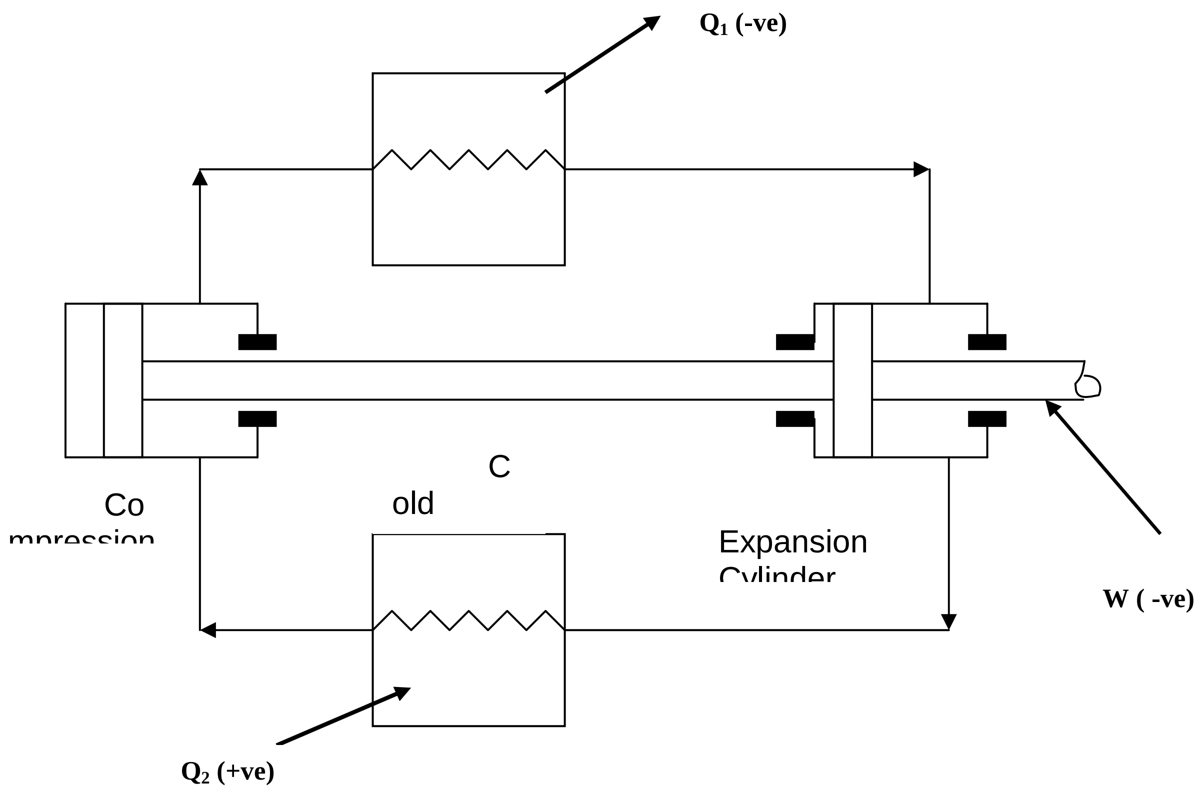
REFRIGERATION

The Reversed Carnot Cycle

A heat engine is a thermodynamic system in which a fluid undergoes a cycle of processes, which enables heat and work transfers to be effected. By definition, a refrigerator is a heat engine operating in reverse, whose function is to cause a continuous transfer of heat from a cold space with a net expenditure of work.

When the same cycle has as its prime purpose the supply of heat at a higher temperature it is referred to as a HEAT PUMP CYCLE.

Condensible fluids are used in order that the temperatures of heat reception and heat rejection are kept almost constant and to reduce the physical size of the heat exchanger plant.



REFRIGERATOR OPERATING ON REVERSED CARNOT CYCLE

The ideal cycle for a heat engine is a **Carnot cycle** and the ideal cycle for a refrigeration system is a **Reversed Carnot cycle** i.e. work is put into the cycle and the cycle performs in the reverse way.

In the above figure, heat Q_2 is extracted from the cold chamber and heat Q_1 is rejected into the surroundings, W is the net work needed.

The effectiveness of a refrigerator is termed the COEFFICIENT OF PERFORMANCE (C.O.P.)

The Coefficient of Performance is defined as the ratio of the heat extracted in the cold chamber to the work done within the system.

$$COP = \frac{HeatExtracted(Q_2)}{WorkDone(W)}$$

For a reversed Carnot cycle it can be shown that:

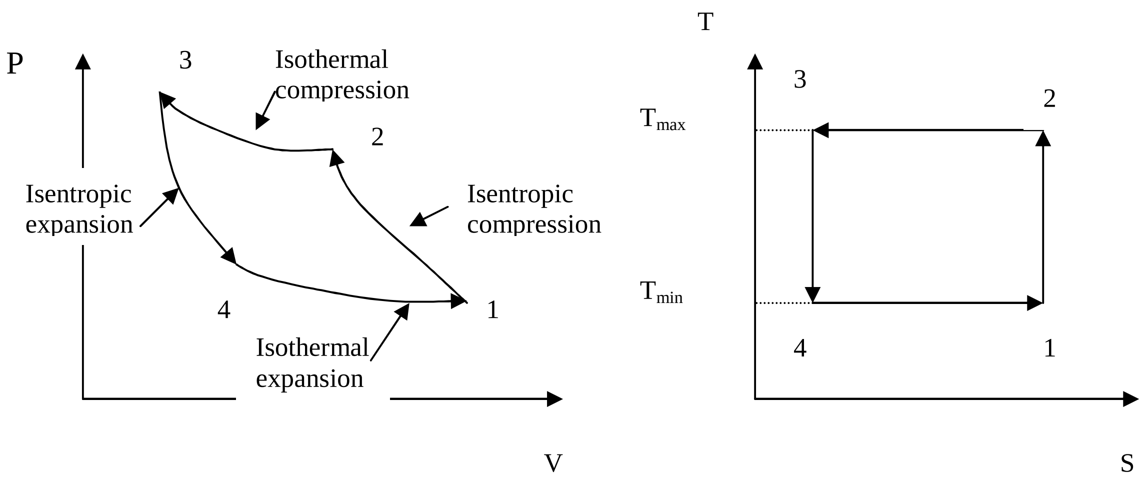
$$C.O.P._{\substack{carnot\\refrig}} = \frac{T_{\min}}{T_{\max} - T_{\min}}$$

The heat pump cycle is similar but in this case the main consideration is not the heat extracted from the cold chamber but the heat which is rejected from the condenser which can be used to heat buildings and homes. The effectiveness of the heat pump is thus defined as follows;

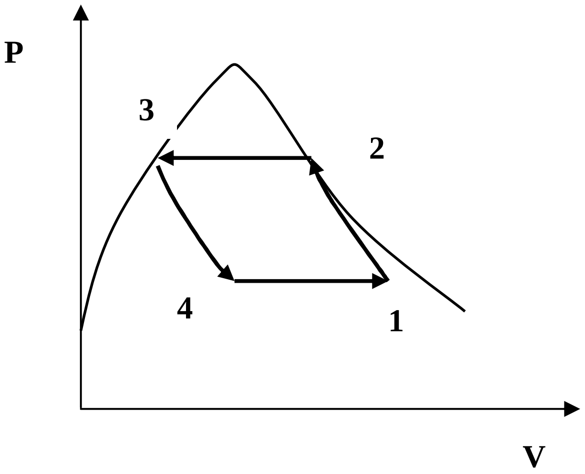
$$COP = \frac{HeatExtracted(Q_1)}{WorkDone(W)}$$

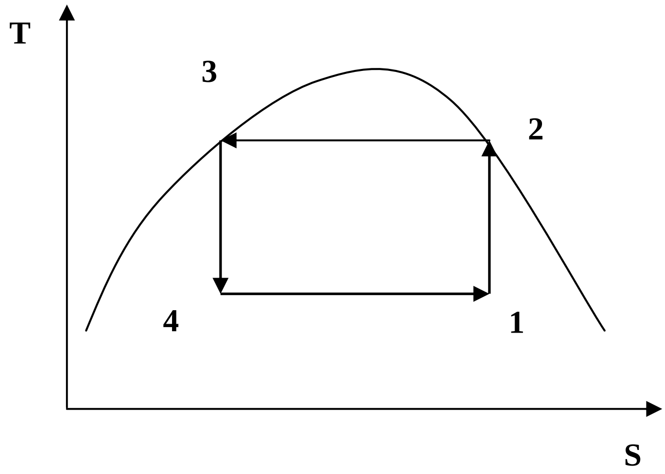
and for the reversed Carnot cycle it can be shown that;

$$C.O.P._{\substack{carnot\\heatpump}} = \frac{T_{\max}}{T_{\max} - T_{\min}}$$



REVERSED CARNOT CYCLE FOR A GAS





REVERSED CARNOT CYCLE FOR A VAPOUR

The Carnot Cycle for a vapour has the same form as for a gas on the TS diagram but alters in shape when plotted on the P-V diagram. The isothermal processes (2-3) and (4-1) are in the wet vapour region and as such occur at constant pressure.

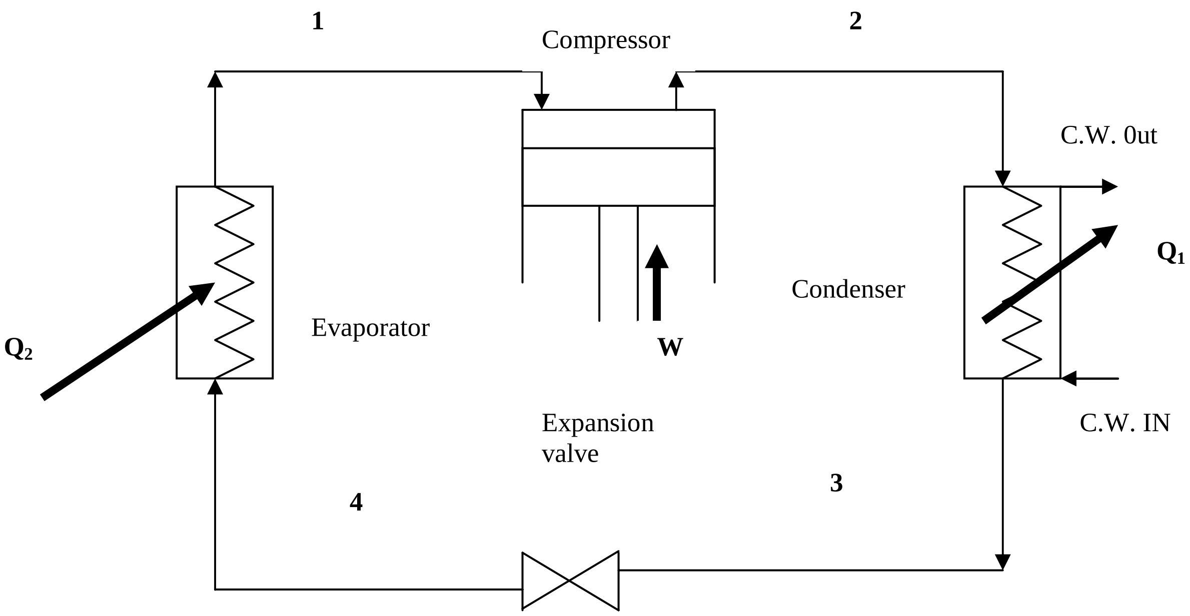
The reversed Carnot cycle implies reversible processes but all actual processes are irreversible. The reversed Carnot cycle is **not** a practical cycle but is merely used as a standard against which all other refrigeration system cycles can be compared.

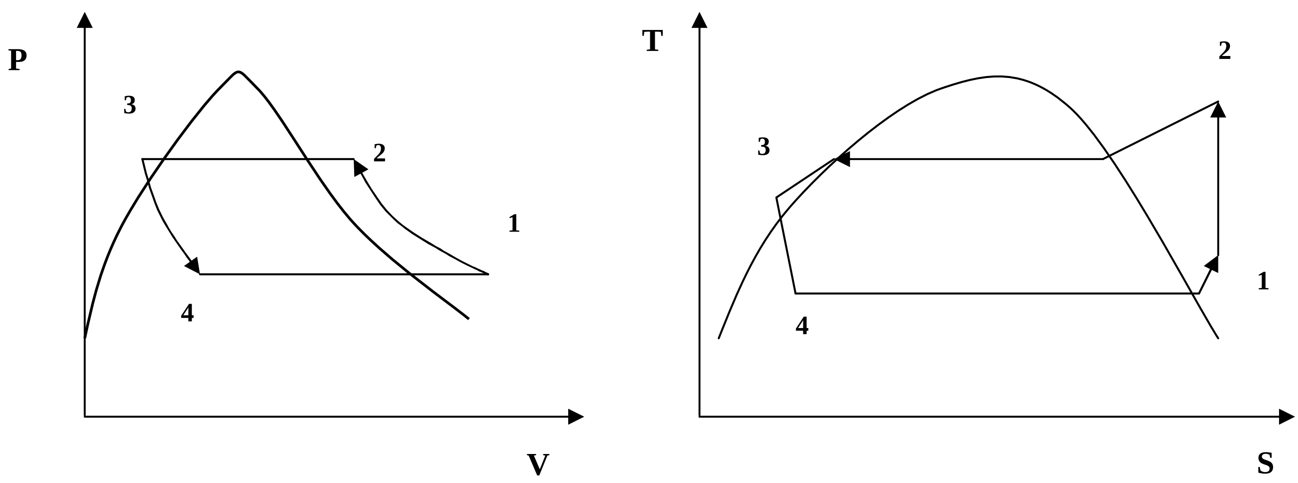
In the practical cycle:

1) the low expansion cylinder is replaced by an expansion valve
2) it is easier to compress a saturated or slightly superheated vapour than to compress a wet vapour up to an exact saturation point.
3) It is usual to achieve the process (2-3) in a condenser and it is extremely difficult to remove only sufficient heat to allow the resulting liquid to be at a saturated liquid point. A certain amount of under-cooling in the condenser will always occur.

The above modifications to the reversed Carnot cycle give a system, which is much more practical. This system is known as the **vapour compression cycle.**

Vapour Compression Cycle





(1-2) Compression of the refrigerant vapour in the compressor. The ideal compression would be isentropic but normally an isentropic efficiency is employed with system problem solving. Small systems usually employ reciprocating compressors and larger systems such as that utilised in cruise ship air conditioning systems will employ axial flow, centrifugal or screw types of compressor.

(2-3) Liquefaction of the refrigerant vapour by cooling in the condenser. This is normally assumed to take place at constant pressure. This is the HEAT REJECTION STAGE.

(3-4) Expansion or throttling of the refrigerant liquid. This is assumed to occur at constant enthalpy. The result is either a liquid or a very wet vapour at a low temperature and pressure.

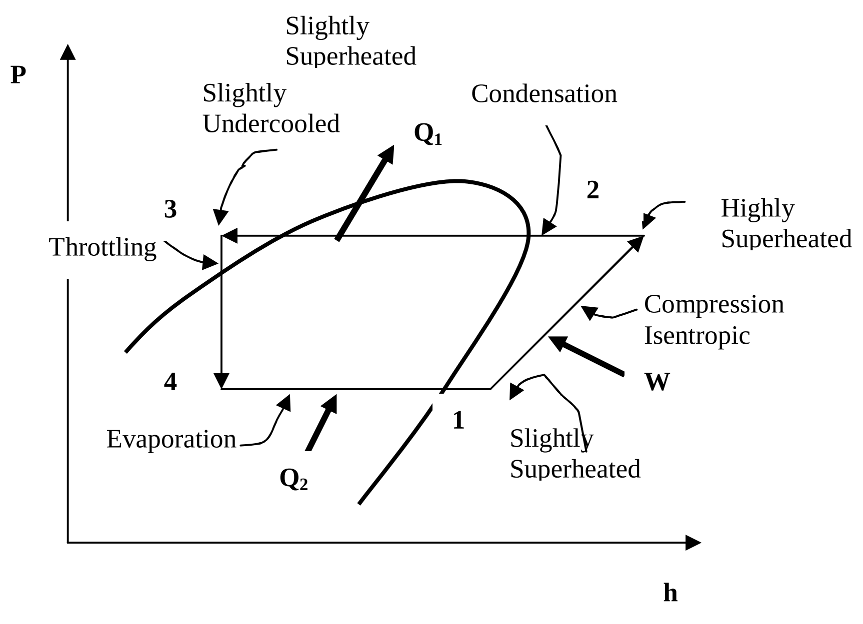
(4-1) Evaporation of the cold wet refrigerant in the evaporator which ideally will occur at constant pressure. For smaller installations the evaporator coils will be installed in the **cold room** thus forming a **Direct System.** For larger systems a secondary refrigerant is employed where the secondary refrigerant such as brine or glycol mixtures is used as the heat carrier. This forms an indirect system. This is the HEAT EXTRACTION SYSTEM

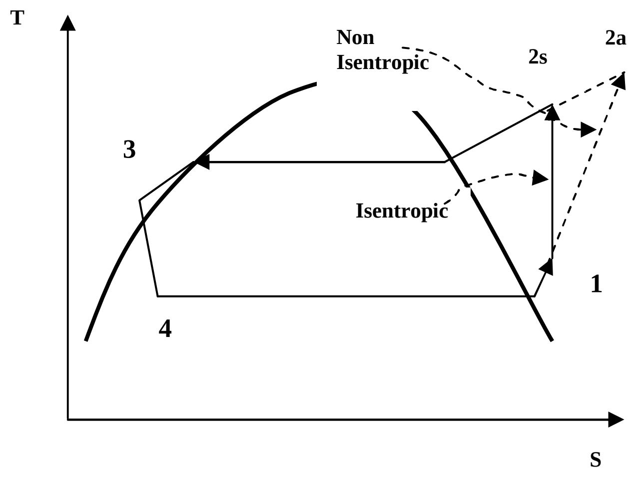
Expansion Valve

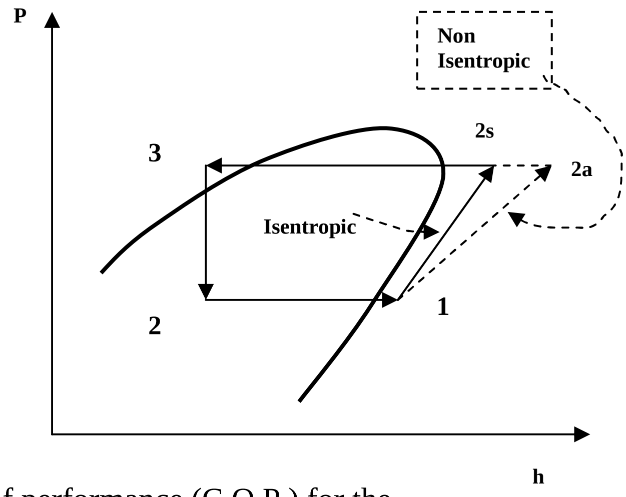
This has the function of controlling the flow rate of the refrigerant in line with the demand from the cold room. The amount of refrigerant admitted to the evaporator is critical. It must be just enough so that the heat absorbed in the evaporator can vaporise it. The opening of the expansion or throttle valve is generally controlled by the outlet superheat temperature of the refrigerant vapour at exit from the evaporator. If it does this correctly then it ensures the optimum effectiveness of the plant.

Pressure Enthalpy Diagram.

The ph diagram can be most helpful in understanding the vapour compression cycle Consider the following:







We have previously defined the CARNOT coefficient of performance (C.O.P.) for the refrigeration and heat pump system. With actual cycles it is usually possible to determine the enthalpies at the cardinal points of the cycle and thus determine the actual C.O.P. required. This Actual C.O.P. can be defined in terms of those enthalpies.

For the refrigeration system:

$$C.O.P_{\mathrm{Re}\,frig} = \frac{Heat.extracted.by.evaporator.coils}{work.done.by.compressor}$$

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

The value (h_1-h_4)/kg is sometimes known as the **refrigerating effect**

The product of this refrigerating effect and the mass flow rate of refrigerant ($\dot{m}_{ref}$) is known as the **refrigerating load (kW)**

$$\text{Refigerating load} = \dot{\mathrm{m}}_{\boldsymbol{ref}} \times (\boldsymbol{h}_1 - \boldsymbol{h}_4)$$

For the heat pump:

$$C.O.P_{\substack{Heat\\pump}} = \frac{Heat.rejected.at.the.condenser}{workdonebythecompressor}$$

$$= \frac{h_2 - h_3}{h_2 - h_1}$$

As can be seen from the Ph diagram

The WORK DONE = HEAT REJECTED – HEAT EXTRACTED

= $H_R - H_E$

$$\therefore C.O.P._{refrig} = \frac{H_E}{H_R - H_E}$$

If we add one (1) to both sides of this equation we then have

$$(C.O.P._{refrig} + 1). = (\frac{H_E}{H_R - H_E}) + 1 \text{ or}$$

$$\frac{H_E}{H_R - H_E} + \frac{H_R - H_E}{H_R - H_E}$$

$$= \frac{H_R}{H_R - H_E} = C.O.P._{\substack{HEAT \\ PUMP}}$$

$$\therefore C.O.P._{\substack{HEAT \\ PUMP}} = C.O.P._{REFRIG} + 1$$

For most problem solving calculations, unless otherwise stated, the following assumptions can be made:

1. Condensation occurs at constant pressure i.e. $P_3=P_2$
2. Expansion is at constant enthalpy i.e.$H_4=H_3$
3. Evaporation is at constant pressure i.e. $P_1=P_4$
4. In the absence of other information it may be assumed that compression is isentropic. i.e. $S_2=S_1$. For chief engineer problems it would be usual to be given an isentropic efficiency for the compressor.

When solving questions it generally good policy to represent the problem on a ph diagram. It is almost always necessary determine enthalpy values from the properties tables.(Steam Tables) Often this may involve interpolation when the required enthalpy or other value lies between those stated in the tables.

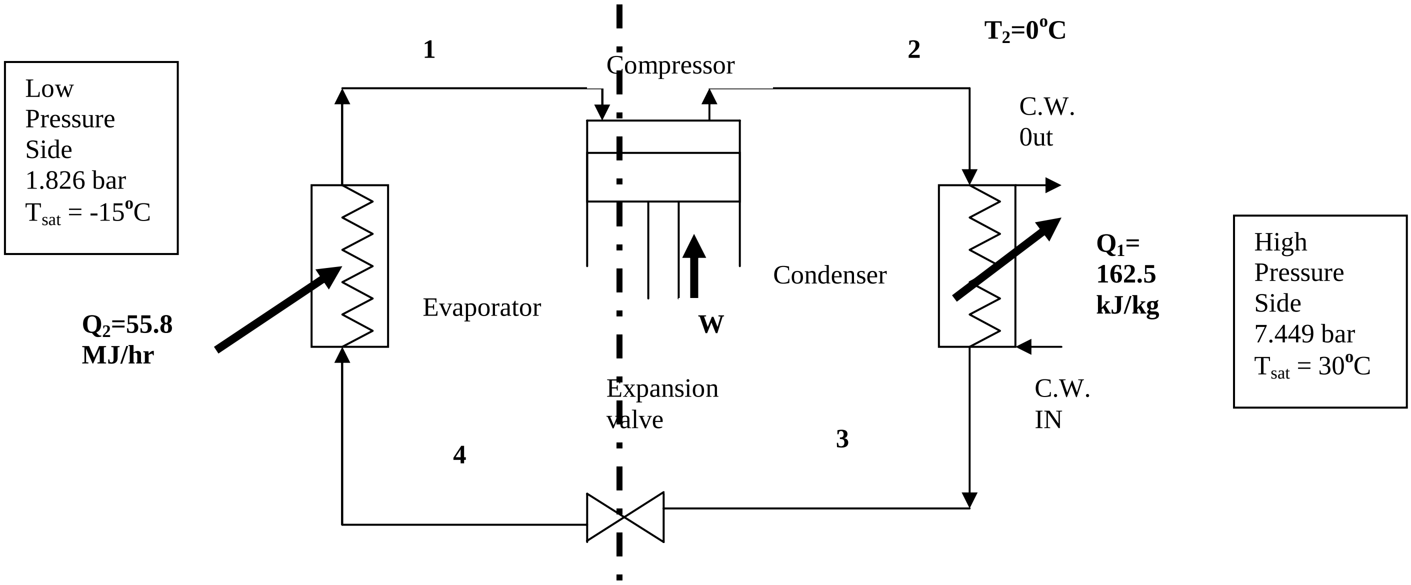
Consider the following worked example:

A vapour compression refrigerating unit utilises R12 which operates between saturation temperatures of –15^{o}C and 30^{o}C. The refrigerant leaves the evaporator at a temperature of 0^{o}C and leaves the compressor at a temperature of 60^{o}C after being isentropically compressed. The cooling water removes 163.2 kJ/kg of refrigerant flow in the condenser and the cooling load is 55.8 MJ/hr.

a) Sketch the ph and TS diagrams
b) Calculate :

i) the condition of the refrigerant entering the evaporator
ii) the degree of under-cooling in the condenser
iii) the coefficient of performance of the plant
iv) the refrigerant mass flow rate

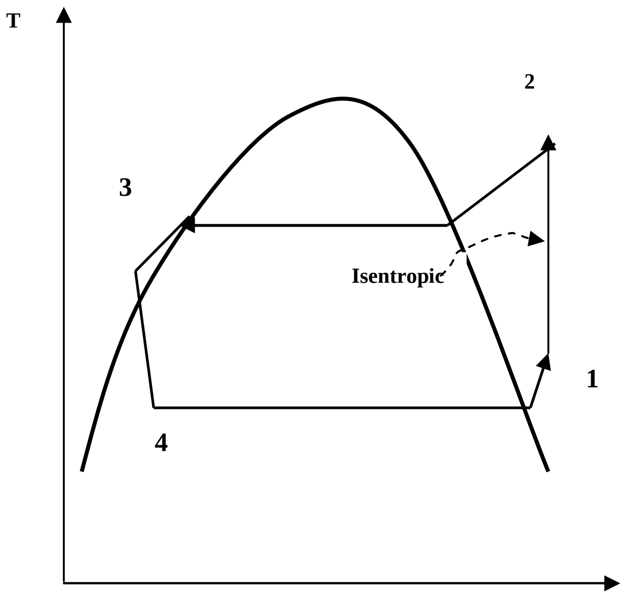
Probably the easiest way to solve this problem is as in many cases to draw a line diagram of the system indicating the known values

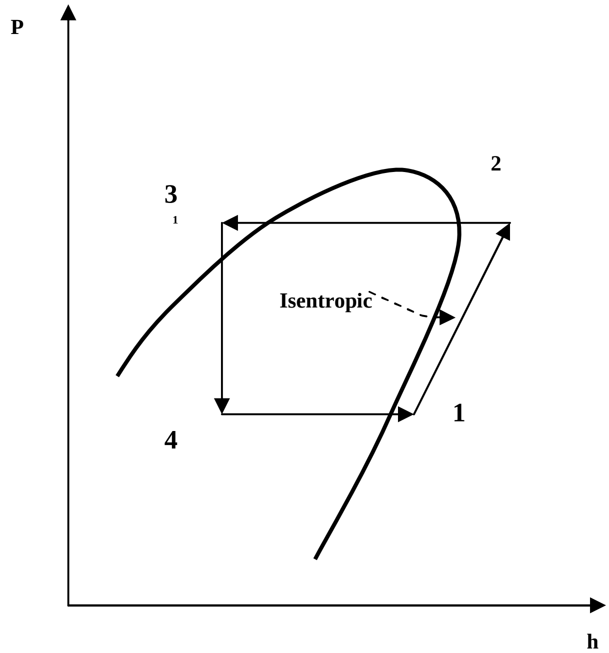


T_1 =0°C $\quad\therefore$ refrigerant has 15 °C of superheat

$\therefore$ h_1 = 190 .15 kJ/kg and S_1 = 0.7397kJ/kgK= S_2 (since the compression is isentropic) h_2 must lie in the superheat region at a pressure of 7.449bar between 15K and 30K hence interpolation is required.

The ph and TS diagram are as shown below





Interpolation for enthalpy values utili the range of entropy values is as shown below;

$$h_2 = h_{15} + \left(\frac{S_2 - S_{15}}{S_{30} - S_{15}}\right) \times (h_{30} - h_{15})$$

the suffixes $_{15}$ and $_{30}$ refer to the property values in these regions of superheat

$$h_2 = \frac{(0.7397 - 0.7208)}{(0.7540 - 0.7208)} \times (221.44 - 210.63)$$

= 216.78 kJ/kg

If 163.2 kJ/kg is removed in the condenser then h_3 = 216.78 – 163.2 =53.58kJ/kg

Saturation temperature at high pressure is 30^0C which corresponds to a h_f value of 64.59 kJ/kg.

Since 53.58<64.55 Then $t_3 < t_{sat}$

The saturation temperature approximately corresponds to the hf of 53.58 kJ/kg This lies between the saturation temperatures of 15^0C and 20^0C and further interpolation is required.

$$t_3 = t_{15} + \frac{h_3 - hf_{15}}{hf_{20} - hf_{15}} \times (t_{20} - t_{15})$$

where suffixes $_{20}$ and $_{15}$ refer to properties at saturation temperatures of 20^0C and 15^0C

$$t_3 = 15 + \frac{(53.58 - 50.10)}{(54.87 - 50.10)} \times (20 - 15)$$

= 18.65^0C

hence degree of undercooling = 30-18.65 = **$\underline{11.35^0C}$**

Coefficient of Performance $COP = \dfrac{refrig.Effect}{Work}$

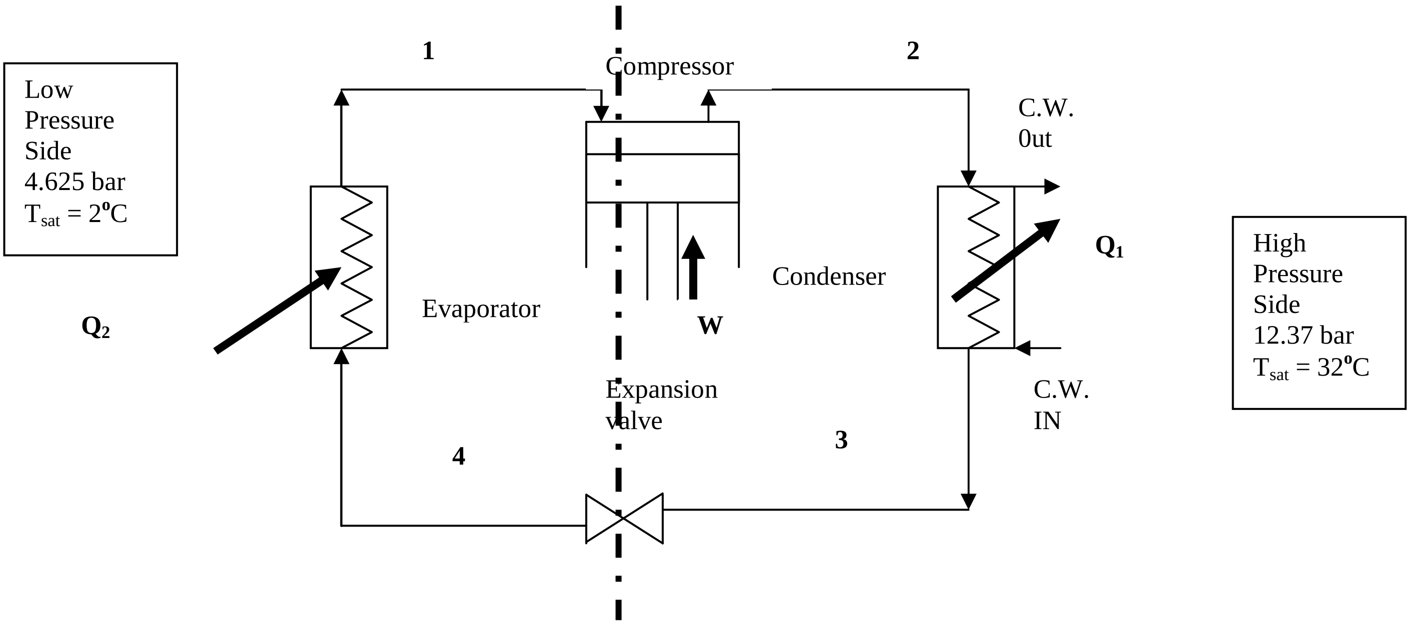
$$= \frac{h_1 - h_4}{h_2 - h_1} = \frac{190.15 - 53.58}{216.63 - 190.15} = \mathbf{5.13}$$

Let's try another example, **this time with an isentropic efficiency at the compression stage.**

In a vapour compression cycle ammonia leaves the evaporator in a dry saturated state at a pressure of 4.625 bar. The vapour is then compressed to a pressure of 12.37 bar with an isentropic efficiency of 0.85. It is cooled in the condenser to a saturated liquid and then passed to the evaporator via the expansion valve:

a) Sketch the cycle on the TS and ph diagrams
b) Calculate
    i) the dryness fraction of the fluid at entry to the evaporator
    ii) the coefficient of performance

Once again make a sketch of the system indicating the salient points and known values.



Draw the TS and ph diagrams ensuring the relevant diagram positions for the fluid states. i.e. saturated liquid at exit from the condenser and saturated vapour at exit from the evaporator