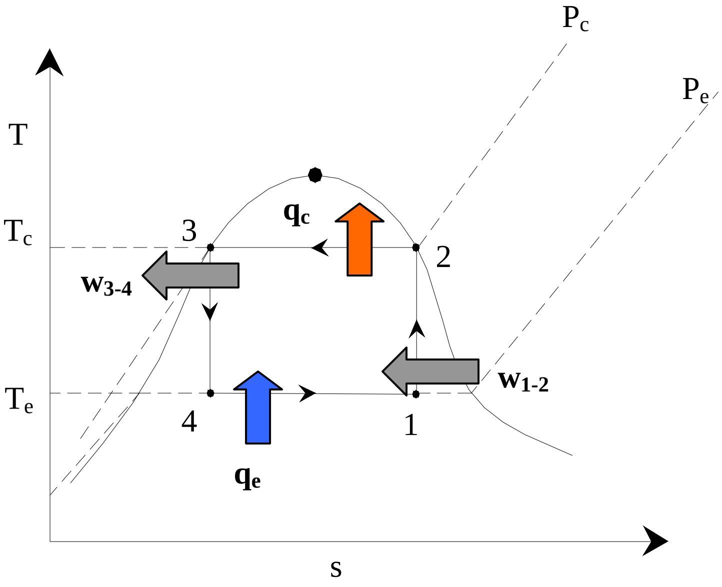


***Fig. 10.1(b)**: Carnot refrigeration cycle on T-s diagram*

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

$$q_c = -q_{2-3} = -\int_2^3 T.ds = T_c(s_2 - s_3) \qquad (10.2)$$

$$q_e = q_{4-1} = \int_4^1 T.ds = T_e(s_1 - s_4) \qquad (10.3)$$

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

$$s_1 = s_2 \text{ and } s_3 = s_4 \qquad (10.4)$$

the Coefficient of Performance (COP) is given by:

$$COP_{Carnot} = \frac{\text{refrigeration effect}}{\text{net work input}} = \frac{q_e}{w_{net}} = \frac{T_e(s_1 - s_4)}{T_c(s_2 - s_3) - T_e(s_1 - s_4)} = \left(\frac{T_e}{T_c - T_e}\right) \qquad (10.5)$$

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 (Lesson 9). 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.

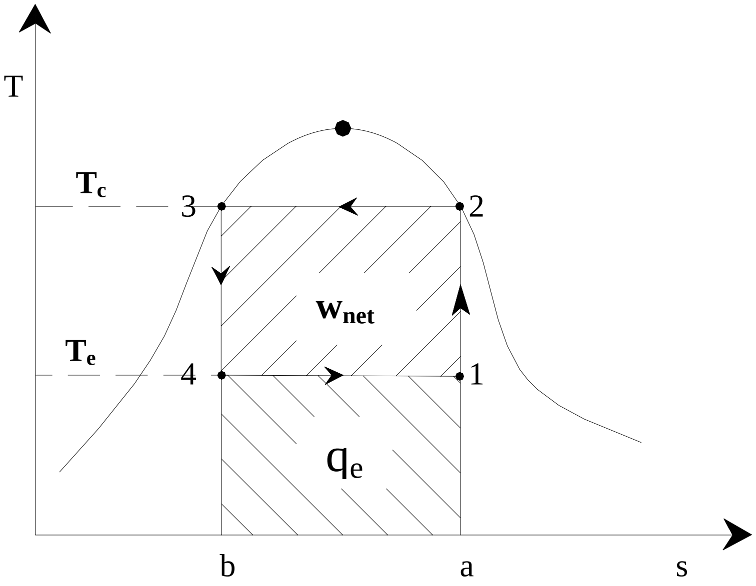


Fig.10.2. *Carnot refrigeration cycle represented in T-s plane*

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.

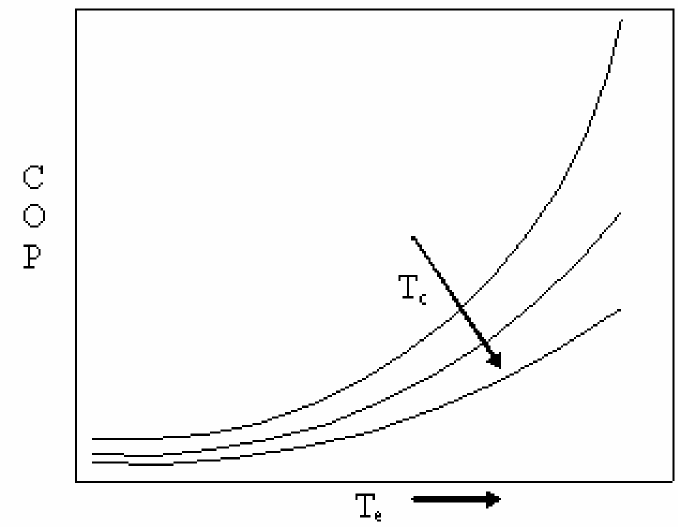


***Fig.10.3**. Effects of evaporator and condenser temperatures on Carnot COP*

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:

$$w_{3-4} = \int_{Pe}^{Pc} v.dP \tag{10.6}$$

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[1].

One way of achieving dry compression in Carnot refrigeration cycle is to have two compressors – one isentropic and one isothermal as shown in Fig.10.4.

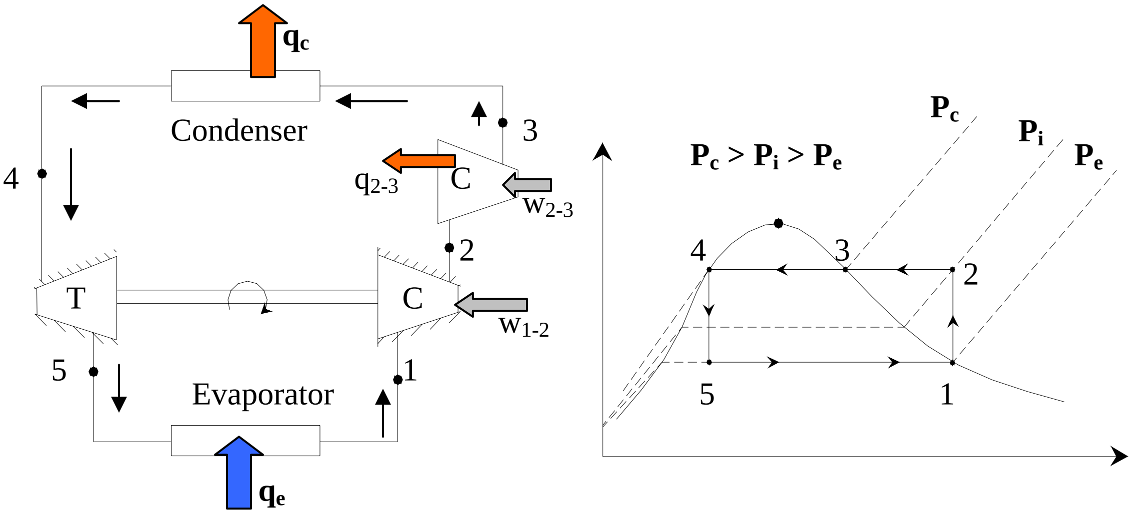


***Fig.10.4**. Carnot refrigeration system with dry compression*

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.

---

[1] However, currently efforts are being made to recover this work of expansion in some refrigeration systems to improve the system efficiency.

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.

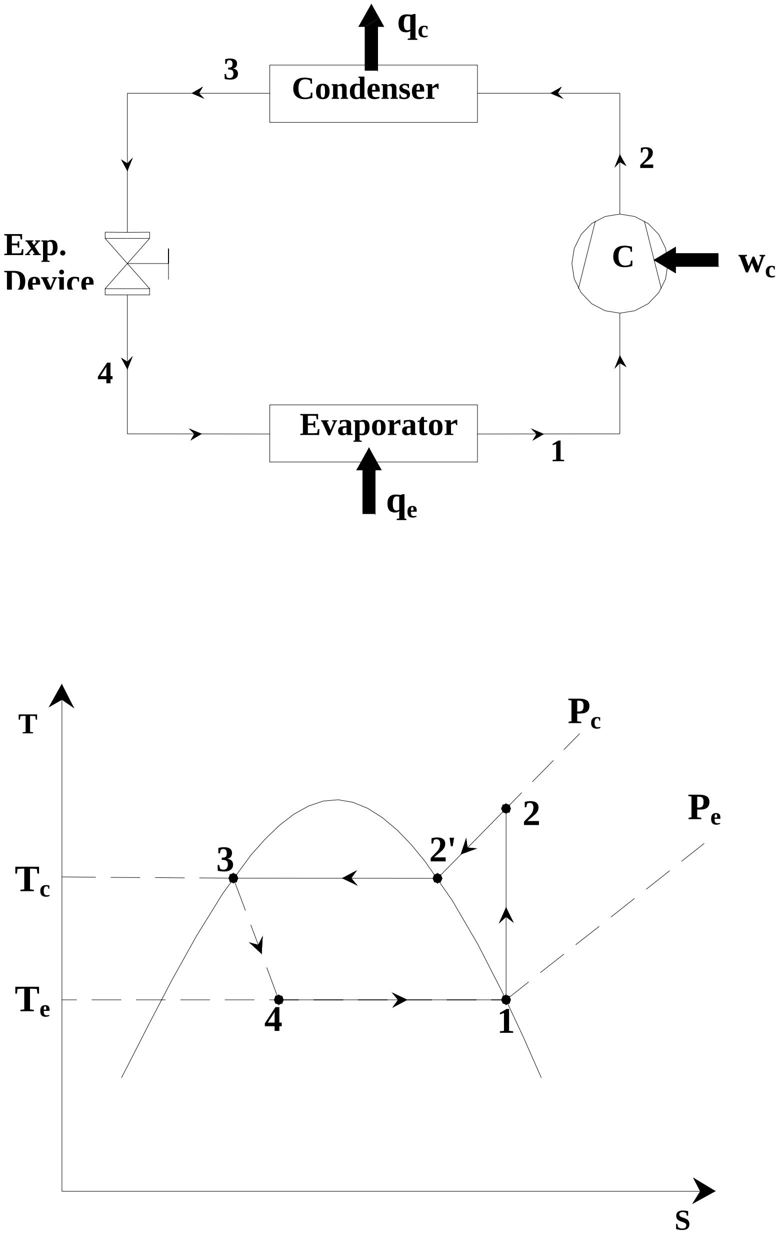


Fig.10.5. *Standard Vapour compression refrigeration system*

10.4. 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 10.6(a) shows comparison between Carnot and standard VCRS in terms of refrigeration effect.

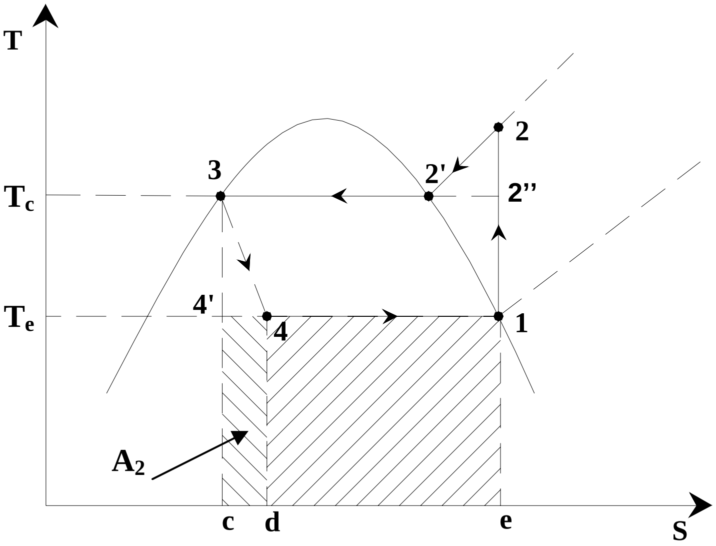


Fig.10.6(a). *Comparison between Carnot and standard VCRS*

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

<u>For Carnot refrigeration cycle (1-2''-3-4'):</u>

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

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

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

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,

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

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.

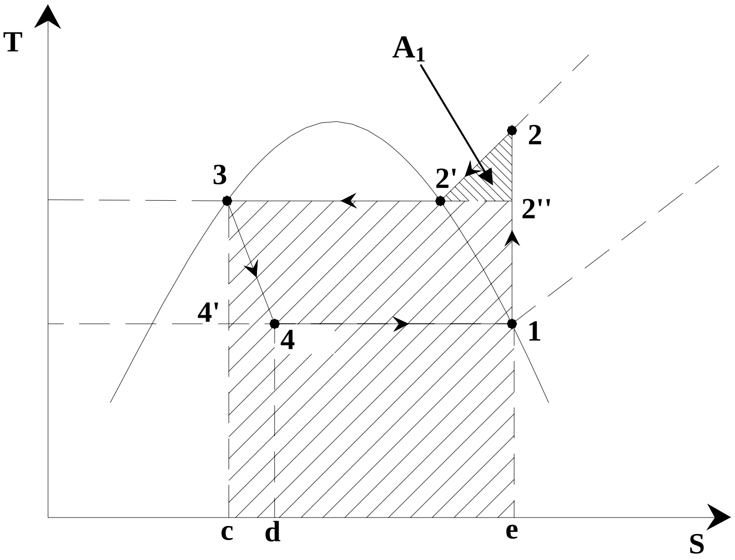


***Fig.10.6(b)**. Comparative evaluation of heat rejection rate of VCRS and Carnot cycle*

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

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

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

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

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 replacement of isothermal heat rejection process of Carnot cycle by isobaric heat rejection in case of VCRS.

Since the heat rejection increases and refrigeration effect reduces when the Carnot cycle is modified to standard VCRS cycle, the net work input to the VCRS increases compared to Carnot cycle. The net work input in case of Carnot and VCRS cycles are given by:

$$w_{net,Carnot} = (q_c - q_e)_{Carnot} = \text{area } 1-2''-3-4'-1 \qquad (10.12)$$

$$w_{net,VCRS} = (q_c - q_e)_{VCRS} = \text{area } 1-2-3-4'-c-d-4-1 \qquad (10.13)$$

As shown in Fig.10.6(c), the increase in net work input in VCRS cycle is given by:

$$w_{net,VCRS} - w_{net,Carnot} = \text{area } 2''-2-2' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2 \qquad (10.14)$$

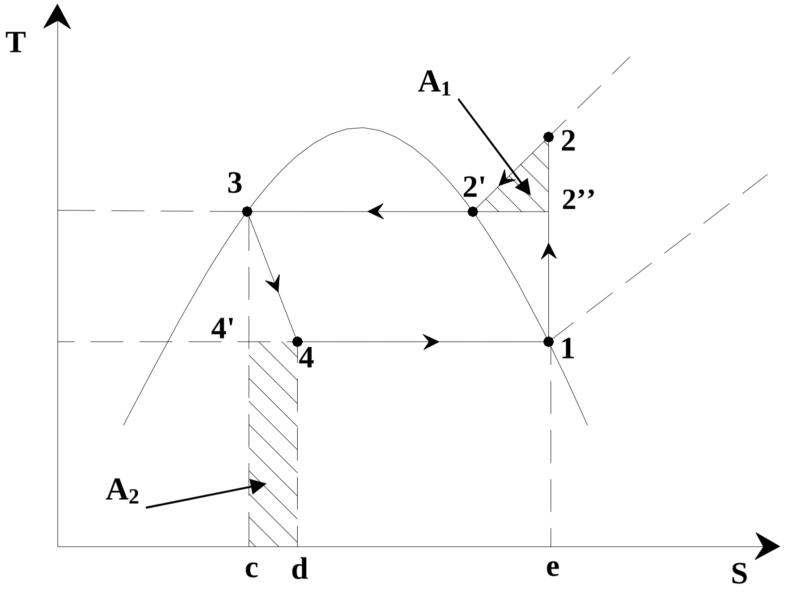


Fig.10.6(c). *Figure illustrating the increase in net work input in VCRS cycle*

To summarize the refrigeration effect and net work input of VCRS cycle are given by:

$$q_{e,VCRS} = q_{e,Carnot} - \text{area } A_2 \qquad (10.15)$$

$$w_{net,VCRS} = w_{net,Carnot} + \text{area } A_1 + \text{area } A_2 \qquad (10.16)$$

The COP of VCRS cycle is given by:

$$COP_{VCRS} = \frac{q_{e,VCRS}}{w_{net,VCRS}} = \frac{q_{e,Carnot} - \text{area } A_2}{w_{net,Carnot} + \text{area } A_1 + \text{area } A_2} \qquad (10.17)$$

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

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

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 $w_{1\text{-}2}$ is given by:

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

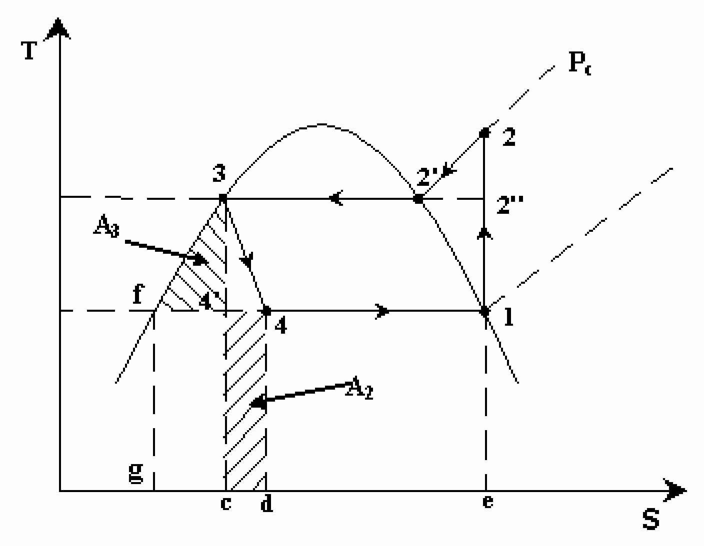


***Fig.10.7**. Figure showing saturated liquid line 3-f coinciding with the constant pressure line*

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;

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

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

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

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.

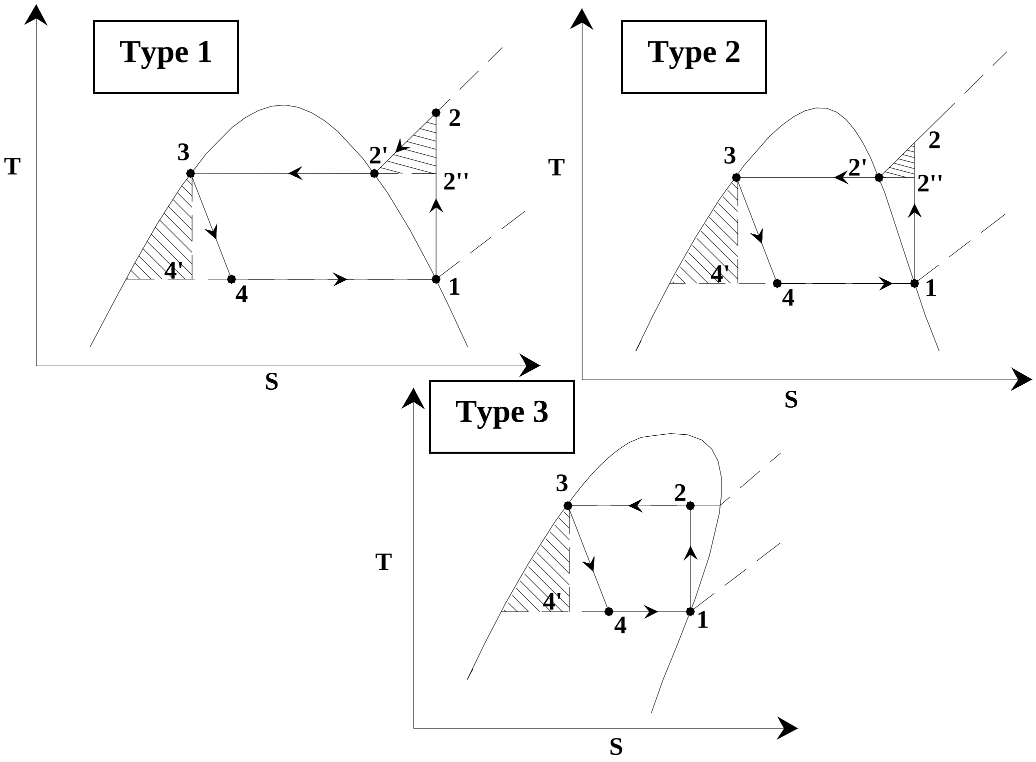


Fig.10.8. *T-s diagrams for three different types of refrigerants*

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 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:

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

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.,

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

<u>Expansion device:</u> 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:

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

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:

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

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:

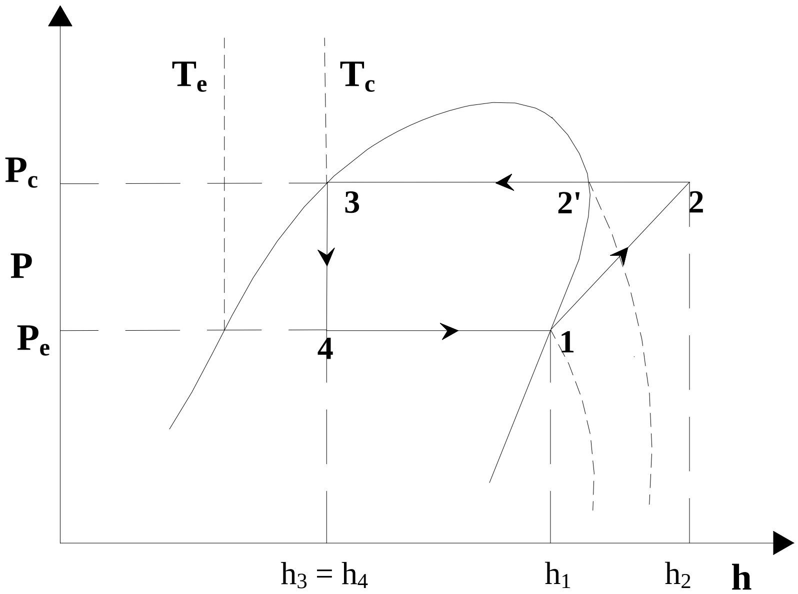


Fig.10.9. *Standard vapour compression refrigeration cycle on a P-h chart*

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.

Questions:

1. A Carnot refrigerator using R12 as working fluid operates between 40ºC and -30ºC. Determine the work of compression and cooling effect produced by the cycle. (Solution)

2. An ideal refrigeration cycle operates with R134a as the working fluid. The temperature of refrigerant in the condenser and evaporator are 40ºC and -20ºC respectively. The mass flow rate of refrigerant is 0.1 kg/s. Determine the cooling capacity and COP of the plant. (Solution)

3. A R-12 plant has to produce 10 tons of refrigeration. The condenser and evaporator temperatures are 40ºC and -10ºC respectively. Determine

a) Refrigerant flow rate
b) Volume flow rate of the compressor
c) Operating pressure ratio
d) Power required to drive the compressor
e) Flash gas percentage after throtting
f) COP (Solution)

4. A NH_3 refrigerator produces 100 tons of ice from water at 0ºC in a day. The cycle operates between 25ºC and -15ºC . The vapor is dry saturated at the end of compression. If the COP is 50% of theoretical COP, calculate the power required to drive the compressor. (Solution)

5. In a refrigerator the power rating impressed on the compressor is 1.2 kW. The circulating wire in evaporator is 5 kW and the cooling water took away 10 kW from condenser coil. The operating temperatures range is 18ºC and 0ºC and their corresponding latent heats are 170 kJ/kg and 230 kJ/kg and the difference between the

liquid energy is 35 kJ/kg. Find the actual COP of the system (2) relative COP, assuming the vapour is just dry and saturated at the end of the compression. (Solution)

6. A water cooler using R12 refrigerant works between 30ºC to 9ºC. Assuming the volumetric and mechanical efficiency of the compressor to be 80 and 90% respectively, and the mechanical efficiency of motor to be 90% , and 20% of useful cooling is lost into water cooler, find:

1) The power requirement of the motor
2) Volumetric displacement of the compressor

Given C_p (saturated vapour at 30ºC) = 0.7 kJ/kg K (Solution)

The properties of F12 at 30ºC and 2ºC are:

Temp ºC	Pressure (Bar)	Liquid		Vapour		
		h_f (kJ/kg)	S_f (kJ/kg K)	h_g (kJ/kg)	S_g (kJ/kg K)	v_s m^3/kg
30	7.45	64.6	0.2399	199.6	0.6854	0.0235
5	3.626	40.7	0.1587	189.7	0.6942	0.0475

UNIT – I

REFRIGERATION

INTRODUCTION

Refrigeration is the Process of Producing and maintaining a body temperature lower than that of the surroundings. i.e atmosphere. According to ASREA, it is defined as the science of providing and maintaining temperatures below that of surroundings. It can also be defined as the process of removing heat from a low temperature level and rejecting it at a relatively higher temperature level. It is nothing but removal of heat from the body. In order to maintain the body temperature lower than the surrounding, heat must be transferred from cold body to the surroundings at high temperature. Since heat cannot naturally flow from cold body to hot body, external energy will have to be supplied to perform this operation. The device used for this purpose is called refrigerating system or mechanical refrigerator of simply refrigerator. The system, which is kept at lower temperature, is termed refrigerated system. Nowadays many human workplaces and factories are air conditioned by using refrigeration system. Not only work places but also perishable goods are maintained at their required temperature to store it for a long time by refrigeration system. This device operates on a reversed power cycle. i.e. heat engine. The medium used to carry the heat from a low temperature level to high temperature level is called a “Refrigerant’.

PRINCIPLE OF REFRIGERATION

Refrigeration is defined as the production of temperature lower than those of the surrounding and maintains the lower temperature within the boundary of a given space. The effect has been accomplished by non cyclic processes such as the melting of ice or sublimation of solid carbon dioxide. However, refrigeration effect is usually produced by transferring heat from a low temperature source to a high temperature source by spending mechanical work. To produce this effect requires certain machinery; hence, the method is called mechanical refrigeration. The working media of such machines are called refrigerants.

METHODS OF REFRIGERATION

The refrigeration effect may be produced by bringing the substance to be cooled in direct or indirect contact with cooling medium such as ice. The common methods of refrigeration are as follows.

1. Art of Ice making by Nocturnal Cooling

The ice refrigeration systems consist of an insulated cabinet equipped with a tray or tank at the top, for holding blocks of ice pieces. Shelves for food are located below the ice compartment. Cold air flows downward from ice compartment and cools the food on the shelves below. Air returns from the bottom of the cabinet up, the sides and back of the cabinet which is warmer, flows over the ice, and again flows down over the shelves to be cooled.

The art of making ice by nocturnal cooling was perfected in India. In this method ice was made by keeping a thin layer of water in a shallow earthen tray, and then exposing the tray to the night sky. Compacted hay of about 0.3 m thickness was used as insulation. The water looses heat by radiation to the stratosphere, which is at around -55˚ C and by early morning hours the water in the trays freezes to ice. This method of ice production was very popular in India.

Direct method and indirect method

a. ***Direct Method*:** In the Direct method, the ice is kept in the same compartment with the articles to be cooled . Ice is placed at elevated position and cools the surrounding air. Cold air due to high density descends below displacing warmer air towards the ice. In the storage space cold air is warmed by absorption of heat from the products to be cooled. And rises through the storage space due to light density. It returns to the ice space. A circulation of air is maintained so long as air cooled on ice space and the cycle is repeated. Direct method of ice refrigeration is shown in fig. This method is used in hotels for cooling the drinks.

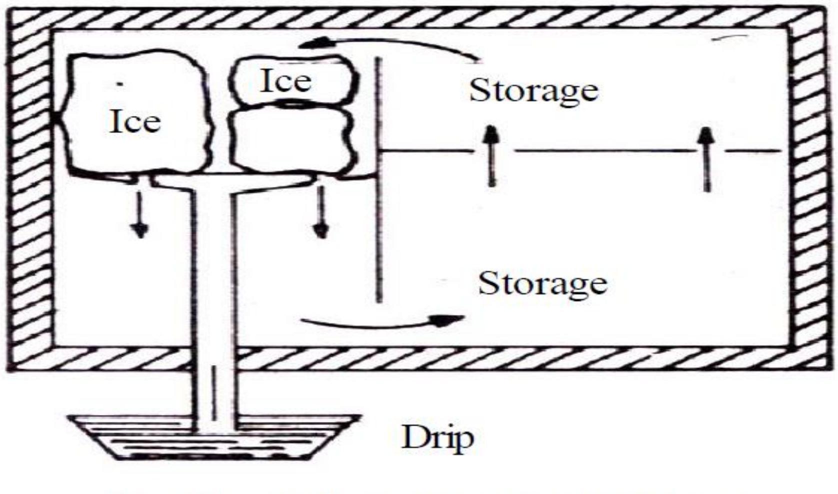


Figure 1 Ice Refrigeration (Direct Method)

b. Indirect Method

In the indirect method, the ice is isolated from the storage space and cooling of the articles is carried by circulation of brine. The ice is supplied to compartment D where it surrounds the vertical tubes containing brine.

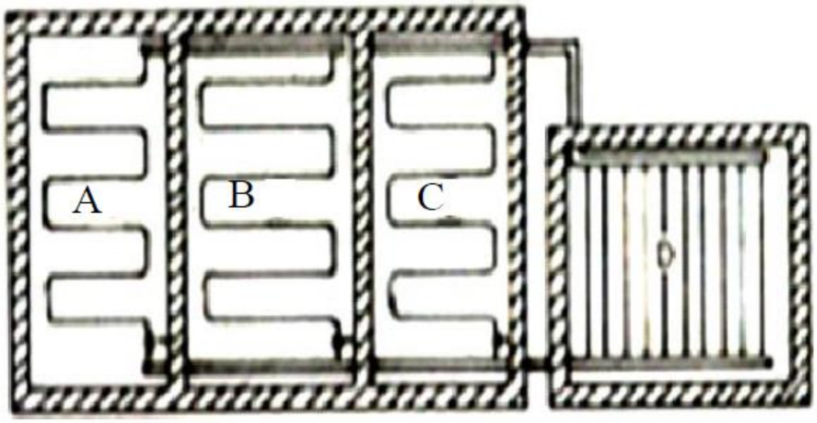


Figure 2 Ice Refrigeration (In Direct Method)

Cold brine is circulated through the storage space A,B and C to cool the articles, and finally the brine returns to ice compartment D. This cycle is repeated. This system is more suitable for large plants and no danger of food infection by impure ice.

3. Dry Ice Refrigeration

Dry ice is the solidified form of CO2. It evaporates directly from solid to vapour without liquid phase. This phenomenon is called sublimation. In this, the dry ice in the form of flakes or slabs Is placed on the cartons containing food stuffs. When dry ice sublimates, it will absorb heat from food stuff in the cartons and thus keeps them in a frozen condition. This is used for preservation of frozen foods and ice creams in storage and transportation. It has twice the heat absorbing capacity of ice refrigeration. However its cost is high.

4. Evaporative Cooling

As the name indicates, evaporative cooling is the process of reducing the temperature of a system by evaporation of water. Human beings perspire and dissipate their metabolic heat by evaporative cooling if the ambient temperature is more than skin temperature. Animals such as the hippopotamus and buffalo coat themselves with mud for evaporative cooling. Evaporative cooling has been used in India for centuries to obtain cold water in summer by storing the water in earthen pots. The water permeates through the pores of earthen vessel to its outer surface where it evaporates to the surrounding, absorbing its latent heat in part from the vessel, which cools the water. It is said that Patliputra University situated on the bank of river Ganges used to induce the evaporative-cooled air from the river. Suitably located chimneys in the rooms augmented the upward flow of warm air, which was replaced by cool air. Evaporative cooling by placing wet straw mats on the windows is also very common in India. The straw mat made from “khus” adds its inherent perfume also to the air.

5. Gas Throttling Refrigeration

In the gas throttling refrigeration process, there is no change in enthalpy and also for a perfect gas, there is no change in temperature. However, for actual gases there is a substantial change, usually a decrease in temperature. This temperature drop depends upon the Joule- Thomas coefficient, the pressure drop, and the initial state of the gas. High pressure gas is throttled through porous plug into the space to be cooled and escapes outside after absorbing heat from the space. This produces the refrigeration effect in the space or the product kept in the space.

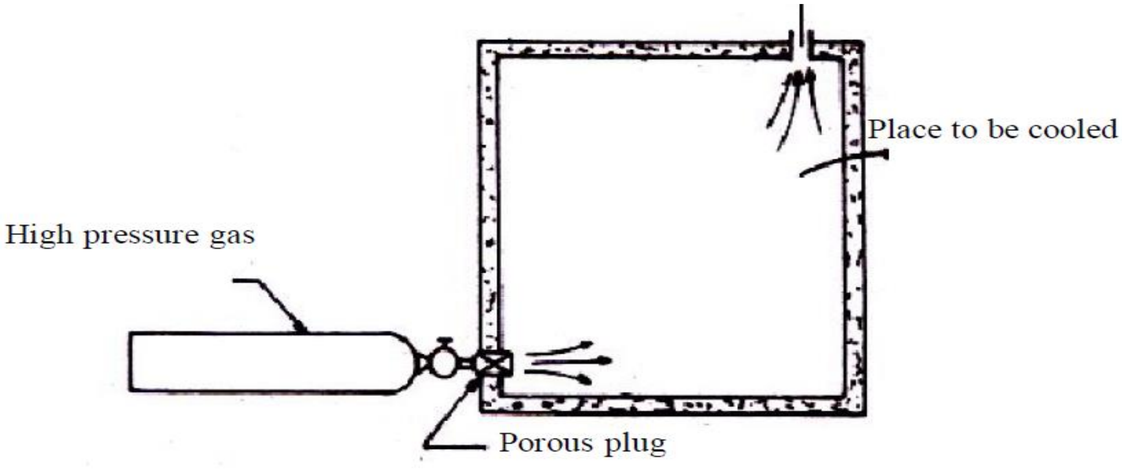


Figure 3 Gas Throttling Refrigeration

UNITS OF REFRIGERATION

Quantity of heat removed from the body per unit time is very important in the design of refrigerating system. The amount of heat removed is expressed in ton of refrigeration (TR). This is also called capacity of refrigerating system. It is defined as the quantity of heat removed in of heat is to be removed from one ton of water at 0˚C to freeze it to ice at 0˚C in one day, i.e. 24 hours.

COEFFICIENT OF PERFORMANCE

Performance of a heat engine is measured by a term known as efficiency. Similarly, the performance of a refrigeration system is measured by a term known as coefficient of performance (COP). It is defined as the ratio of desired effect (quantity of heat removed from a cold body) to the net work done on the system.

A part of heat supplied to the engine (Q_s) is converted into useful work (W_E) and the remaining part of the heat is rejected (Q_R) to the surrounding (sink). According to the 1st law of thermodynamics,

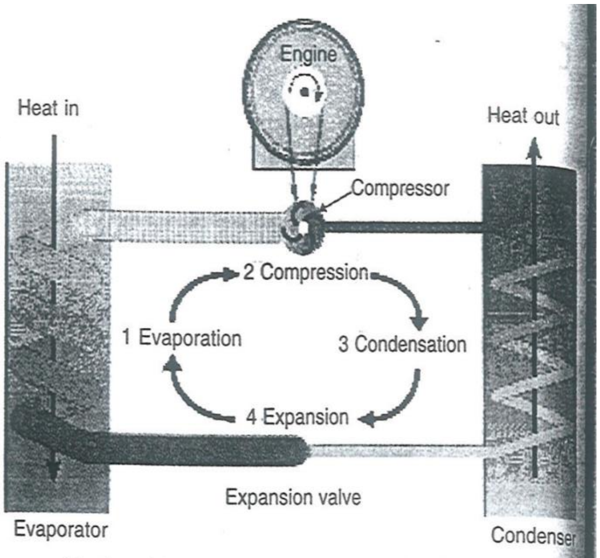
$$W_E = Q_S - Q_R$$

Efficiency of heat engine, $\eta_E = \frac{NETworkdone}{Heat\,\mathrm{sup}\,plied} = \frac{W_E}{Q_S} = \frac{Q_S - Q_R}{Q_S} = 1 - \frac{Q_R}{Q_S}$

SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEMS

A vapors compression refrigeration system is an improved type of air refrigeration system in which a suitable working substance, termed as refrigerant, is used. It condenses and evaporates at temperatures and pressures close to the atmospheric conditions. The refrigerants, usually, used for this purpose are ammonia (NH), carbon dioxide (CO_2) and sulphur dioxide (SO_2). The refrigerant used, does not leave the system, but is circulated

- Since low pressure vapor refrigerant from the evaporator is changed into high pressure vapor refrigerant in the compressor, therefore it is named as vapor compression refrigeration system throughout the system alternately condensing and evaporating, the refrigerant absorbs its latent heat from the brine* (salt water) which is used for circulating it around the cold chamber. While condensing, it gives out its latent heat to the circulating water of the cooler. The vapors compression refrigeration system is therefore a latent heat from the brine and delivers it to the cooler.



Engine-driven vapour compression heat pump.

The vapor compression refrigeration system is now-a-days used for all purpose refrigeration. It is generally used for all industrial purposes from a small domestic refrigerator to a big air conditioning plant.

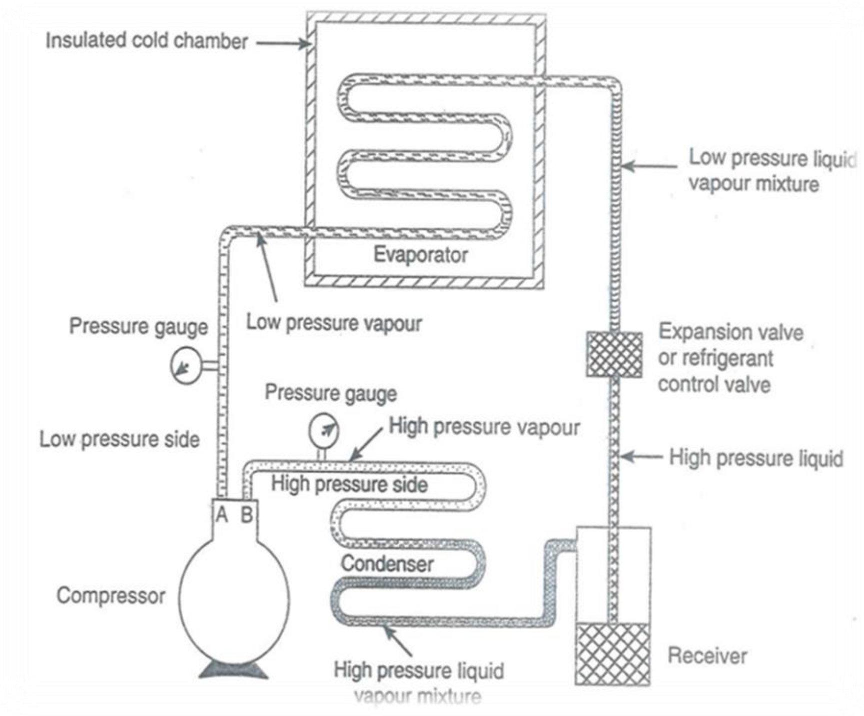


Figure 4 simple Vapour compression refrigeration systems

1. **Compressor**. The low pressure and temperature vapor refrigerant from evaporation is drowning into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. The high pressure and temperature vapor refrigerant is discharged into the condenser through the delivery or discharge valve B.
2. **Condenser.** The condenser or cooler consist of coils of pipe in which the high pressure and temperature vapor refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.
3. **Receiver.** The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver form where it is supplied to the evaporator through the expansion valve or refrigerant control valve.
4. **Expansion valve.** It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the grater potion is vaporized in the evaporator at the low pressure and temperature.

5. **Evaporator.** An evaporator consists of coils of pipe in which the liquid-vapor refrigerant at low pressure and temperature is evaporated and changed into vapor refrigerant at low pressure and temperature. In evaporating, the liquid vapor refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which is to be cooled.

PRESSURE-ENTHALPY (P-H) CHART

The most convenient chart for studding the behavior of a refrigerant is the p-h char, in which the vertical ordinates represent pressure and horizontal ordinates represent enthalpy (i.e. total heat). A typical chart is shown in Fig. 5, in which a few important lines of the complete chart are drowned. The saturated liquids line and situated vapor line merge into one another at the critical point. A saturated liquid is one which has a temperature equal to the saturation temperature corresponding to its pressure. The space to the left of the saturated liquid line will; therefore, be sub-cooled liquids region. The space between the liquids and the vapor lines is called wet vapor region and to the right of the saturated vapor line is a superheated vapor region

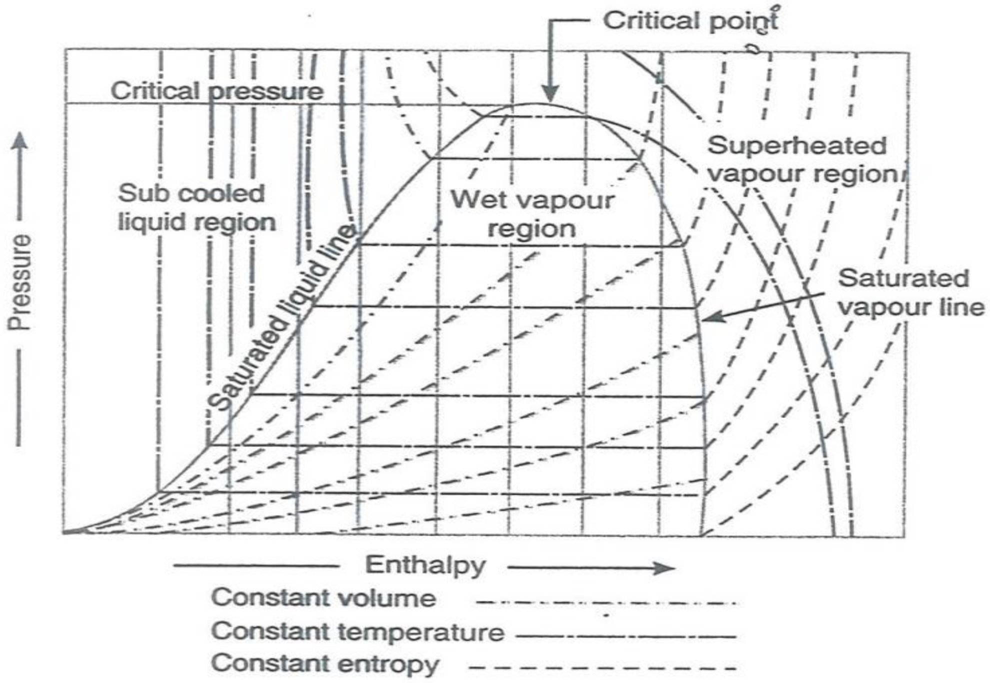


Figure 5 Pressure –enthalpy (p-h) Chart

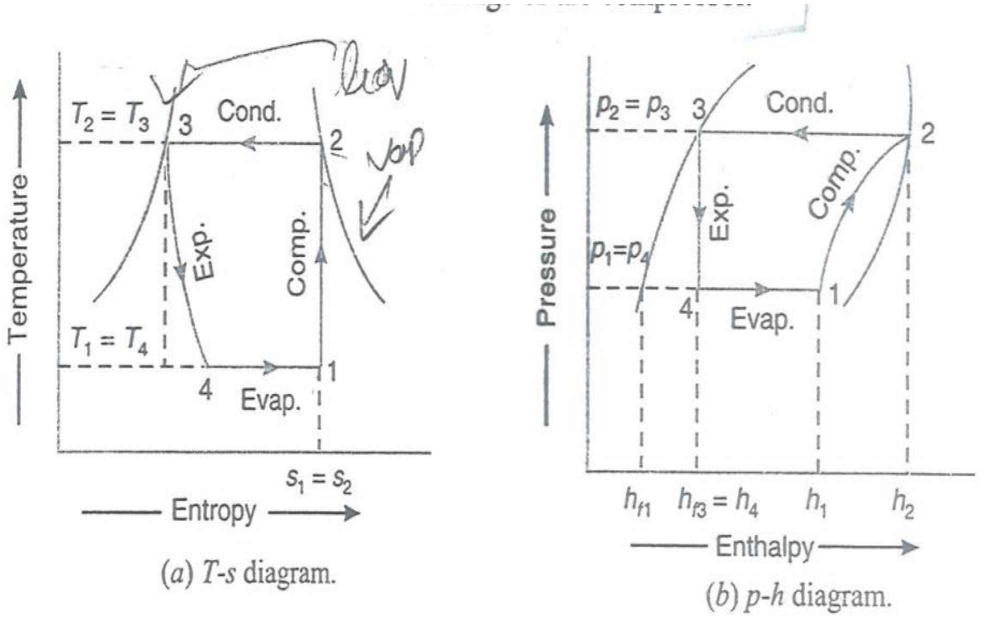
TYPES OF VAPOR COMPRESSION CYCLES

We have already disused that vapor compression cycle essentially consists of compression, condensation, throttling and evaporation. Many scientists have focused their attention to increase the coefficient of performance of the cycle. Through there are many cycles, yet the following are important from the subject point of view:

1. Cycle with dry saturated vapor after compression,
2. Cycle with wet vapor after compression,
3. Cycle with superheated vapor after compression,
4. Cycle with superheated vapor before compression, and
5. Cycle with under cooling or sub cooling of refrigerant,

1. Theoretical Vapor Compression Cycle with Dry Saturated vapor after compression

A vapour compressing cycle with dry started vapour after compressions is shown on T-s and p-h diagrams in Fig 6. (a) and (b) respectively. At point 1,let T_1 p_1 and s_1 be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycles are as follows:



(a) T-s diagram.

(b) p-h diagram.

Figure 6 Theoretical vapour compression cycle with dry saturated vapour after compression

1. Compassion process. The vapour reliant at low pressure p_1 and temperature T_1 is compressed is entropic ally to dry saturated vapour as shown by the vertical line 1-2 onT_1 is and by the cove 1-12 on p-h diagram. The pressure and temperature rise from p_1 to p_1 and T_1 to T_2 respectively.

   The work done during isentropic compression per kg of refrigerants is given by

   $$W = h_2 - h_1$$

   $h_1 =$ Enthalpy of vapour refrigerant at temperature T_1

   h_2=Enthalpy of the vapor refrigerant at temperature T_2

2. **Condensing process.** The high pressure and temperature vapor refrigerant from the compressor is passed through the condensed T_2, as shown by the horizontal line 2-3 on T-s and p-h diagrams. The vapor refrigerant is changed into liquied refrigerant. The refrigerant, where passing through the condenser, given its latent heat to the surrounding condensing medium.

3. **Expansion process.** The liquid refrigerant at pressure p_3=p_2 and temperature T_3=T_2 is expanded by * throttling process through the expansion valve to a low pressure p_4=p_1 and temperature T_4=T_1, as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on through the expansion value, but the greater portion is vapored in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

4. Vaporizing process. The liquid-vapour mixture of the refrigerant at pressure p_4=p_1 and temperature T_4=T1is evaporated and changed into vapor refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporates continues up to point 1 which is the starting point and thus the cycle is completed.

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapor refrigerant during evaporation per kg of refrigeration is given by

$$R_E = h_1 - h_4 = h_1 - h_{f3} \,.....(\because h_{f3} = h_4)$$

Where h_{f3} = Sensible heat at temperature T_3, i.e., enthalpy of liquid

Refrigerant leaving the condenser.

It may be noticed from the cycle that the liquid-vapor refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapor refrigerant.

$\therefore$ Coefficient of performance,

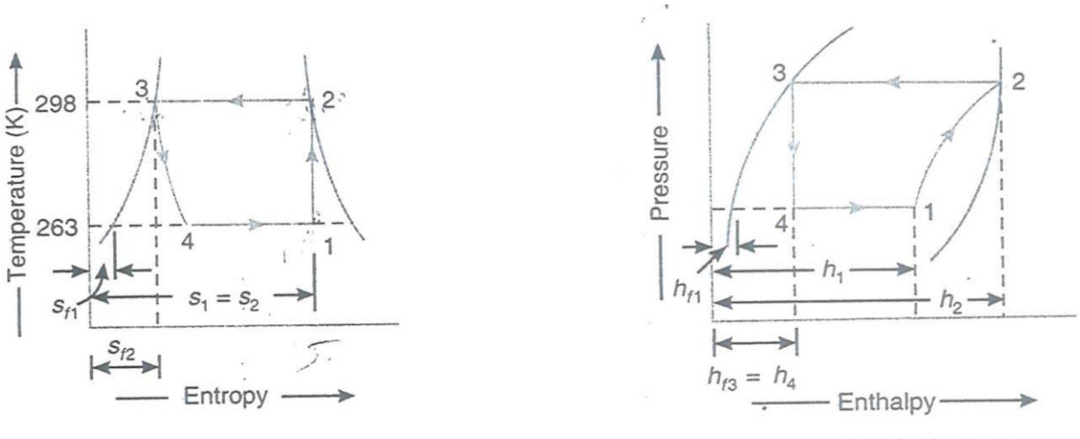
$$c.o.p = \frac{refigeratingeffect}{Workdone} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

PROBLEMS

1. The temperature limits of an ammonia refrigerating system are 20ºC and -10ºC. If the gas is dry at the end of compression, calculate the coefficient of performance of the cycle assuming no under cooling of the liquid ammonia. Use the following table for properties of ammonia:

Temperature (ºC)	**Liquid heat (kJ/kg)**	**Latent heat (kJ/kg)**	**Liquid entropy (KJ/Kg K)**
25	**298.9**	**1166.94**	**1.1242**
-10	**135.37**	**1297.68**	**0.5443**

Solution. Given: T_2=T_3=25ºC=25+273=298 K; T_1=T_4= -10ºC =-10+273=263 K; h_3=h_4=298.9 KJ/Kg; h_{fg2}=1166.94 KJ/Kg; S_{f2}=1.1242 Kj/Kg K; h_{f1} =135.37KJ/Kg ;h_{fg1}=1297.68 KJ/Kg;S_{f1}=0.5443 KJ/KgK



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

The T-Sand p-h diagram is shown in above figure respectively.

Let x_1=Dryness fraction at point 1.

We know that entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$$= 0.5443 + 4.934 x_1 \qquad \text{..... (i)}$$

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{h_{fg2}}{T_2} = 1.1242 + \frac{1166.94}{298} = 5.04 \qquad \text{.....(ii)}$$

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$$0.5443 + 4.934 x_1 = 5.04 or x_1 = 0.91$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x1 h_{fg1} = 135.37 + 0.91 \times 1297.68 = 1316.26 KJ / Kg$$

And enthalpy at point 2, $h_2 = h_{f2} + h_{fg2} = 298.9 + 1166.94 = 1465.84 KJ / Kg$

$\therefore$ Coefficient of performance of the cycle

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1316.26 - 298.9}{1465.84 - 1316.26} = 6.8 Ans.$$

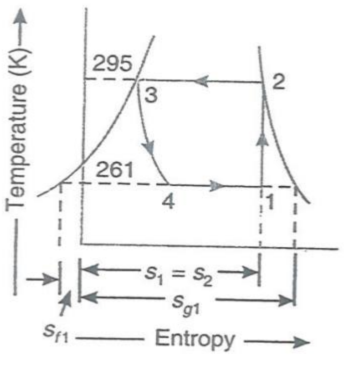
2. A vapor compression refrigerant work between the pressure limits of 60 bars and 25 bars. The working fluid is just dry at the end of compression and there is no under cooling of the liquid before the expansion valve. Determine: 1. C.O.P of the cycle; and 2. Capacity of the refrigerant if the fluid flow is at the rate of 5Kg/min.

Pressure (bar)	**Saturation temperature(K)**	**Enthalpy (KJ/Kg)**		**Entropy (KJ/KgK)**	
		Liquid	**Vapor**	**Liquid**	**Vapor**
60	**295**	**151.96**	**293.29**	**0.554**	**1.0332**
25	**261**	**56.32**	**322.58**	**0.226**	**1.2464**

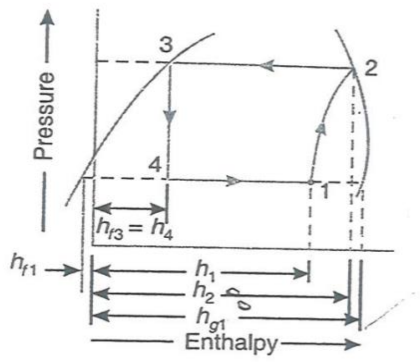
Solution. Given: $P_2=p_3=60$ bar; $p_1=p_4=25$ bar; $T_2=T_3=295$ K$T_1=T_4$261K;$h_{f3}=h_4=151.96$ KJ/Kg K;$S_{f1}=0.226$ KJ/KgK; $S_{g2}=S_2=1.0332$ KJ/Kg K; $S_{g1}=1.2464$ KJ/KgK

1. **C.O.P. of the cycle**

The T-s and p-h diagrams are shown in Fig 4.5 (a) and (b respectively.



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

Let x_1=Dryness fraction of the vapor refrigerant entering the compressor at point 1.

We know that entropy at point 1.

$$s_1 = s_{f1} + x_1 s_{fg1} = s_{f1} + x_1(s_{g1} - s_{f1}).....(\because s_{g1} = s_{f1} + s_{fg1})$$

$$= 0.226 + x_1(1.2464 - 0.226) = 0.226 + 1.0204x_1....(i)$$

And entropy at point 2, $s_2 = s_{g2} = 1.0332 KJ / KgK......(Given)...(ii)$

Since the entropy at point 1 is equal to entropy at point 2, therefore equaling equations (i) and (ii)

$$0.226+1.0204X_1=1.0332 \qquad \text{or} \quad x_1=0.791$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = h_{f1} + x_1(h_{g1} - h_{f1}) .... (\because h_{g1} = g_{f1} + h_{fg1})$$

$$=56.32+0.791(322.58-56.32)=266.93KJ/Kg$$

$\therefore$ C.O.P of the cycle

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{266.93 - 151.96}{293.29 - 266.93} = 4.36 Ans.$$

2. **Capacity of the refrigerator**

   We know that the heat extracted or refrigerating effect produced per kg of refrigerant

   $$= h_1 - h_{f3} = 266.93 - 151.96 = 114.97 KJ / Kg$$

   Since the fluid flow is at the rate of 5kg/min, therefore total heat extracted

   $$= 5 \times 114.97 = 574.85 kj / kg$$

   $\therefore$ Capacity of the refrigerator

   $$= \frac{574.85}{210} = 2.74 TRAns. ..... (\therefore \ 1TR=210kj/min)$$

3. **28 tons of ice from and at 0ºCis produced per day in an ammonia refrigerator. The temperature range in the compressor is from 25ºC to -15ºC. The vapor is dry and saturated at the end of compression and an expansion valve is used. There is no liquid sub cooling. Assuming actual C.O.P of 62% of the theoretical. Calculate the power required to drive the compressor. Following properties of ammonia are given:**

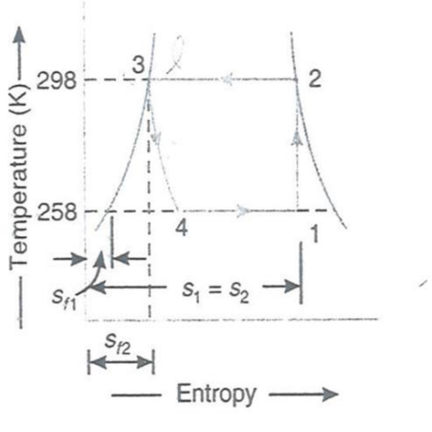
Temperature	**Enthalpy (kJ/Kg)**		**Entropy (kj/kg K)**	
0ºC	**Liquid**	**Vapor**	**Liquid**	**Vapor**
25	**298.9**	**1465.84**	**1.1242**	**5.0391**
-15	**112.34**	**1426.54**	**0.4572**	**5.5490**

Take latent heat of ice =335 kJ/kg.

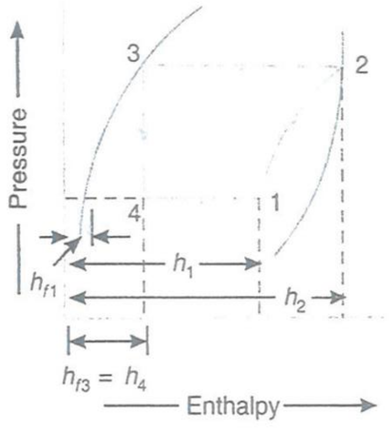
Solution. Given: Ice produced =28t/day; $T_2=T_3=25$ºC =25+273=298K; $T_1=T_4=-15$ºC=-15+273=258K; $h_{f3}=h_4=$298.9 kJ/kg ; h_{f1}=112.34 kJ/kg; $h_{g2}=h_2$=1465.84 kj/kg; h_{g1}=1426.54kj/kg; *s_{f2}=1.1242kj/kg K; S_{f1}=0.4572 kj/kg K; $S_{g2}=s_2$=5.0391 kJ/kg; s_{g1}=5.5490 kJ/kg K.

The T-s and p-h diagrams are shown in below figure respectively.

First of all let us find the dryness fraction (x_1) of the vapor refrigerant entering the compressor at point 1.



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

We know that entropy at point 1,

$$s_1 = s_{f1} + s_{fg1} = s_{fg1} = x_1(s_{g1} - s_{f1}) \qquad \ldots(\therefore s_{g1} = s_{f1} + s_{fg1})$$

$$=0.4572+ x_1 (5.5490-0.4572)$$

$$= 0.4572+5.0918\, x_1 \qquad (kj/kgK.....(Given)....(ii)$$

And entropy at point 2,

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$$0.4572+5.0918\,x_1=5.0391 \quad \text{or} \quad x_1=0.9$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = h_{f1} + x_1(h_{g1} - h_{f1}) \;\ldots.(\because h_{g1} = h_{f1} + h_{fg1})$$

$$=112.34+0.9(1426.54-112.34)=1295.12 \text{ kJ/kg}$$

$\therefore$ Theoretical C.O.P = $\dfrac{h_1 - h_{f3}}{h_2 - h_1} = \dfrac{1295.12 - 298.9}{1465.84 - 1295.12} = \dfrac{996.22}{170.72} == 5.835$

Since actual C.O.P is 62%of theoretical C.O.P therefore

$$\text{Actual C.O.P} == 0.62 \times 5.835 = 3.618$$

We know that ice produced from and at 0ºC

$$= 28t / day = \frac{28 \times 1000}{24 \times 3600} = 0.324 kg / s$$

Latent heat of ice = 335KJ/kg

$\therefore$ Refrigeration effect produced

$$= 0.324 \times 335 = 108.54 kj / s$$

We know that actual C.O.P

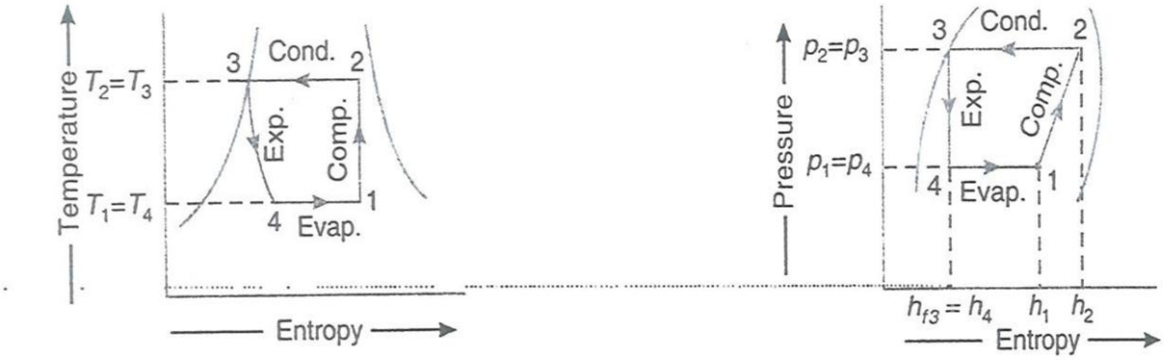
$$3.618 = \frac{refrigeration effect}{Workdone} = \frac{108.54}{Workdone}$$

$\therefore$ Work done or power required driving the compressor

$$= \frac{108.54}{3.618} = 30 \text{ kJ/s} \quad \text{or KW ans.}$$

2. Theoretical Vapor Compression cycle with wet Vapor after compression

A vapor compression cycle with wet vapor after compression is shown on T-s and p-h diagrams I Fig 7 (a) and (b) respectively. In this cycle, the enthalpy at point 2 is found out with the help of dryness fraction at this point. The dryness fraction at points 1 and 2 may be obtained by equating entropies at point 1 and 2.



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

Figure 7 Theoretical vapour compression cycle with wet vapour after compression

Now the coefficient of performance may be found out as usual from the relation,

$$C.O.P = \frac{refrigeratingeffect}{workdine} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

4. Find the theoretical C.O.P for a CO_2machine working between the temperature range of 25ºC and -5ºC. The dryness fraction of CO_2 gas during the suction stroke is 0.6. Following properties of CO_2 are given:

Temperature ºC	**Liquid**		**Vapor**		**Latent heat KJ/Kg**
	Enthalpy KJ/Kg K	**Entropy KJ/Kg K**	**Enthalpy KJ/Kg**	**Entropy KJ/Kg K**	
25	164.77	0.5978	282.23	0.9918	117.46
-5	72.57	0.2862	321.33	1.2146	248.76

Solution. Given T_2=T_3=25ºC = 25+273=298 K; T_1=T_4= -5ºC =-5+273=268 K; x_1 =0.6;h_{f3} h_{f2}=164.77 kJ/kg; h_1=h_{f4} =72.57 kJ/kg ; S_{f2}=0.5978 kJ/kg K;s_{f1}=0.2862 kJ/kg K; $h_{2'}$=282.23 kj/kg; $h_{1'}$=321.33 kj/kg ; *$s_{2'}$=0.9918 kj/kg K; *$s_{1'}$ =1.2146 kj/kg h_{fg2}=117.46 kj/kg ;h_{fg1}=248.76 kj/kg

The T-s and p-h diagram are shown in Figure.

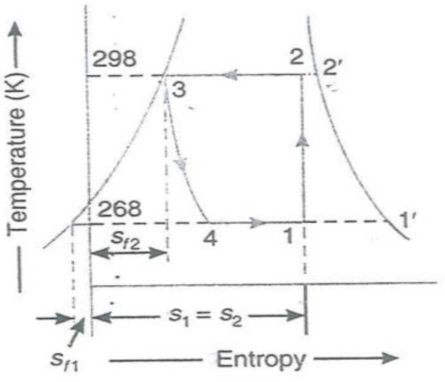
First of all, let us find the dryness fraction at point 2,i.e x_2,. We know that th entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.2862 + \frac{0.6 \times 248.76}{268} = 0.8431....(i)$$

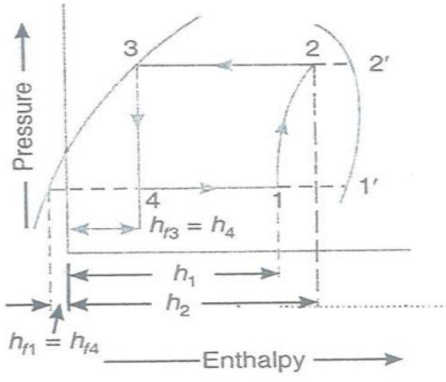
Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{x_2 h_{fg2}}{T_2} = 0.5978 + \frac{x_2 \times 117.46}{268}$$

$$=0.5978+0.3941\, x_2 \qquad ….(ii)$$



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

Since the entropy at point 1(s_1) is equal to trophy at point 2(s_2), therefore equaling equations (i) and (ii),

$$0.8431=0.5978+0.3941\, x_2 \quad \text{or} \quad x_2 =0.622$$

We knows that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = 72.57 + 0.6 \times 248.76 = 221.83 kj/kg$$

And enthalpy at point 2, $h_2 = h_{f2} + x_2 h_{fg2} = 164.77 + 0.622 \times 117.76 = 237.83 kj/kg$

$\therefore$ Theretical C.O.P.$= = \dfrac{h_1 - h_{f3}}{h_2 - h_1} = \dfrac{221.83 - 164.77}{237.83 - 221.83} = \dfrac{57.06}{16} = 357 \, Ans.$

5. An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of -10°C and 30°C. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the fluid leaving the condenser is at 30°C. Assuming actual C.O.P. as 60% of the theoretical, calculate the kilograms of ice produced per kW hour at 0°C from water at 10°C. Latent heat of ice is 335 kj/kg. Ammonia has the following properties:

Temperature °C	**Liquid heat (h_1) kj/kg °C**	**Latent heat (h_{f8}) kj/kg**	**Liquid entropy (S_f)**	**Total entropy of dry saturated vapour**
30	323.08	1145.80	1.2037	4.9842
-10	135.37	1297.68	0.5443	5.4770

Solution: Given: $T_1 = T_4$ = -10°C = -10 + 273 = 263 K: $T_2 = T_3$=30°C+273=303K ; x_2 = 0395 ; $h_{f3} = h_{f2}$ = 323.08 kj/kg ; $h_{f1} = h_{f4}$ = 135.37 kl/kg ; h_{f82} = 1145.8 kj/kg ; h_{f81} = 1297.68 kj/kg, s_{f2} = 1.2037 ; s_{f1} = 0.5443 ‘ *$s_{2'}$ = 4.9842 , *$s_{1'}$ = 5.4770

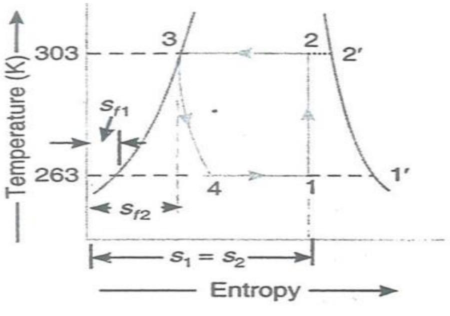
The T-s and p-h diagrams are shown in Figure respectively.

Let x_1 = Dryness fraction at point 1.

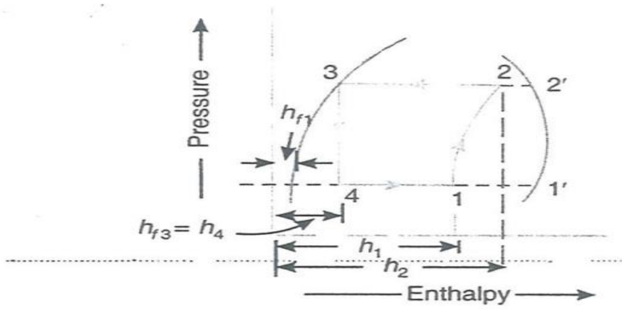
We know that entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{f81}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$$= 0.5443 + 4.934 x_1 \qquad \text{..... (i)}$$



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

Similarly, entropy at point 2,

$$s_1 = s_{f2} + \frac{x_2 h_{f8^2}}{T_2} = 1.2037 + \frac{0.95 \times 1145.8}{303} = 4.796 \qquad \text{...(ii)}$$

Since the entropy at point 1 (s_1) is equal to entropy at point 2 (s_2), therefore equating equations (i) and (ii)

$0.5443 + 4.934\, x_1 = 4.796$ or x1 = 0.86

∴ Enthalpy at point 1, $h_1 = h_{f1}$ + x1 h_{fg1} = 135.37 + 0.86 x 1297.68 = 1251.4 kj/kg

and enthalpy at point2, h2 = h_{f2} + x2 h_{f2} = 323.08 + 0.95 x 1145.8 = 1411.6 kj/kg

We know that theoretical C.O.P.

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1251.4 - 323.08}{1411.6 - 1251.4} = 5.8$$

∴ Actual C.O.P. = 0.6x5.8=3.48

Work to be spent corresponding to 1kW hour,

W = 3600 kJ

∴ Actual heat extracted or refrigeration effect produced per kW hour

=W x Actual C.O.P. = 3600 x 3.48 = 12 528 kJ

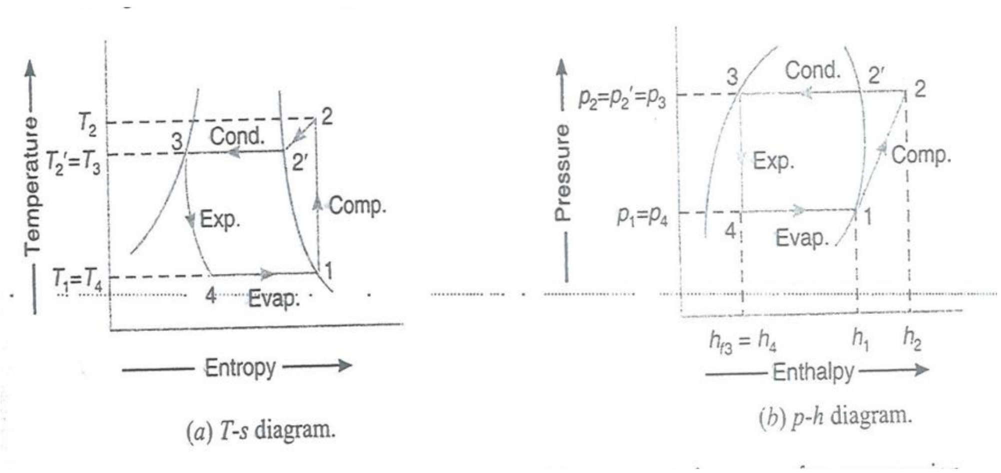
We know that heat extracted from 1kg or water at 10ºC for the formation of 1kg of ice at 0ºC

= 1x4 4.187 x 10 + 335 = 376.87 kJ

∴ Amount of ice produced

$$= \frac{12528}{376.87} = 33.2 kg / kWhourAns.$$

3. Theoretical Vapour Compression Cycle with Superheated Vapour after Compression



(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

Figure 8 Theoretical vapour compression cycle with Superheated after compression

A vapour compression cycle with superheated vapour after compression is shown on T-s and p-h diagrams in Figure. In this cycle, the enthalpy at point 2 is found out with the help of degree of superheat. The degree of superheat may be found out by equating the entropies at points 1 and 2.

Now the coefficient of performance may be found out as usual from the relation.

$$C.O.P.=\frac{\text{Re}\,frigeratingeffect}{workdone}=\frac{h_1-h_{f3}}{h_2-h_1}$$

A little consideration will show that the superheating increases the refrigerating effect and the amount of work done in the compressor. Since the increase in refrigerating effect is less as compared to the increase in work done, therefore, the net effect of superheating is to have low coefficient of performance.

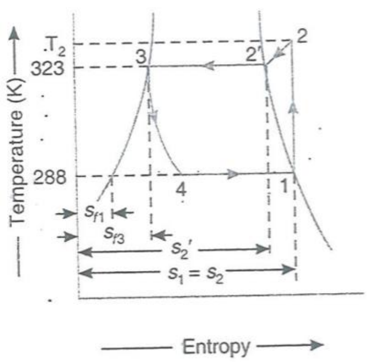
6. A simple refrigerant 134a (tetraflurocthane) heat pump for space hearing, operates between temperature limits of 15°C and 50°C. The heat required to be pumped is 100MJ/h. Determine: 1. The dryness fraction of refrigerant entering the evaporator; 2.The discharge temperature assuming the specific heat of vapour as 0.996 kJ/kg K; 3. The theoretical piston displacement of the compressor; 4. The theoretical power of the compressor; and 5. The C.O.P.

The specific volume of refrigerant 134a saturated vapour at 15°C is 0.04185 m3/kg. The other relevant properties of R-134a are given below:

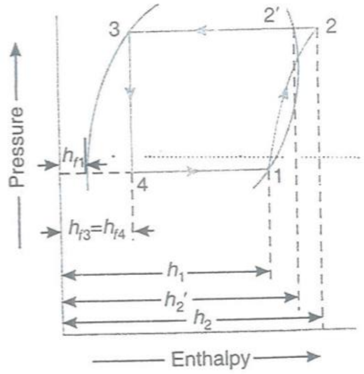
Saturation temperature (°C)	Pressure (bar)	Specific enthalpy (kJ/kg)		Specific entopy (kJ/kg K)	
		Liquid	Vapour	Liquid	Vapour
15	4.887	220.26	413.6	1.0729	1.7439
50	13.18	271.97	430.4	1.2410	1.7310

Solution: Given: T1 = T4 = 15^oC = *15 + 273 = 288 K ; T2 = T3 = 50^oC = 50 + 273 = 323 K ; Q = 100 MJ/h = 100* X *10^{3}kJ/h; c_p=0.996 kJ/kg K ; v_1 = 0.04185 m^3/kg; h_{f1} = 220.26 kJ/kg ; h_{f3} = h_4 = 271.97 kJ/kg ; h_1 = 413.6 kJ/kg ; h_2 = 430.4 kJ/kg ; s_{f1} = 1.0729 kJ/kg K ; s_1 = s_2 = 1.7439 kJ/kg K ; s_{f3} = 1.2410 kJ/kg K ; s_2 = 1.7312 kJ/kg K*

The *T-s* and *p-h* diagrams are shown in Figure



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

1. Dryness fraction of refrigerant entering the evaporator

We know that dryness fraction of refrigerant entering the evaporator *i.e.* at point 4,

$$x_4 = \frac{h_4 - h_{f1}}{h_1 - h_{f1}} = \frac{271.97 - 220.26}{413.6 - 220.26} = \frac{51.71}{193.34} = 0.2675 Ans.$$

2. Discharge temperature

Let T_2= Discharge temperature.

We know that entropy at discharge *i.e.* at point 2,

$$s_2 = s_{2'} + 2.3 c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

$$1.7439 = 1.7312 + 2.3 \times 0.996 \log\left(\frac{T_2}{T_{2'}}\right)$$

$$\log\left(\frac{T2}{T2'}\right)=\frac{1.7436-1.7312}{2.3\times0.996}=0.00554$$

$$\frac{T_2}{T_{2'}}=1.0128$$ ...(Taking antilog of 0.005 54)

$$T2=T2'\times1.0128=323\times1.0128=327.13K=54.13^{\circ}CAns.$$

3. Theoretical piston displacement of the compressor

We know that enthalpy at discharge *i.e.* at point 2,

$$h_2=h_{2'}+c_p(T_2-T_{2'})$$

=430.4 + 0.996 (327.13-323)=434.5 kJ/kg

And mass flow rate of the refrigerant,

$$m_R=\frac{Q}{h_2-h_{f3}}=\frac{100\times10^3}{434.5-271.97}=615.3kg/h=$$ 10.254 kg/min

∵ Theoretical piston displacement of the compressor

$=m_R x v_1=10.254 \text{ x } 0.4185 = 4.59 \text{ m}^3\text{/min}$ Ans.

4. ***Theoretical power of the compressor***

We know that work done by the compressor

$=m_R(h_2-h_1) = 10.254 (434.5 – 413.6) = 214.3$ kJ/min

∵ Power of the compressor = 214.3/60 = 3.57 kJ/s or kW Ans.

5. ***C.O.P.***

We know that $$C.O.P.=\frac{h_1-h_{f3}}{h_2-h_1}=\frac{413.6-271.97}{434.5-413.6}=\frac{141.63}{20.9}6.8Ans.$$

7. A refrigeration machine using R-12 as refrigerant operates between the pressures 2.5 bar and 9 bar. The compression is isentropic and there is no undercooling in the condenser.

The vapour is in dry saturated condition at the beginning of the compression Estimate the theoretical coefficient of performance. If the actual coefficient of performance is 0.65 of theoretical value, calculate the net cooling produced per hour. The refrigerant flow is 5 kg per minute. Properties of refrigerant are

Pressure, bar	Saturation temperature, C	Enthalpy, kJ/kg		Entropy of saturated vapour, kj/kg K
		Liquid	Vapour	
9.0	36	70.55	201.8	0.6836
2.5	-7	29.62	184.5	0.7001

Take Cp for superheated vapour at 9 bar as 0.64 kj/kg K.

Solution.*Given :* $T_{2'} = T_3 = 36C = 36 + 273 = 309$ *K ;* $T_1 = T_4 = -7C = 7 + 273 = 266$ *K l; (C.O.P.)* $_{actual}$ *= 0.65 (C.O.P.)* $_{th}$ *;m = 5kg / min ;* $h_{f3} = h_4 = 70.55$*kJ/kg ; **$h_{f1} = h_{f4} = 293.62$*kJ/kg ;* $h_2 = 201.8$ *kJ/jkg;* $h_1 = 184.5$ *kJ/kg;* $s_2 = 0.6836$ *kJ/kg K;* $s_1 = s_2 = 0.7001$ *kJ/kg K;* $c_p = 0.64$ *kJ/jkg K*

The *T*-s and *p*-*h* diagrams are shown in Figure

(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

Theoretical coefficient of performance

First of all, let us find the temperature at point 2 (T_2)

We know that entropy at point 2,

$$s_2 = s_{2'} + 2.3 c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

$$0.7001 = 0.6836 + 2.3 \times 0.64 \log\left(\frac{T_2}{309}\right)$$

$$\log\left(\frac{T_2}{309}\right) = \frac{0.7001 - 0.6836}{2.3 \times 0.64} = 0.0112$$

$$\left(\frac{T_2}{309}\right) = 1.026 \qquad \text{...(Taking antilog of 0.0112)}$$

∵ $T_2 = 1.026 \times 309 = 317$ K

We know that enthalpy of superheated vapour at point 2,

$$h_2 = h_{2'} + c_p(T_2 - T_{2'})$$

$$=201.8 + 0.64\ (317-309) = 206.92 \text{ kJ/kg.}$$

$\because$ Theoretical coefficient of performance,

$$(C.O.P.)_{th} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1845 - 7055}{206.92 - 184.5} = 5.1 Ans.$$

Net cooling produced per hour

We also know that actual C.O.P. of the machine.

$$(\text{C.O.P})_{\text{actual}} = 0.65 \times (\text{C.O.P.})_{\text{th}} = 0.65 \times 5.1 = 3.315$$

and actual work done, $w_{\text{actual}} = h_2 - h_1 = 206.92 - 184.5 = 22.42$ kJ/kg

We know that net cooling (or refrigerating effect) produced per kg of refrigerant

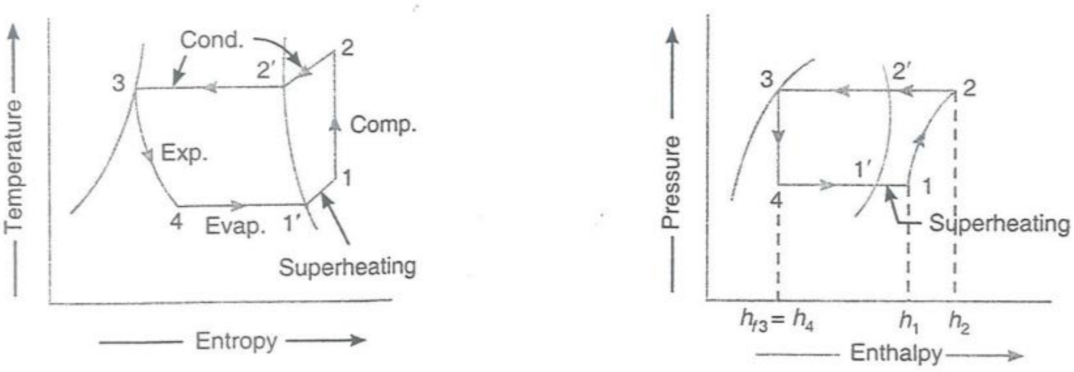
$$= w_{\text{actual}} = \times (\text{C.O.P.})_{\text{actual}} = 22.42 \times 3.315 = 74.3 \text{ kJ/kg}$$

$\because$ Net cooling produced per hour

$$= m \times 74.3 = 5 \times 74.3 = 371.5 \text{ kJ.min}$$

$$= \frac{375.5}{210} = 1.77 TR Ans. \qquad ...(\because 1 \text{ TR} = 210 \text{ kJ/min})$$

4. Theoretical Vapour Compression Cycle with Superheated Vapour before compression



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

Figure 9 Theoretical vapour compression cycle with Superheated before compression

A Vapour compression cycle with superheated vapour before compression is shown on T-s and p-h diagrams in Fig. 9 (a) and (b) respectively. In this cycle, the evaporation starts at point 4 and continues upto point 1' ,when it is dry saturated. The vapour is now superheated before entering the compressor upto the point 1.

The coefficient of performance may be found out as usual from the relation.

$$\text{C.O.P} = \frac{\text{Re}\ frigeratingeffect}{workdone} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

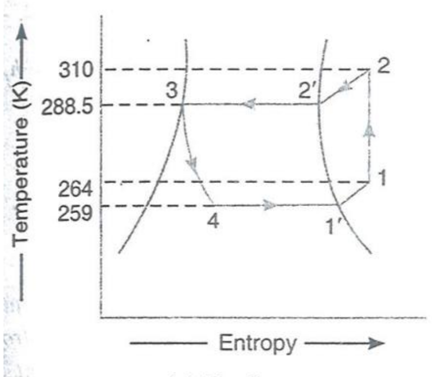
8. A vapour compression refrigeration plant works between pressure limits of 5.3 bar and 2.1 bar. The vapour is superheated at the end of compression. its temperature being performance of the plant. Use the data given below:

Pressure , bar	**Saturation temperature, $^\circ C$**	**Liquid heat, kJ/kg**	**Latent heat, kJ/kg**
5.3	**15.5**	**56.15**	**144.9**
2.1	**-14.0**	**25.12**	**158.7**

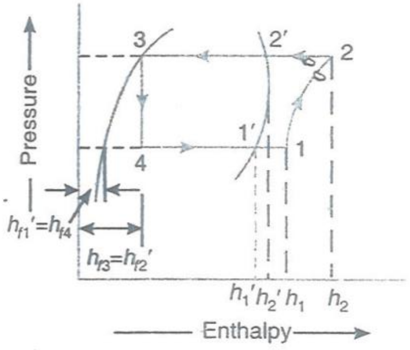
Solution. Given: P_2=5.3 bar; p_1=2.1 bar; $T_2=37^\circ C$ =37+273=310K; $T_1-T_{1'}=5^\circ C$; C_P=0.63 kJ/kg K; $T_{2'}=15.5^\circ C$ =15.5+273=288.5k; $T_{1'}=-14^\circ C$ =14+273=259k; $h_{f3}=h_{f2}$=56.15kJ/kg; $h_{f1'}$=25.12 kJ/kg; h_{fg2}=144.9kJ/kg; $h_{fg1'}$=158.7 kJ/kg

the T-s and p-h diagrams are shown in Figure

we know that enthalpy of vapour at point 1,



(a) T-s diagram.



(b) p-h diagram.

$$h_1 = h_{1'} + C_p(T_1 - T_{1'}) = (h_{f1'} + h_{fg1'}) + C_p(T_1 - T_{1'})$$

$$= (25.12 + 158.7) + 0.63 \times 5 = 186.97 \text{ kJ/kg}$$

Similarly, enthalpy of vapour at point 2,

$$h_2 = h_{2'} + C_p(T_2 - T_{2'}) = (h_{f2'} + h_{fg2'}) + C_p(T_2 - T_{2'})$$

$$= (56.15 + 144.9) + 0.63\,(310 - 288.5) = 214.6 \text{ kJ/kg}$$

$\therefore$ Coefficient of performance of the plant,

$$\text{C.O.P} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{186.97 - 56.15}{214.6 - 186.97} = \frac{130.82}{27.63} = 4.735 \text{ Ans.}$$

5. Theoretical vapour compression cycle with under cooling or subcooling of Refrigerant

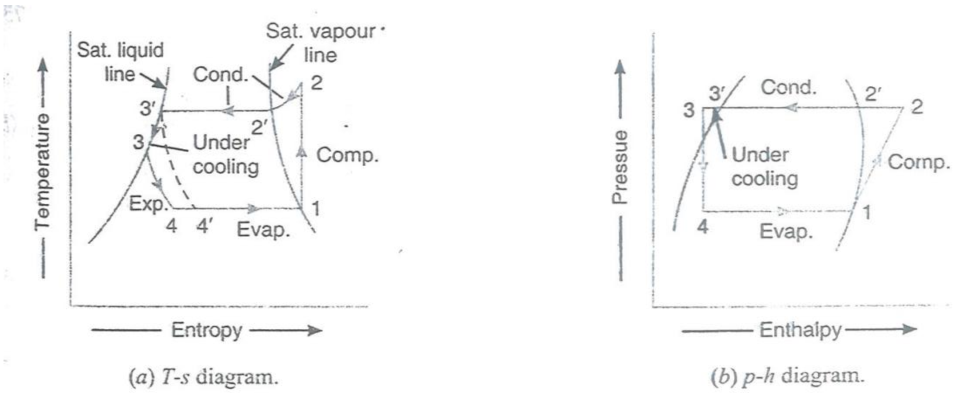


Figure10 Theoretical vapour compression cycle with under cooling or sub cooling of Refrigerant

Sometimes, the refrigerant, after condensation process 2'-3', is cooled below the saturation temperature (T_3') before expansion by throttling. such a process is called under cooling or sub cooling of the refrigerant and is generally done along the liquid line as shown in Figure 10 (a) and (b). The ultimate effect of the under cooling is to increase the value of coefficient of performance under the same set of conditions.

The process of under cooling is generally brought about by circulating more quantity of cooling water through the condenser or by using water colder than the main circulating water. Sometimes, this process is also brought about by employing a heat exchanger. In actual practice. The refrigerant is superheated after compression and under cooled before throttling, as shown in

Fig. 10 (a) and (b). A little consideration will show, that the refrigerating effect is increased by adopting both the superheating and under cooling process as compared to a cycle without them, which is shown by dotted lines in Fig. 10 (a).I n this case, the refrigerating effect or heat absorbed or extracted,

$$R_E=h_1-h_4=h_1-h_{f3} \qquad \text{--------}(\because h_4=h_{f3})$$

$$w = h_2-h_1$$

$$\text{C.O.P} = \frac{\text{Re } frigeratingeffect}{Workdone} = \frac{h_1-h_{f3}}{h_2-h_1}$$

Note: The value of h_{f3} may be found out from the relation,

$$h_{f3}=h_{f3'}-C_p \text{ x Degree of undercooling}$$

8. A vapour compression refrigerator uses R-12 as refrigerant and the liquid evaporates in the evaporator at - 15 $^\circ C$. The temperature of this refrigerant at the delivery from the compressor is 15 $^\circ C$ when the vapour is condensed at 10 $^\circ C$. find the coefficient of performance if (i) there is no under cooling, and (ii) the liquid is cooled by 5 $^\circ C$ before expansion by throttling. Take specific heat at constant pressure for the superheated vapour as 0.64 kJ/kg K and that for liquid as 0.94 kJ/kg K. the other properties of refrigerant are as follows:

Temperature in $^\circ C$	**Enthalpy in kJ/kg**		**Specific entropy in kJ/kg K**	
	Liquid	**vapour**	**Liquid**	**Vapour**
-15	**22.3**	**180.88**	**0.0904**	**0.7051**
+10	**45.4**	**191.76**	**0.1750**	**0.6921**

Solution. Given: $T_1=T_4=-15\ ^\circ C=-15+273=258K$; $T_2=15\ ^\circ C=15+273=288K$; $T_2'=10\ ^\circ C=10+273=283K$; $C_{pv}=0.64$ kJ/kg K; $C_{PL}=0.94$ kJ/kg K; $h_{f1}=22.3$ kJ/kg; $h_{f3}'=45.4$ kJ/kg; $h_{1'}=180.88$ kJ/kg; $h_2' = 191.76$ kJ/kg; $S_{f1}0.0904$ kJ/kg K; $*S_{f3}=0.1750$kJ/kg K; $S_{g1}=0.7051$ kJ/kg K; $S_2' = 0.6921$ kJ/kg K

(i) Coefficient of performance if there is no undercooling

The T-s and p-h diagrams, when there is no undercooling, are shown I fig. 4.19 (A) and (b) respectively.

Let x_1= Dryness fraction of the refrigerant at point 1.

we know that entropy at point 1,

$S_1=S_{f1}+x_1S_{fg1}=S_{f1}+x_1(S_{g1}-S_{f1})$ ----($\because S_{g1}=S_{f1}+S_{fg1}$)

$=0.0904+x_1(0.7051-0.0904)=0.0904+0.6147\ x_1$ -----------------(i)

and entropy at point 2,

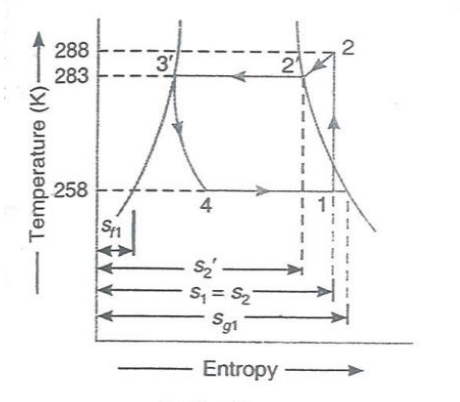
$$S_2=S_2'+2.3\ C_{pv}\log\frac{T_2}{T_2'}$$

$$=0.06921+2.3 \times 0.64 \log\frac{288}{283}$$

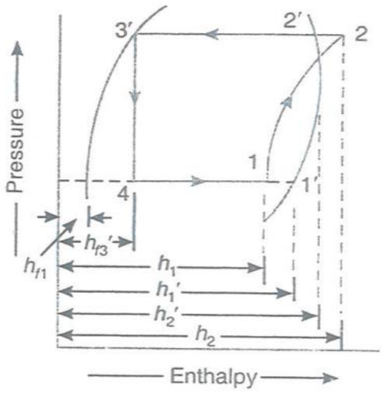
$=0.06921+2.3 \times 0.64 \times 0.0077 = 0.7034$ --------(ii)

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$0.0904+0.6147\ x_1=0.7034$ or $x_1=0.997$



(a) T-s diagram.



(b) p-h diagram.

We know that the enthalpy at point 1,

$h_1=h_{f1}+x_1h_{fg1}=h_{f1}+x_1(h_{g1}-h_{f1})$

$=22.3+0.997\ (180.88-22.3)=180.4$ kJ/kg -------($\because h_{g1}-h_1'$)

and enthalpy at point 2,

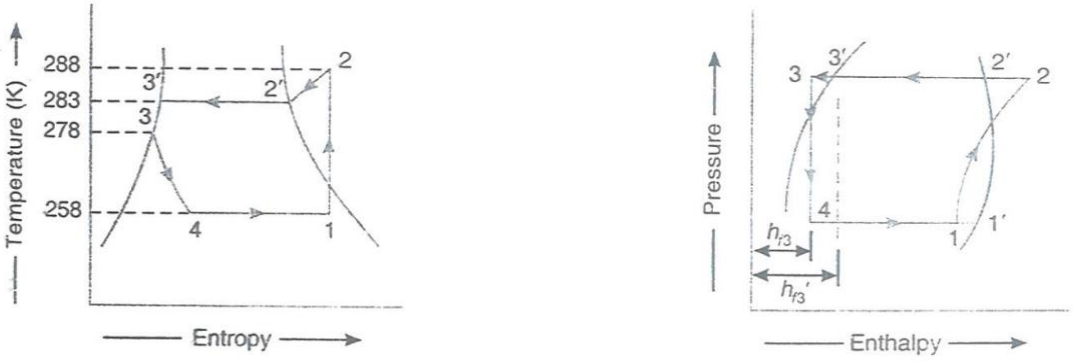
$h_2 = h_2' + C_{PV}\ (T_2 - T_2')$

$=191.76+0.64\ (288-283)=194.96$ kJ/kg

$$\text{C.O.P} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{180.4 - 45.4}{194.96 - 180.4} = 9.27 \text{ Ans.}$$

(ii) Coefficient of performance when there is an under cooling of 5 $^\circ C$

The T-s and p-h diagrams, when there is an under cooling of 5 $^\circ C$, are shown in Figure



(a) T-s diagram. (b) T-s diagram.

We know that enthalpy of liquid refrigerant at point 3,

$h_{f3} = h_{f3}' - C_{pl}$ x Degree of under cooling

$= 45.5 - 0.94 \times 5 = 40.7$ kJ/kg

$$\text{C.O.P} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{180.4 - 40.7}{194.96 - 180.4} = 9.59 \text{ Ans.}$$

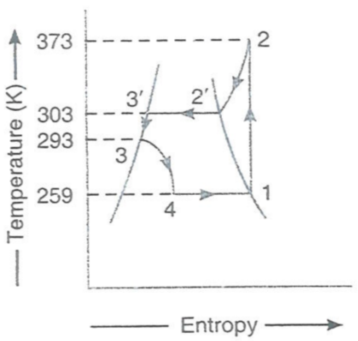
9. A simple NH_3 Vapour compression system has compressor with piston displacement of 2 m^3/min, a condenser pressure of 12 bar and evaporator pressure of 2.5 bar. The liquid is sub-cooled to 20 $^\circ C$ by soldering the liquid line to suction line. The temperature of vapour leaving the compressor is 100 $^\circ C$, heat rejected to compressor cooling water is 5000kJ/hour. and volumetric efficiency of compressor is 0.8.

Compute: Capacity: Indicated power; and C.O.P of the system.

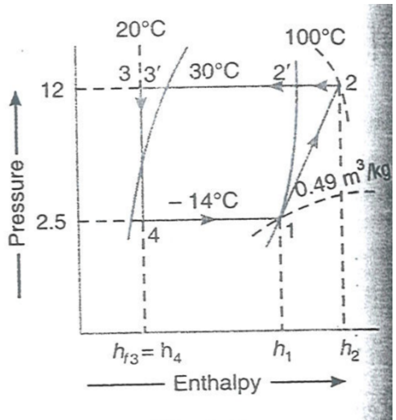
Solution: Given: $V_p=2\ m^3/min$; p2 = p2’ = p3 = 12 bar ; p1 = p4 = 2.5 bar T3 = $20^\circ C$ = 20 + 273 = 293 K ; $T_2 = 100^\circ C$=100+273=373K; η_v=0.8

Capacity of the system

The T-s and p-h diagrams are shown in Figure



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

From p-h diagram, we find that the evaporating temperature corresponding t 2.5 bas is $T_1=T_4=-14^\circ C$=-14+273 = 259K

Condensing temperature corresponding to 12 bar is

$T_2’=T_3’=30^\circ C$=30+273=303K

Specific volume of dry saturated vapour at 2.5 bar (i.e at point 1).

$V_1=0.49\ m^3/kg$

Enthalpy of dry saturated vapour at point 1,

h_1=1428 kJ/kg

Enthalpy of superheated vapour at point 2,

h_2=1630 kJ/kg

and enthalpy of sub-cooled liquid at $20^\circ C$ at point 3,

$h_{f3}=h_4=270$ kJ/kg

m_R= Mass flow of the refrigerant in kg/min.

We know that piston displacement,

$$V_p=\frac{m_R\times v_1}{\eta_v} \text{ or } m_R=\frac{v_p\times\eta_v}{v_1}=\frac{2\times 0.8}{0.49}=3.265 \text{ kg/min}$$

We know that refrigerating effect per kg of refrigerant

$=h_1-h_{f3}=1428-270=1158$kJ/kg

and total refrigerating effect $=m_R(h_1-h_2)= 3.265(1428-270)=3781$ kJ/min

$\therefore$ Capacity of the system = 3781/210 = 18 TR Ans.

Indicated power of the system

We know that work done during compression of the refrigerant

$= m_R(h_2-h_1)=3.265\ (1630-1428)=659.53$kJ/min

Heat rejected to compressor cooling water

=5000 kJ/h=5000/60=83.33 kJ/min …… (Given)

$\therefore$ Total work done by the system

=659.53 + 83.33 = 742.86 kJ/min

and indicated power of the system

=742.86/60=12.38kW Ans.

COP of the system ,We know that C.O.P of the system

$$=\frac{Total refrigerating effect}{Total work done}=\frac{3781}{742.86}=5.1 \text{ Ans}$$

COMPOUND VAPOUR COMPRESSION REFRIGERATION SYSTEMS

INTRODUCTION

Compression refrigeration system in which the low pressure vapour refrigerant from the evaporator is compressed in a single stage (or a single compressor) and then delivered to a condenser at a high pressure. But sometimes, the vapour refrigerant is required to be delivered at a very high pressure as in the case of low temperature refrigerating systems. In such cases either we should compress the vapor refrigerant by employing a single stage compressor with a very high pressure ratio between the condenser and evaporator or compress it in two or more compressors placed in series. The compression carried out in two or more compressors is called compound or multistage compression.

In vapour compression refrigeration systems, the major operating cost is the energy input to the system in the form of mechanical work. Thus any method of increasing coefficient of performance is advantageous so long as it does not involve too heavy an increase in other operating expenses, as well as initial plant cost and consequent maintenance.

Since the coefficient of performance of a refrigeration system is the ratio of refrigerating effect to the compression work, therefore the coefficient of performance can be increased either by increasing the refrigerating effect or by decreasing the compression work. A little consideration will show that in a vapour compression system, the compression work is greatly reduced if the refrigerant is compressed very close to the saturated vapour line. This can be achieved by compressing the refrigerant in more stages with intermediate inter cooling. But it is economical only where the pressure ratio is considerable as would be the case when very low evaporator temperatures are desired or when high condenser temperature may be required. The compound compression is generally economical in large plants.

ADVANTAGES OF COMPOUND (OR MULTI-STAGE) VAPOR COMPRESSION WITH INTERCOOLER

Following are the main advantages of compound or multi-stage compression over single stage compression:

1. The work done per kg of refrigerant is reduced in compound compression with intercooler as compared to single stage compression for the same delivery pressure.
2. It improves the volumetric efficiency for the given pressure ratio.
3. The sizes of the two cylinders (i.e. high pressure and low pressure) may be adjusted to suit the volume and pressure of the refrigerant.
4. It reduces the leakage loss considerably.
5. It gives more uniform torque, and hence a smaller size flywheel is needed.
6. It provides effective lubrication because of lower temperature range.
7. It reduces the cost of compressor.

TYPES OF COMPOUND VAPOUR COMPRESSION WITH INTERCOOLER

In compound compression vapour refrigeration systems, the superheated vapour refrigerant leaving the first stage of compression is cooled by suitable method before being fed to the second stage of compression and so on. Such type of cooling the refrigerant is called inter cooling. Though there are many types of compound compression with intercoolers, yet the following are important from the subject point of view:

1. Two stage compression with liquid intercooler.
2. Two stage compression with water intercooler.
3. Two stage compression with water intercooler, liquid sub cooler and liquid flash chamber.
4. Two stage compression with water intercooler, liquid sub cooler and flash intercooler.
5. Three stage compression with flash chambers.
6. Three stage compression with water intercoolers.
7. Three stage compression with flash intercoolers..

TWO STAGE COMPRESSION WITH LIQUID INTERCOOLER

The arrangement of a two stage compression with liquid intercooler is shown in figure and the corresponding p-h diagram is shown in figure

The various points on the p-h diagram are plotted as discussed below:

1. First of all, draw a horizontal pressure line representing the evaporator pressure p_e (or suction pressure of low pressure compressor) which intersects the saturated vapour line

at point 1. At this point, the saturated vapour is supplied to the low pressure compressor. Let, at point 1, the enthalpy of the saturated vapour is h1 and entropy s.

2. The saturated vapour refrigerant admitted at point 1 is compressed isentropic ally in the low pressure compressor and delivers the refrigerant in a superheated state. The pressure rises from pe to p2. The curve 1-2 represents the isentropic compression in the low pressure compressor. In order to obtain point 2, draw a line from point 1, with entropy equal to along the constant entropy line intersecting the intermediate pressure line at point 2. Let enthalpy at this point is h.
3. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is cooled (or desuperheated) at constant pressure $p_2=p_3$ in a liquid intercooler by the liquid refrigerant from the condenser. The refrigerant leaving the liquid intercooler is in saturated vapour state. The line 2-3 represents the cooling or desupertheating process. Let the enthalpy and entropy at point 3 is h and s respectively.
4. The dry saturated vapour refrigerant now supplied to high pressure compressor where it is compressed isentropic ally from intermediate or intercooler pressure ps to condenser pressure. The curve 3-4 represents the isentropic compression in the high pressure compressor. The point 4 on the p-h diagram is obtained by drawing a line of entropy equal to along the constant entropy line as shown in fig.11 (b). let the enthalpy of superheated vapor refrigerant at point 4 is h_4.

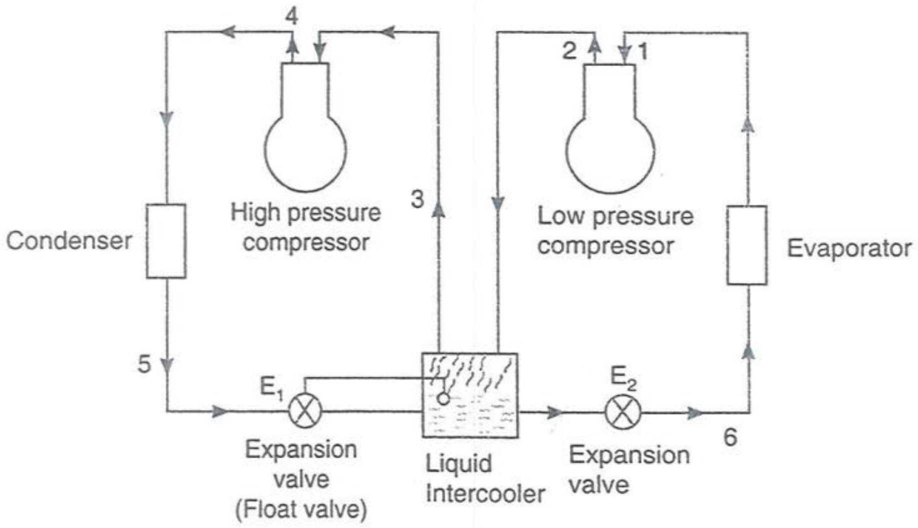


Figure11 Two stage compression with liquid intercooler

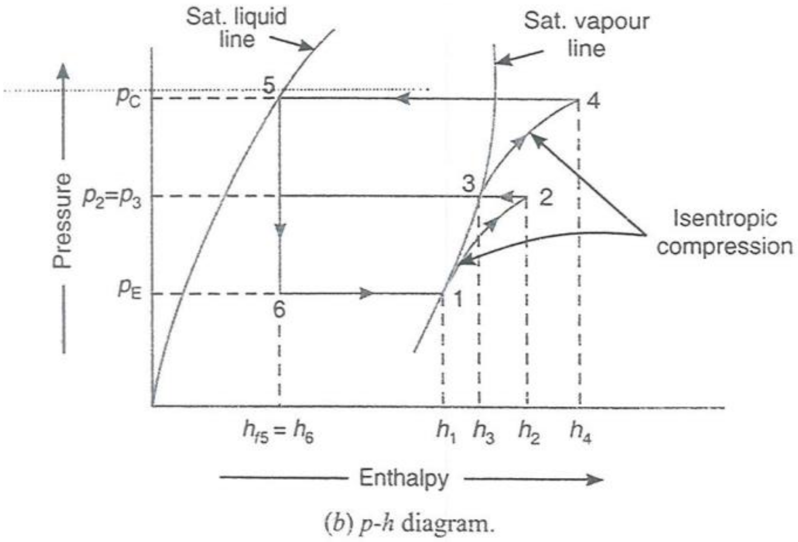


Figure12 P-h Diagram of Two stage compression with liquid intercooler

5. The superheated vapor refrigerant leaving the high pressure compressor at point 4 is now passed through the condenser at constant pressure pc as shown by a horizontal line 4-5. The condensing process 4-5 changes the state of refrigerant from superheated vapour to saturated liquid.
6. The high pressure saturated liquid refrigerant from the condenser is passed to the intercooler where some of liquid refrigerant evaporates in desupertheating the superheated vapour refrigerant from the low pressure compressor. In order to make up for the liquid evaporated i.e. to maintain a constant liquid level, an expansion value E which acts as a float value is provided.
7. The liquid refrigerant from the intercooler is first expansion value E_2 and then evaporated in the evaporator to saturated vapour condition as shown in fig. 5.1(b).

M_1 = mass of refrigerant passing through the evaporator
(or low pressure compressor) in kg/min, and
M_2 = mass of refrigerant passing through the condenser
(or high pressure compressor) in kg/min.

The high pressure compressor in a given system will compress the mass of refrigerant from low pressure compressor (m) and the mass of liquid evaporated in the liquid intercooler during cooling or desupertheating of superheated vapour refrigerant from low pressure compressor. If m is the mass of liquid evaporated in the intercooler, then

$$M_1 = m_2 - m_1$$

The value of m2 may be obtained by considering the thermal equilibrium for the liquid intercooler as shown in fig. 10, i.e.

Heat taken by the liquid intercooler = heat given by the liquid intercooler

$$M_2h_{f5}+m_1h_2=m_1h_6+m_2h_3$$

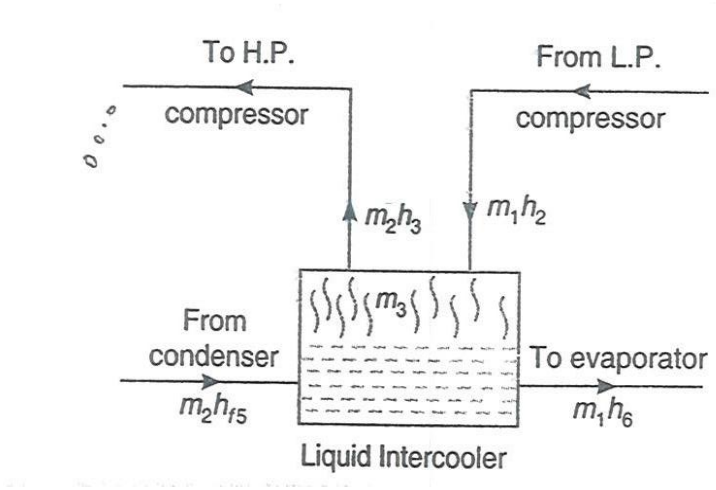


Figure13 Thermal equilibrium for liquid intercooler

$$m_2 = \frac{m_1(h_2 - h_6)}{h_s - h_{f5}} = \frac{m_1(h_2 - h_{f5})}{h_3 - h_{f5}} - m_1 = \frac{m_1(h_2 - h_3)}{h_3 - h_{f5}}$$

and mass of liquid refrigerant evaporated in the intercooler,

$$m_3 = m_2 - m_1 = \frac{m_1(h_2 - h_{f5})}{h_3 - h_{f5}} - m_1 = \frac{m_1(h_2 - h_3)}{h_3 - h_{f5}}$$

We know that refrigerating effect,

$$R_E = m_1(h_1 - h_f) = m_1(h_1 - h_{f5}) = 210QkJ / \min$$

where Q is the load on the evaporator in tonne of refrigeration.

Total workdone in both the compressors.

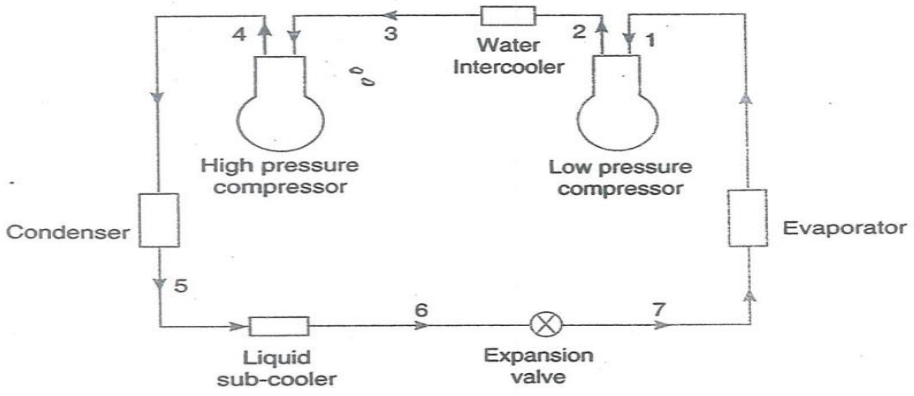
$$P = \frac{m_1(h_2 - h_1) + m_2(h_4 - h_3)}{60} kW$$

and C.O.P. of the system $= \frac{R_E}{W} = \frac{m1(h_1 - h_{f5})}{m_1(h_2 - h_1) + m_2(h_4 - h_3)} \frac{210Q}{P \times 60}$

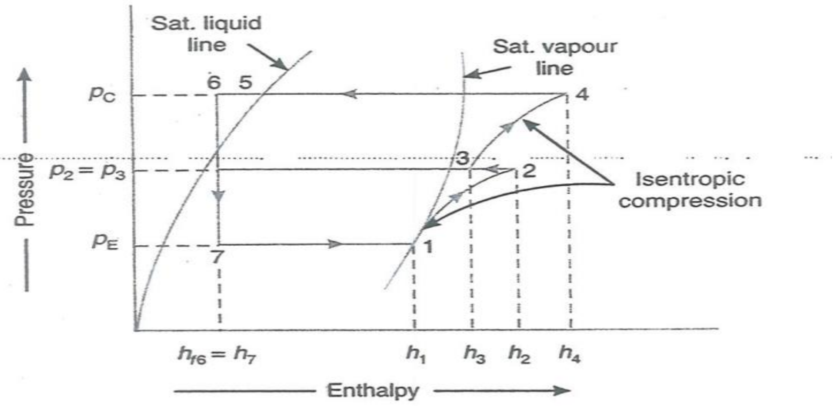
TWO STAGE COMPRESSION WITH WATER INTERCOOLER AND LIQUID SUB-COOLER

The arrangement of a two-stage compression with water intercooler and liquid sub-cooler is shown in fig14 (a). the corresponding p-h diagram is shown in fig. 14 (b). the various processes in this system are as follows:

1. The saturated vapour refrigerant at the evaporator pressure p_E is admitted to low pressure compressor at point 1. In this compressor, the refrigerant is compressed isentropic ally from the evaporator pressure p_e to the water intercooler pressure p_2, as shown by the curve 1-2 in fig14 (b).
2. The refrigerant leaving the low pressure compressor at point 2 is in superheated state. This superheated vapour refrigerant is now passed through the water intercooler at constant pressure, in order to reduce the degree of superheat. The line 2-3 represents the water intercooling or desuperheating process.
3. The refrigerant leaving the water intercooler at point 3 (which is still in the superheated state) is compressed isentropic ally in the high pressure compressor to the condenser pressure pe. The curve 3-4 shows the isentropic compression in high pressure compressor.
4. The discharge from the high pressure compressor is now passed through the condenser which changes the state of refrigerant from superheated vapour to saturated liquid as shown by process 4-5.
5. The temperature of the saturated liquid refrigerant is further reduced by passing it through a liquid sub-cooler as shown by process 5-6.
6. The liquid refrigerant from the sub-cooler is now expanded in an expansion value (process 6-7) before being sent to the evaporator for evaporation (process 7-1).



(*a*) Two stage compression with water intercooler and liquid sub-cooler.



(*b*) *p-h* diagram.

Figure14 Two stage compression with water intercooler and liquid sub-cooler

It may be noted that water intercooling reduces the work to be done in high pressure compressor. It also reduces the specific volume of the refrigerant which requires a compressor of less capacity (or stroke volume). The complete desuperheating of the vapour refrigerant is not possible in case of water intercooling. It is due to the fact that temperature of the cooling water used in the water intercooler is not available sufficiently low so as to desuperheat the vapour completely.

Let Q = Load on the evaporator in tonnes of refrigeration.

$\therefore$ Mass of refrigerant passing through the evaporator (or passing through the L.P. compressor),

$$m = \frac{210Q}{h_1 - h_7} = \frac{210Q}{h_1 - h_{f6}} kg/\min \ldots (\because h_7 = h_{f6})$$

Since the mass of refriger5ant passing through the compressors is same, therefore, total work done in both the compressors,

W=Work done in L.P. compressor + Work done in H.P. compressor

$$= m(h_2 - h_1) + m(h_4 - h_3) = m[(h_2 - h_1) + (h_4 - h_3)]$$

$\therefore$ Power required to drive the system.

$$P = \frac{m[(h_2 - h_1) + (h_4 - h_3)]}{60} kW$$

We know that refrigerating effect,

$$R_E = m(h_1 - h_{f6}) = 210QkJ/\min$$

$$\therefore \text{C.O.P. of the system } = \frac{\mathrm{R}_E}{W} = \frac{m(h_1 - h_{f6})}{m[(h_2 - h_1) + (h_4 - h_3)]} = \frac{210Q}{P \times 60}$$

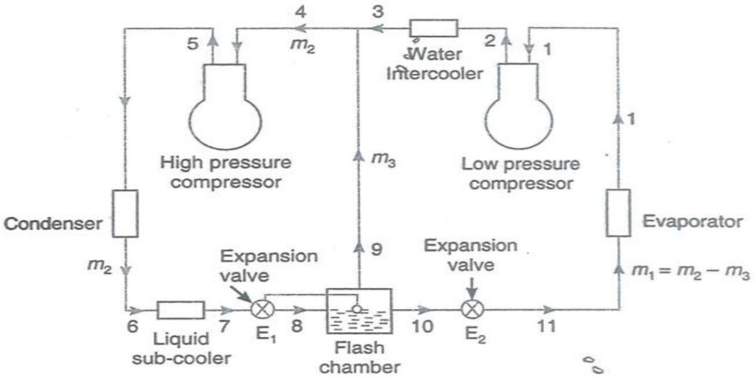
TWO STAGE COMPRESSION WITH WATER INTERCOOLER, LIQUID SUB-COOLER AND LIQUID FLASH CHAMBER

The arrangement of a two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber is shown in fig. 15(a). the corresponding p-h diagram is shown in fig. 15(b). the various processes, in this system, are as follows:

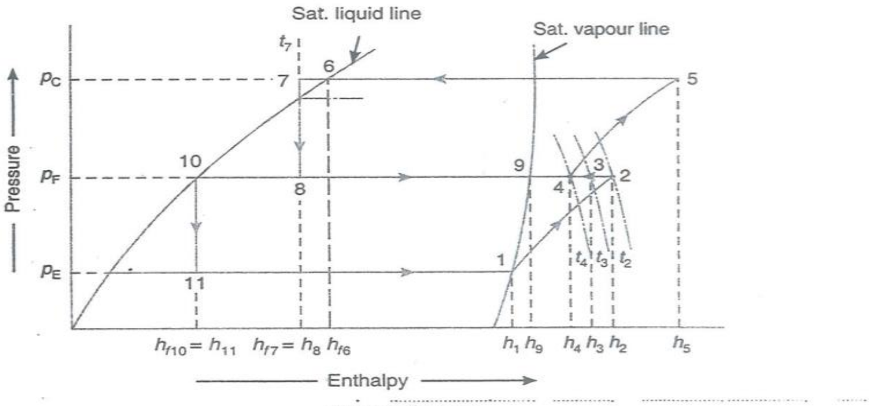
1. The saturated vapour refrigerant at the evaporator pressure p_e is admitted to low pressure compressor at point 1. In this compressor, the refrigerant is compressed isentropically from evaporator pressure p_e to water intercooler (or flash chamber) pressure p_f as shown by the curve 1-2 in fig. 15 (b).
2. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is now passed through the water intercooler at constant pressure pf in order to reduce the degree of superheat (i.e, from temperature t_2 to t_3). The line 2-3 represents the water intercooling or de-superheating process.
3. The superheated vapour refrigerant leaving the water intercooler at point 3 is mixed with the vapour refrigerant supplied by the flash chamber at point 9. The condition of

refrigerant after mixing is shown by point 4 which is in superheated state. Let the temperature at this point is t_4.

4. The superheated vapour refrigerant admitted at point 4 to the high pressure compressor is compressed isentropically from the intercooler or flash chamber pressure p_f to condenser pressure p_c as shown by the curve 4-5. The temperature rises from t_4 to t_5.
5. The superheated vapour leaving the high pressure compressor at pressure pc is passed through a condenser at constant pressure as shown by a horizontal line 5-6. The condensing process 5-6 changes the state of refrigerant from superheated vapour to saturated liquid.
6. The saturated liquid refrigerant from the condenser is now cooled in liquid sub-cooler to a temperature, say t_7. The line 6-7 represents a sub-cooling process.



Two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber.



(*b*) *p-h* diagram.

Figure15 Two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber

7. The liquid refrigerant leaving the sub-cooler at pressure p_c is expanded in an expansion value E_1 to a pressure equal to the flash chamber pressure pf as shown by vertical line 7-8. The expanded refrigerant which is a mixture of vapour and liquid refrigerants is admitted to a flash chamber at point 8. The flash chamber separates the vapour and liquid refrigerants at pressure. The vapour refrigerant from the flash chamber at point 9 is mixed with the refrigerant from the water intercooler. The liquid refrigerant from the flash chamber at point 10 is further expanded in an expansion value E_2 as shown by the vertical line 10-11.
8. The liquid refrigerant leaving the expansion value E_2 is evaporated in the evaporator at the evaporator pressure pe (usually 2 bars) as shown by the horizontal line 11-1 in fig. 5.7 (b)

Let M_2= mass of refrigerant passing through the condenser or high Pressure compressor), and
M_3=mass of vapor refrigerant formed in the flash Chamber.

∴ Mass of refrigerant passing through the evaporator (or low pressure compressor.)

$$M_1=m_2-m_3$$

If *Q* tonne of refrigeration is the load on the evaporator, then the mass of refrigerant passing through the evaporator.

$$m1=\frac{210Q}{h_1-h_{11}}=\frac{210Q}{h_1-h_{f10}}\,kg/\min \qquad \ldots(\because h_{11}=h_{f10})$$

Now let us consider the thermal equilibrium of the flash chamber. Since the flash chamber is an insulated vessel, therefore there is no heat exchange between the flash chamber and atmosphere. In other words, the heat taken and given by the flash chamber are same. Mathematically.

Heat taken by the flash chamber = heat given by the flash chamber

or $$m_2h_8=m_3h_9+m_1h_{f10}$$

$$=m_3h_9+(m_2-m_3)h_{f10} \qquad \ldots(\because m_1=m_2-m_3)$$

$$m_2(h_8 - h_{f10}) = m_3(h_9 - h_{f10})$$

$$m3 = m2\left(\frac{h_8 - h_{f10}}{h_9 - h_{f10}}\right) = m_2\left(\frac{h_{f7} - h_{f10}}{h_9 - h_{f10}}\right) \qquad \ldots(\because h_8 - h_{f7})\ldots(i)$$

The vapour refrigerant from the water intercooler (represented by point 3) is mixed with vapour refrigerant m3 from the flash chamber (represented by point 9) at the same pressure before entering the high pressure compressor. The enthalpy of the mixed refrigerant (represented by point 4) may be calculated by using the equation.

$$m_2 h_4 = m_3 h_9 + m_1 h_3$$

$$= m_3 h_9 + (m_2 - m_3)h_3$$

We know that refrigerating effect of the system.

$$R_E = m_1(h_1 - h_{11}) = 210Qkj / \min$$

Work done in low pressure compressor, $W_L = m_1(h_2 - h_1)$

Work done in high pressure compressor, $W_H = m_2(h_5 - h_4)$

Total workdone in both the compressors,

$$W = W_L + W_H = m_1(h_2 - h_1) + m_2(h_5 - h_4)$$

$\therefore$ Power required to drive the system.

$$P = \frac{M_1(h_2 - h_1) + m_2(h_5 - h_4)}{60} kW$$

and C.O.P. of the system $= \frac{\mathrm{R}_E}{W} = \frac{m_1(h_1 - h_{11})}{m_1(h_2 - h1) + m_2(h_5 - h_4)} = \frac{210Q}{P \times 60}$

REFRIGERANTS

INTRODUCTION

The refrigerant is a heat carrying medium which during their cycle (i.e comparison, condensation, evaporation) in the refrigeration system absorbs heat from allow temperature system and discards the eat so absorbed to higher temperature system.

The natural ice and mixture of ice and salt were the first refrigerants. In 1834, edger ammonia, sulphur dioxide, methyl chloride and carbon dioxide came into use as refrigerants in compression cycle refrigeration machines. Most of the early refrigerant materials have been discarded for safety reasons or for lack of chemical or thermal stability. In the present days, many new refrigerants including halo-carbon compounds, hydro-carbon compounds are used for air –condition and refrigeration applications.

The suitability of a refrigerant for a certain application is determined by its physical thermodynamic, chemical properties and by various practical factors. There is no one refrigerant which can be used for all types of applications i.e there is no ideal refrigerant. It one refrigerant has certain good advantages, it will have some disadvantages also. Hence, a refrigerant is chosen which has greater advantages and less disadvantages.

DESIRABLE PROPERTIES OF AN IDEAL REFRIGERANT

A refrigerant is said to be ideal it it has all of the following properties:

1. Low boiling and freezing point,
2. High critical pressure and temperature,
3. High latent heat of vaporization,
4. Low specific heat of liquid, and high specific heat of vapour,
5. Low specific volume of vapour,
6. High thermal conductivity,
7. Non-corrosive to metal,
8. Non-flammable and non-explosive,
9. Non-toxic
10. Low cost,
11. Easily and regularly available,
12. Easy to liquefy at moderate pressure and temperature,
13. Easy to locating leaks by our or suitable indicator,
14. Mixes well with oil,
15. High coefficient of performance, end
16. Ozone friendly.

CLASSIFICATION OF REFRIGERANTS

The refrigerants may, broadly, be classified in the following two group:

1. Primary refrigerants and 2. Secondary refrigerant

The refrigerants which directly take part in the refrigeration system are called primary refrigerants whereas the refrigerants which are first cooled by primacy refrigerants and then used for cooling purposes as known as secondary refrigerants.

The primary refrigerates are further classified into the following four groups:

1. Halo-carbon or organic refrigerants,
2. Azoetrope refrigerants,
3. Inorganic refrigerants, and
4. Hydro-carbon refrigerants

These above mentioned refrigerants are discussed, in detail, in the following pages.

HALO-CARBON REFRIGERANTS

The American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) identifies 42 halo-carbon compounds as refrigerants, but only a few of them are commonly used. The following table gives some of the commonly used halo-carbon refrigerants:

1. R-11, Trichloro-monofluoro-methane (CCl3F). the R-11 is a synthetic chemical product which can be used as a refrigerant. It is stable, non-flammable and one- toxic. It is considered to be low-pressure refrigerant. It has a low side pressure of 0.202 bar at - 15°C and high side pressure of 1.2606 bar at 30°C. The latent heat at - 15°C is 195 kJ/kg. The boiling point at atmospheric pressure is 23.77°C. Due to its low operating pressures, this refrigerant is exclusively used in large centrifugal compressor stems of 200 TR and above. The leaks may be detected by using a soap solution, a halide torch or by using an electronic detector.

   R-11 is often used by service technicians as a fishing agent for cleaning the internal parts of a refrigerator compressor when overhauling systems. It is useful after aa system had a motor burn our or after it has great deal of moisture in the system. By flushing moisture from the system with R-11, evacuation time is shortened. R-11 is one of the safest cleaning solvents that can be used for this purpose. The cylinder colour code for R-11 is orange.

2. R-12, Dichloro-diflluoro-methane (CCl_2F_2). The R-12 is a very popular refrigerant. It is a colourlesss, almost odorless liquid with boiling point of -29°C at atmospheric pressure. It is non-toxic, non-corrosive, non-irritating and non-flammable. It has a relatively how latent heat value which is an advantage is small refrigerating machines. The large amount of

refrigerant circulated will permit the use of less sensitive and more positive operating and regulating mechanisms. It operates at allow but positive head and back pressure and with a good volumetric efficiency. This refrigerant is used in many different types of industrial and commercial applications such as refrigerators, freezers, water coolers, room and window air-conditioning units etc. Its principal use is found in reciprocating and rotary compressor. But its use in centrifugal compressor for large commercial air-conditioning is increasing.

R-12 has a pressure of 0.82 bar at - 15°C and a pressure of 6.4 bar at 30°C. the latent heat of R-12 at - 15°C is 159 kJ/kg. the leak may be detected by soap solution, halide torch or an electronic leak detector. Water is only slightly soluble in R-12. At -18°C it will hold six parts per million by mass. The solution formed is very slightly corrosive to any of the common metals used in refrigerator construction. The addition of mineral oil to the refrigerant has no effect upon the corrosive action.

R-12 is more critical as to its moisture content when compared to R-22 and R-502. It is soluble in oil down to- 68°C. the oil will begin to separate at this temperature and due to its lightness than the refrigerant, it will collect on the surface of the liquid refrigerant. The refrigerant an available in a variety of cylinder sixe and the cylinder colour code is white.

3. R-13, Monochloro-trifluore-methane ($CClF_3$). The R-13 has a boiling temperature of 81.4°C at atmospheric pressure and a critical temperature of + 28.8°C. This refrigerant is used for the low temperature side of cascade systems. It is suitable with reciprocating compressors.
4. R-14, Carbontetrafluoride (CF_4). The R-14 has a boiling temperature of - 128°C at atmospheric pressure and critical temperature of -45.5°C. It serves as an ultra-low temperature refrigerant for use in cascade systems.
5. R-21, Dichloro-monofluoro-methane ($CHCl_2F$). The R-21 has a boiling temperature of +9°C at atmospheric pressure. It has found its principal use in centrifugal compressor systems for relatively high temperate refrigeration requirements.
6. R-22, Monochloro-difluoro-methane ($CHClF_2$). The R-22 is a man-made refrigerant developed for refrigeration installations that need low evaporating temperatures, as in fast freezing unit which maintain a temperature of -29°C to -40°CC. it has also been successfully used in air-conditioning units and in household refrigerators. It is use with reciprocating and centrifugal compressors. It is not necessary to use R-22 at below atmospheric pressures in order to obtain the low temperatures.

The boiling point of R-22 is -41°C at atmospheric pressure. It has a latent heat of 216.5 kJ/kg at -15°C. The normal head pressure at 30°C is 10.88 bar. These refrigerants stable and are non –toxic, non-corrosive, non-irritating and non-flammable. The evaporator pressure of this refrigerant at -15°C is 1.92 bar. Since water mixes better with R-22 than

R-12 by a ratio of 3 to 1, therefore driers (dessicants0 should be used to remove most of the moisture to keep water to a minimum. This refrigerant has good solubility in oil sown to -9°C. However, the oil remains flow enough to flow down the suction line at temperatures as low as -40°C. The oil will begin to separate at this point. Since oil is lighter, therefore it will collect on the surface of the liquid refrigerant. The leaks may be deiced with a sop solution, a halide torch or with an electronic leak detector. The cylinder colour code for R-2 is green.

7. R-30, Methylene chloride (CH_2Cl_2). The R-30 is a clear, water-white liquid with sweet, non-irritating odour similar to that of chloroform. It has a boiling point of 9.8°C at atmospheric pressure. It is non-flammable, non-explosives and non-toxic. Due to its high boiling point, this refrigerant may be stored in cloded cans instead of in compressed gas cylinders. This high and low side of refrigeration system using R-30 operates under a vacuum. Since the volume of vapour at suction conditions is very high, therefore the use of R-30 is restricted to rotary or centrifugal compressor. This refrigerant was extensively used for air conducting of theatres auditorium, and office buildings. Now –a days, the refrigerant R-11 is used in place of R-30.

   In order to detect leaks in a system using R-30, the pressure must be increased above atmosphere. A halide torch is used for detecting leaks.

8. R-40, Methyl-chloride (CH3Cl). The R-40 is colorless liquid with a faint, wet, and non-irritating odour. Its boiling pint at atmospheric pressure is -23.7°C and the usual condenser pressure is 5 to 6.8 bar. The latent heat of vaporization at - 15°C is 423.5kJ/kg. it is flammable and explosive when mixed with air in concentrations from 8.1 to 17.2 per cent. This regieranct is non-corrosive in its pure state, but it becomes corrosive in the presence of moisture. Aluminum zinc and magnesium alloys should never be used with thus refrigerant as they will cored considerably and pollute the lubricating oil. since the refrigerant R-40 is a solvent for many materials used in ordinary refrigeration compressor, therefore rubber and gaskets containing rubber and gaskets containing rubber should never be used. However, synthetic rubber is not affected byR-40. Thus metallic asbsstor-fibre gaskets containing insoluble binders should be used. The mineral oils are soluble in this refrigerant to a small extent.

   This refrigerant has been used in domestic units with both reciprocating and rotary compressors and in commercial units with reciprocating compressors up to approximately 10 TR capacities. The leaks with R-40 may be detected by soap solution or electronic leak detector.
9. R-100, Ethyl chloride (C_2H_5Cl). The R-100 is a colure less liquid and in many respites it is similar to R-40 (Methyl Chloride) but with low operating pressures. It has b a boiling point of 13.1°C at atmospheric pressure. It is both toxic and flammable. Due to it low operating pressure, it is not used in refrigerating equipment.

10. R-113, Tricholor-trifluoro-etane ($CC_{l2}FCClF_2$ or C_2ClE_3). The R-113 has a boiling point of 47.6°C at atmospheric pressure. It is used in commercial and industrial air-conditioning with centrifugal compressor systems. Since this refrigerant has the advantage of remaining liquid at room temperatures and pressures, therefore it can be carried in sealed tins rather than cylinders.

11. R-114, Dichloro-tetrafluoro-ethane ($CClF_2$ or $C_2Cl_3F_3$). The R-114 has a oblong point of 3.6°C at atmospheric pressure. At-15°C,it evaporates at pressure of 0.54 bar boiling and at +30°C it condenses at a pressure of 1.5 bar. Its latent heat of vaporization at -15°C is 143kJ/kg. it is non-toxic, non-explosive and non corrosive even in the presence of water. It is used in fractional power household refrigerating system and drinking water coolers employing rotor vane type compressors.

12. R-123., Dichloro- trifluore –methane (CF_3CHCl_2). The R-123 is a potential substitute to R-11. It has about 4.3°C higher boiling postman R-11. It is, therefore, a tower pressure replacement for R-11, thus having larger specific volume of suction vapourt. Hence, its use results in 10 to 15%reduion in capacity, it used in capacity, if used in existing R-11 centrifugal compressors.

13. R-134 ,Tetrafluoro-ethane (CF3CH2F0. The R134a is considered to be the most preferred substitute for refrigerant R-12. Its boiling point is -26.15 °C which is quite close to the boiling point of R-12 which is -29°C at atmospheric pressures. Since the refrigerant R-134 has no chlorine atom, there for this refrigerant has zero * ozone depleting potential (ODP) and has 74% less global warming potential (GWP) as compared to R-12. It has lower suction pressure and large suction vapor volume sis not soluble in mineral oil (polyester based) is used. Care should be taken to prevent moisture from getting into their figuration system. For use in existing R-12 reciprocating compressors, it would require either an average increases in compressors speed of 5 to 8% or an equivalent increases in cylidervalum. Since the moledecules of R-134a are smaller than R-12, therefore Avery sensitive leak detector is used to detect leaks. .
    14.R-152a, Difluoro- enhance (CH3CHF2). The R-15a has similar characteristic as R 134a except that R-152 has slight vacuum in the evaporator at -25°C and the discharge temperature is higher because of its high value of the ratio of specific heats (y)

AZEOTROPE REFRIGERANTS

The term“Azeotrope’ refers to stabel mixture of refrigerants whose vapour and liquid phases retain identical compositions over a wide range of temperatures. However, the these mixtures, usually, have properties that differ from either of the components. Some of the Azeotrope are given is the following table.

Table 1 Azeotrope refrigerants

Refrigerant number	Azetrope mixing refrigerant	Chemical formula
R-500	73.8% R-12 and 26.2% R-152	CCL_2F_2/CH_3CHF_2
R-502	48.8% R-22 and 51.2% R-115	$CHCIF_2/CCIF_2CF_3$
R-503	40.1% R-23 and 59.9% R-13	$CHF_3/CCLF_3$
R-504	48.2% R-32 and 51.8% R-115	$CHF_2F_2/CCLF_2CF_3$

These refrigerants are discussed, in discussed, in detail, as below:

1. R-500. The R-500 is an Azeotrope mixture of 73.8% R-12 (CCl2F2) and 26.2% R-152 (CH2CHF2). It is non-flammable, low in toxicity and non- corrosive. It is used in both industrial and metrical applications but only in systems enteric oration compressors. It has a fairly constant vapor presser temperature curve which is different from the vaporizing curves for either R-152a or R-12. This refrigerant officers about 20% greater refrigerating capacity than R-12 for the same size of motor when used for the same purpose. The evaporator pressure of this refrigerant is 1.37 bar at-15°C is 192 kJ/Kg. it can be used whenever a higher capacity that that obtained with R-12 is needed. The sedulity of water in R-500 is highly critical. It has fairly high salinity with oil. The leakage may be erected by using soap solution, a halide. I this fairly high solubituyt with oil. The leakage may be detected by using soap solution, a halide torch, an electronic teak detector or a cored tracing agent. The servicing refrigeration using that refrigerant does not present any unusual problem. Water is quite soluble in the regieranct. It necessary to keep moisture out of the system by careful dehydration and by using dryers. The cylinder colour code for this refrigerant is yellow.
2. R-502. The R-502 is an azeotopic-misture of 48.8% R-22 ($CHCLF_2$) and d51.2% R-115 ($CCLF_2CF_3$). It is non-flammable, non-corrosive, practically non-toxic liquid. It is a good refrigerant of obtaining medium and low temperature. It is suitable where temperatures from -18°C to -15°C are needed. It is often used in frozen food lockers, frozen food processing plant frozen food displeases and it storage in its for frozen foods and ice-cream. It is only used with reciprocating compressors. The boiling point of this refrigerant at atmospheric pressures is -46°C. its evaporating pressure at -15°C is 2.48 bar and the condensing pressure at 30°C is 12.6. its latent het at-29°C is 168.6 kJ/Kg.The R-502 combines many of the good properties of R-12 and R-22. It gives a machine capacity equal to that of R-22 with just about the condemn sing temperature of a system using R-12. Since this refrigerant has a relatively low condensing pressure and temperature, therefore increases the light of compressor values and other parts. Better lubricating is possible because of the increases

viscosity of the oil at low condensing temperature. It is possible to eliminate liquid injection to cool the compressor because of the low condensing pressure. This refrigerant has all the qualities found in other halogenated (fluorocarbon) regirgernt. It is non-toxic. Non-flammable, non-irradiating, stable and non-corrosive. The leaks may be detected by soap solution, halide torch or electronic leak detector. It will hold 1.5 times more moisture at 18°C than R-12. It has fair solubility in oil above 82°C. Below this temperature, the oil tries to separate and tends to collect on the surface of the liquid refrigerant. However, oil is carried to the compressor at temperature down to -40°C. Special device are sometimes used to the oil to the compressors. The cylinder color code for this refrigerant is orchid.

3. R-503. This anAzeotrope mixture of 40.1% R-23 (CHF_3) and 59.9% of R-1 ($CCIF_3$). This is a non-flammable, non-corrosive, practically non-toxic liquid. Its temperature at atmospheric pressure is -88°C which is lower than either R-23 or R-13. It evaporating pressure at -15°C is 17.15 bar. Its critical temperature is 20°C and its critical pressure is 14.15 bar. This is a low temperature refrigerant and good for use in the low state of cascade systems which enquire temperatures in the range of -73°C to 87°C. The latent heat of vaporization at atmospheric pressure is 173 kJ/Kg. The leaks in R-503 system may be detected with the use of soap solution, a halide torch an electronic leak detector. This refrigerant wills hold more mister than some other temperature refrigerants. It may be not that all low temperature applications must have control devices. The oil does not circle are well a low temperatures. The cascade and other lower temperatures equipments are normally fleeted will oil separators and other devices of retuning the oil to the compressor the colder colure code for R-503 is aquamarine.

4. R-504. The R-504 is an azetropic mixture of 48.2% R-32 (CH2F2) and 51.8 % R-115 (CCIF2CF3). It non- flammable, non-corrosive and non-toxic.the boiling g temperature at atmospheric pressure is -57°c. its evaporating pressure at -15°C is 5.88 bar and its critical with the oil circulation. With the addition of 2 to 5% R-170 (ethane), the oil will be taken into the solution with the refrigerant and will circle through the system with it. The leaks in R-504 system may be easily detected by using soap solution, a halide torch or an electronic leak detector. This refrigerant is used in industrial processes where a low temperature range of -40°C to -62°C is desired. The cylinder color code for R-504 is ton. The leaks in R-504 systems may be classily detected by using soap solution, a halide torch or an electronic leak detector. This refrigerant is used in industrial processes where a low temperature range of -40°C to - 62°C is desired. The cylinder color code for R-504 is tan.

INORGANIC REFRIGERANTS

The inorganic refrigerant swore exclusively used before the introduction of halo-carbon refrigerants. These regiments are still is use due to their inherent thermodynamic and physical properties. The vacuous inorganic regiments are given in the flowing table:

Table 2 inorganic refrigerants

Refrigerant number	Chemical name	Chemical formula
R-171	Ammonia	NH3
R-729	Air	--
R-744	Carbon dioxide	CO_2
R-764	Sulphur dioxide	SO_2
R-118	Water	H_2O

These refrigerants are discussed, in detail, as below:

1. R-717 (Ammonia). The R-717, i.e ammonia (NH3) is one the oldest and most widely used of l the reigigerants. Its greatest applications found in large and commercial reciprocating compression suttees where high toxicity is secondary. It is also widely in absorption systems. It is a chemical compound of nitrogen an hydrogen and under ordinary conditions, it is a colorless gas. Its boiling point at atmospheric pressure is-33.3°C and its melting point from the solid is -78°C. the low boiling point makes it possible to have regeneration at temperatures considerably below 0°C without using pressures below atmospheric in the evaporators. Its latent heat of vaporization at -15°C is 1315 kJ/Kg. thus, large refrigerating effects are possible with relatively small sized machinery. The condenser pressure at 30°C is 10.78 bar. The conddensersofrR-717 are usually of water cooled type. It is poisonous gas if inhaled in large quantities. In lesser quantities, it is irritating to the eyes, nose and throat. This regieranct is somewhat flammable and, when mixed with air the ratio of 16% to 25% gas by volume, will form an explosive mixture. The leaks of this refrigerant may be quickly and easily detected by the use of boring sulphur candle which in the presence of ammonia forms white fumes of ammonium sulphite. This refrigerant attacks copper and bronzed in the presence of a little moisture but does not corrode iron or steel. It presents no special problems in connection temperatures are encountered. Since the refrigerant R-717 is lighter than oil, therefore, its separation does not rate any problem. The excess oil in the evaporator may be removed by opening a valve in the bottom of the evaporator. This refrigerant is used in large

compression machines using reciprocating compressors and in many absorption type systems. the use of this refrigerant is extend silvery foundin cold storage, warehouse plants, ice cram manufacture, ice manufacture, been manufacture, food freezing plants etc.

2. R-729 (Air). The dry air us used as gaseous refrigerant in some compression system, particularly in aircraft air conditioning systems, particularly in aircraft air-conditioning.
3. R-744 (Carbon dioxide). The principal refrigeration use of carbon dioxide is same as that of dry ice. It is non-toxic, non-irritating and non-flammable. The boiling point of this refrigerant is so extremely low (-73.6°C) that at -15°C, a pressure of well over 20.7 bar is enquired to pre cent its evaporation. At a condenser temperature of +30°C, a pressure of approximately 70 bar is required to liquefy the gas. Its critical temperature is 31°C and triple point is -56.6°C. due to its high operating pressure, the compressor of carbon dioxide refrigerator unities very small even for a comparatively large regenerating capacity, however, because of its low efficiency as compared to other common refrigerants, it is seldom in household units, but is used is some industrial applications and aboard ships.
4. R-764 (Sulphur dioxide). The refrigerants are produced by the combustion of sulphur in air. In the form years, it widely used in household and small commercial units. This boiling point of sulphur dioxide is -10°c at atmospheric pressure. The condensing pressure varies between 4.1 bar and 6.2 bar infer normal operating conditions. The latent heat of sulphur dioxide at -15°C is 396 kJ/kg. it is a very stable refrigerant with a high critical temperature and it is non-flammable and non-explosive. It has very unpleasant and irritating odour. This reigigerants is not injurious to food and is used commercially as a ripened and preservative of foods. It is, however, extremely injurious to flower, plants and shrubbery. The sulphur dioxide in its pure steins not corrosive, but when there is moister present, the mixture forms sulphurous accede which is corrosive to steel. Thus it is very important and that the moisture in the refrigerating system be held to a minimum. The sulphur dioxide does not mix readily withoil. Therefore, an oil light than that used with other refrigerants may be used in the compressors. The refrigerant in the evaporator with oil floating on be top has atedecy to have a higher boiling point than that corresponding to its pressure. The mourn evaporator resovercoe this by having the liquid introduced in such a way that the reigigerants is kept agitated while the unit is in operation. The leaks in the system with sulphur dioxide may be easily detected by means of soap solution or ammonia swab. A dense hate smoke forms when sulphur dioxide and ammonia fumes come in contact.
5. R-118 (water). The principle refrigeration use of water is as ice. The high freezing temperature of water limits its use in vapour compression systems. It is use as there fragrant vapour is some absorption systems and in systems with steam jet compressors.

HYDRO-CARBON REFRIGERANTS

Most of the hydro-carbon refrigerants are successfully used in industrial and commercial installations. They possess satisfactory thermodynamic properties but are highly flammable and explosive. The various hydra-carbon refrigerants are given in the following table

Table 3 Hydro-carbon refrigerants

Refrigerant number	Chemical name	Chemical formula
R-170	Ethane	$C_2 H_2$
R-290	Propane	C_3H_3
R-600	Butane	C_4H_{10}
R-600$_a$	Isobutene	C_4H_{10}
R-1120	Trichloroethylene	$C_2H_4Cl_3$
R-1130	Dichloroethylene	$C_2H_4Cl_2$
R-1150	Ethylene	C_2H_4
R-1270	Propylene	C_3H_6

DESIGNATION SYSTEM FOR REFRIGERANTS

The refrigerants are internationally designated as ‘R’ followed by certain numbers such as R-11, R-12, R-114 etc. A refrigerant followed by a two-digit numbers indicates that a refrigerant is derived from methane base while three-digit number represents ethane bases. The numbers assigned to hydro-carbon and halo- and refrigerants have a special meaning. The first digest on the rightist the number of fluorine (F) atoms in the reigigerants. The second digit from the right is one more than the number of hydrogen (H) atoms present. The third digit from the right is one less than the number of carbon (C) atoms, but when this digit is zero, it is omitted. The general chemical formula for the refrigerant, either for methane or ethane base, is given as $C_mH_nCl_pF_q$ in which $n + p + q = 2m + 2$

Where m = Number of carbon atoms,

n = Number of hydrogen atoms,

p = Number of chlorine atoms, and

q = Number of fluorine atoms.

As discussed above, the number of the refrigerants is given by R9m -1) (n + 1) (q). Let us consider the following refrigerants to find its chemical formula and the number.

1. ***Dichloro-difluoro-methane***

We see that in this refrigerant

Number of chlorine atoms, $p = 2$

Number of fluorine atoms $q = 2$

And number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m + 2$

$0 + 2 + 2 = 2m + 2$ or $m = 1$

i.e Number of carbon atoms = 1

Thus the chemical formula for discolor- difluoro-methane becomes CCl_2F_2 and the number of refrigerant becomes R (1-1) (0 +1)(2) or R-012 i.e R-12.

2. ***Dichloro-tetrafluoro-ethane***

We see that in this refrigerant

Number of chlorine atoms, $p = 2$

Number of fluorine atoms, $q = 4$

And number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m + 2$

$0 + 2 + 4 = 2m + 2$ or $m=2$

i.e Number of carbon atoms = 2

Thus the chemical formula for Dichloro-tetrafluoro-ethane becomes $C_2Cl_2F_4$ and the numbers of Refrigerant becomes R(2-1) (0+1) (4) or R-114.

3. ***Dichloro-trifluore-ethane***

We see that in this refrigerant

Number of chlorine atoms, $p = 2$

Number of fluorine atoms, $q = 3$

And number of hydrogen atoms, $n = 1$

We know that $n + p + q = 2m + 2$

$1 + 2 + 3 = 2m + 2$ or $m = 2$

i.e Number of carbon atoms =2

Thus the chemical formula for Dichloro – trifluore-ethane become $CHCL_2CF_3$ and the number of refrigerant becomes R(2-1) (1+1) (3) or R-123.

The inorganic refrigerants are designated by adding 700 to the molecular mss of the compound. For example, the molecular mass of ammonia is 17, therefore it is designated by R-(700+17) or R-717.

SUBSTITUTES FOR CHLORO-FLUORO-CARBON (CFC) REFRIGERANTS

The most commonly used halo-carbon or organic refrigerants are the Chloro-fluoro derivatives of methane (CH4) and ethane (C_2H_6). The fully halogenated refrigerants with chlorine (CI) atom in their molecules are referred to as Chloro-fluoro-carbon (CFC) refrigerants. The refrigerants such as R-11, R-12,R-13,R-113,R-114 and R-115 are CFC refrigerants.

The refrigerants which contain hydrogen (H) and ethane (C_2 H_6). The fully halogenated refrigerants with chlorine (C1) atom in their molecules are referred to as Chloro-carbon (CFC) refrigerants. The refrigerants such as R-11, R-12, R-13, R-13, R-14 and are CFC refrigerants.

The refrigerants which contain hydrogen (H) atoms in their molecule along with chlorine (C1) and fluorine (F) atoms are referred as hydro-Chloro-flour-carbon (HCFC) refrigerants.

The refrigerants which contain no chlorine and fluorine atoms in their molecule are referred to as hydrocarbon (HC) refrigerants. The refrigerants such as R-290, R-600a are HC refrigerants.

It may be notes that the fluorine (F) atom in the molecule of the refrigerants make s them physiologically more favorable. The chlorine (CI) atom in the molecule of the refrigerants is considered to be responsible for the depletion of ozone layer in the upper atmosphere which allow harmful ultra-violet rays from the sun to penetrate through the atmosphere and reach the earth's surface coursing skin cancer. The Chloro-fluoro-carbon (FCF) refrigerants have been linked to the defection of the ozone layer. They have varying degree of *ozone depletion potential (ODP).

In addition to the ozone depletion effect on the environment, the halo-carbon refrigerants have a global warming effect, which may cause serious changes in the environment.

According to an international agreement (Montreal protocol, 1987), the use of halogenated CFC refrigerants that are considered to have high OFP (such as commonly used refrigerants R-11, R-12, R-113, R-114 andR-02) have been phased out. The refrigerant R-22 which is a hydro-chore-fluorne carbon (HCFC) refrigerant is not covered under the original Montreal Protocol as it ODP is one-twentieth of R-11 and R-12. But because of its GWP, it has to be phased out. Nevertheless, R-22 is found to be of greater use these days as it its being employed not only on its existing R-22 applications but also as substitue for R-11 in very large capacity air-conditioning.

The hydrocarbon (HC) and hydro-fluoro carbon (HFC) refrigerants or vide and alternative for fully halogenated CFC refrigerant. Since they contain no chlorine atom at all, therefore they have zero ODP. Even hydro-Chloro-fluoro carbon (HCFC) refrigerants which cantina some chlorines (CI) atoms, but in association with hydrogen) atoms, have much reduced ODP. However, the hydro-fluoro carbon(HFCs), because of their hydrogen (H) content, may be slightly flammable. The degree of flammability depends upon the number of H-atoms in the molecule. The pure hydrocarbons (HCs) are of course, high flammable:

At present, the following substitutes are available:

1. The HCFC refrigerant R-123 (CF_3 CF_3CHCl_2) in place of R-11 (CCl_3F).
2. The HFC refrigerant R-134a (FG_3CH_2F) and R-152a (CH_3CHF_2) in place of R-12.
3. The HFC refrigerant R-143a (CH_3CH_3) and R-125 (CHF_2CF_3) in place of R-502 (a mixture of R-22 and R-115).
4. The HC refrigerants, propane i.e R-290 (C_3H_8) and is isobutene i.e R-600a (C_4H_{10}) may also be used in place of R-12.

COMPARISON OF REFRIGERANTS

There is no such refrigerant (i.e. ideal refrigerants.) which can be used all operating conditions. The characteristic of some refrigerants. make them suitable for use with reciprocating compressors and other refrigerants. best suited to centrifugal or rotary compressors. Therefore in order to select correct refrigerants. it is necessary that it should satiety those properties which make it ideal to be used for the application. We shall now discuss the thermodynamic, chemical and physical properties of some important refrigerants.

THERMODYNAMIC PROPERTIES OF REFRIGERANTS

The thermodynamic properties of refrigerants are discussed, in detail, as follows:

1. Boiling temperature. The boiling temperature of the refrigerants. at atmospheric pressure should be low. It the bolting temperature of the refrigerants. is high at atmospheric pressure, the cool presser should be opened at high vacuum. The high boiling temperature refuses the capacity and operating cost of the system. The following table shows the boiling temperatures at atmospherics pressure of some commonly used refrigerants.

Table 4 Boiling temperatures

Refrigerant	Boiling temperature (°C) at atmospheric pressure
R-11	+ 23.77
R-12	- 29
R-21	+ 9
R-22	-14
R-30	+ 39.8
R-40	-23.7

R-113	+ 47.6
R-114	+3.6
R-123	+ 27.85
R-134a	-26.15+
R-717	-33.3
R-744	-73.6
R-764	-10

2. Freezing temperature. The freezing temperature of a refrigerant should be well below the operating exploratory temperature. Since the freezing temperature of most of the refrigerants are below- 35°C, therefore this property is taken into consideration only in low temperature operation. The following table shows the freezing temperatures of some common refrigerants.

Table 5 Freezing temperature

Refrigerant	Freezing temperature (°C)
R-11	-111
R-12	-157.5
R-21	-135
R-22	-160
R-30	-96.6
R-40	-97.5
R-113	-35
R-114	-94
R134a	-101
R-717	-77.8
R-744	-56.7
R-764	-75.6

3. Evaporator and condenser pressure.
   Both the evaporation (low side) and condensing (high side) pressure should be positive (i.e above atmospheric) and it should be as bear to the atmospheric pressure as possible. The positive pressures are necessary in order to prevent leakage of air and moisture into the refrigerating system. It also permits easier dictation of leaks. Too high evaporating and condensing pressures (above atsmpsheric) would require stronger refrigerating equipment (i.e. compressors, evaportoator and condenser) resulting higher initial coast. The following table shows the evaporating and condensing pressure, and compression ratio for various refrigerant when operating on the standard cycle of -15°C evaporator temperature and +30°C condenser temperature.

4. **Critical temperature and pressure.** The critical temperature of a refrigerant the highest temperature at which it can be condensed to a liquid, regardless of a higher pressure. It should be above the highest condensing temperature that might be encountered. It the critical temlperarue of results. The following table shows the critical temperature, the excessive power consumption refrigerants. The critical temperature for most of the common reigigerants is well above the normal condensing temperature with the exception of carbon dioxide (R-744) whose critical temperature is 31°C.
5. **Coefficient of performance and power requirements.**
   For an ideal refrigerant operating between -15°C evaporator and 30°C condenser temperature, the theoretical coefficient of performance for the reversed Carnot cycle is 5.74. the following table shows thecaules of theoretical coefficient ofperformance and power per tonne of refrigeration for some common refrigerants operating between -15°CC evaporator temperature and 30°C condenser temperature.
6. **Latent of heat of vaporization.**
   A refrigerant should have a high latent heat of vapors ton at the evaporator temperature. The high latent heat results high refrigerating effect per kg of regirgernat circulated with reduces the mass of refrigerant to be circulated pep tonne of refrigerating. Table 8.10 shows the refrigerating effect for the common refrigerants operating between -15°C evaporator temperarue and 30°C condenser temperature. It also shows the lateen theta, mass of refrigerant circulated per tonne of refrigeration and the volume of the liquid refrigerant per tonne of reiteration.
7. **Specific volume.**
   The specific volume of the refrigerant vapour at evaporator temperature (i.e volume of suction vapour to the compressor) indicated the theoretical displacement of the compressor. The reciprocating compressor are used with referents having high pressures and low volumes of the suction vapour. The centrifugal or turbo compressors are used with refrigerants having low pressure and high volumes of the suction vapour. The rotary compressor are used with refrigerants having intermediate pressures and volumes of the suction vapour. Table 8.11 shows the specific volume of the refrigerant vapor and the optical piston displacements for various refrigerants.

CHEMICAL PROPERTIES OF REFRIGERANTS

1. **Flammability.** We have already discussed that hydro-carbon refrigerant such as ethane, propane etc, are highly flammable. Ammonia is also somewhat flammable and becomes explosive when mixed with air the radio of 16 to25 per cent of gas by volume. He halo-carbon refrigerants are neither flammable nor explosive.
2. **Toxicity.** The toxicity of referent may be prime or secondary importance, depending upon the application. Some non-toxic refrigerants. (i.e all fluorocarbon refrigerants) when mixed with certain portage of air certain percentage of air become toxic

3. **Solubility of water.** Water is only slightly soluble in R-12. At -18°C, it will hold six parts per million by weight. The solution formed is very slightly corrosive to any of the common medial. The solubility of water with R-22 is by a ratio of 3 to 1. It more water is preset than can be dissolved by the refrigerant, the ice will be formed which chokes the expansion valve or capillary tube used for throttling the system. This may be avoided by the proper dehydration of the refrigerating unit before charging and by the use of sepia gel drier of the liquid line. Ammonia is highly soluble in water. Due to this reason, a wetted cloth is put at the point of leak to avoid harm to the persons working in ammonia refrigerating plants.
4. **Miscibility.** The ability of a refrigerant to mix with oil is called miscibility. This property of refrigerant is considered be to secondary factor in the selection of are grievant. The degree of miscibility depends upon the temperature of the oil and presser of the refrigerating vapour. The dioxide and methyl chloride are relatively non-miscible.
4. **Effect on perishable materials.** The refrigerants used in cold stronger plant and in domestic refrigerators should be such that in case of leaker, it should have no effect on the perishable materials. The Freon group of refrigerants has no effect upon dairy products, meats vegetables, flowers and ford. There will be no changeling colour, tests or texture of the material when exposed to Freon. Methyl chloride vapors have no effect upon furs, flowers, eating, foods or drinking beverages. Sulphur dioxide destroys flowers, plant and furs, but it does not affect foods. Ammonia dissolves easily in water and becomes alkaline in nature. Since most fruits and vegetables are acidic in nature, therefore ammonia reacts with these products and spills the taste.

PHYSICAL PROPERTIES OF REFRIGERANTS

1. **Stability and Inertness.** An ideal refrigerant should not decompose at any temperature normally encountered in there refrigerating system. It should not form higher boiling point liquids or solid substance through polymerization. Some refrigerants disinter grate forming non-condensable gases which causes high condensing pressure and vapour lock. The disintegration of refrigerant may be due to reaction with melts. In order to avoid this, a refrigerant should be inert with respect all materials used in refrigents system. The freon group of refrigerants is stable up to a temperature of 535°C. Above this temperature, it decomposes and forms corrosive and poisonous products. The freon refrigerants not used with rubber gaskets as if acts as a solvent with rubber, since sulphur dioxide does not decompose below 1645°C, therefore are One of the most stable refrigerants.
2. **Corrosive property.**

The corrosive property of a refrigerant must be taken into consideration while selecting the refrigerants. The freon group of refrigerants are non- corrosive with practically all metals. Ammonia is used only with iron or steel. Sulphur dioxides is non-corrosive to al metals in the absence of water because suphur dioxide reacts with water and forms sulphuric acid.

3. Viscosity. The refrigerant in the liquid and vapour states should have low viscosity. The low viscosity of the refrigerant is desirable because the pressure drops in passing through liquid and such tin lines are small. The heat transfer through condenser and evaporator is improved at low viscosities. The following table shows the victories (in centipoises) at atsmpsheric pressure for the common refrigerants.

4. **Thermal conductivity.**The refrigerant in the liquid and vapour states should have high thermal conductivity. This property is required in finding the health transfer coefficients in evaporators and condensers. Table 8.14 shows the thermal conductivities of common refrigerants.
5. **Dielectric strength**. The dialectic strength of a refrigerant is important in hermetically sealed units in which the eclectic motor is exposed the refrigerant. The relative dielectric strength of the refrigerant is the ratio of the dielectric strength of nitrogen and the refrigerant vapour mixture shows the alive dielectric strengths of common refrigerants.

6. Leakage tendency. The leakage tendency of a refrigerant should be low. It there is leakage of refrigerants, it should be easily detectable. The leakage occurs due to opening in the joints of flaws in material, used for constructing. Since the fluorocarbon refrigerants are collarless, therefore, their leakage will increase the operating cost. The ammonia leakage is easily detected due to its pungent odour. The leakage of fluorocarbon refrigerants may be detected by soap solution, a halide torch or and electronic leak detector. The latter is generally used in big refrigerating plants. The ammonia leakage is detected by using burring sulphur candle which in the presence of ammonia forms while fumes of ammonium sulphite.
7. **Cost.**The cost of refrigerant is not so important in small refrigerating units but is very important in high capacity refrigerating systems like industrial and commercial. The ammonia, being the cheapest, is widely used in large industrial plants such as cold storage and ice plants. The refrigerant R-22 is costlier then refrigerant R-12. The cost of losses due to leakage is also important.

SECONDARY REFRIGERANTS – BRINES

Brines are secondary referents and are generally used where temperatures are required to be maintained below the freezing point of water i.e 0°C. In case the temperature involved is above the freezing point of water (0°C), then water is commonly used as a secondary refrigerant.

Brine is a solution of salt in water. It may be noted that when salt is mixed in water, than the freezing temperature of the solution becomes lower than the of the water. This is due to the fact that the salt while dissolving in water takes of it its latent health form the solution and cods it bellwether freezing point of water. The mass of the salt in resolution expressed as the percentage of the mass of the solution is known as concentration of the solution. As he concentration of the solution increases, it’s the salt is increased beyond a certain point, the freezing point decreases. But it the concentration of the salt is increased beyond a certain point, the freezing point increases instead of decreasing. A which the freezing point decreases. But if the concentration of the salt is increase beyond a certain point, the freezing point increases of decreasing. The point at which the freezing temperature is minimum, is known aesthetic temperature and the conception at this points known as eutectic temperature and the concentration at this point is known as eutectic concentration. Brine used in a particular application should have a concentration fort which the freezing point of the brined is at last 5°C to 8°C lower than the than brine temperature required.

The brine commonly used are calcium chloride ($CaCl_2$), sodium chloride i.e common salt (NaCl) and glycols such as ethylene glycol, propylene glycol etc.

The calcium chloride brine has the eutectic temperature of -55°C at salt concentration of 30% by mass. This brine is primarily used where temperatures below- 18°C are required. It is generally used in industrial process cooling and product freezing. The chief disadvantages of calcium chloride brine are its dehydrating effect and its tendency to impart a bitter taste to food products.

The sodium chloride brine has the eutectics temperature of -21.1°C at salt concentration of23% by mass. This brine is used in chilling and freeing of meat and fish.

Both of the above two brines are corrosive in nature for metallic containers which put limitation on their use. Also the thermal properties of the above tow brines are less satisfactory.

Other water soluble compounds known as antifreeze are also used for decreasing the freezing point of water for certain refrigeration used. Ethylene and propylene’s glycol have a number of good properties. Since they are non-corrosive and non-elecolytic even in the in the presence of water, therefore, these brines are most extensively used as antifreeze elements. The following table shows typical applications of various brines.