Refrigeration and Air Conditioning

Unit 8

Vapour Compression Refrigeration

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Contents

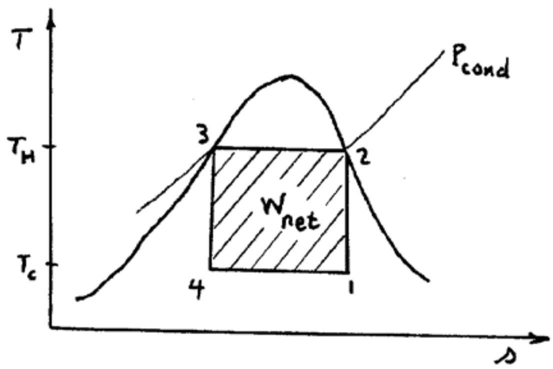
Vapour Compression Refrigeration

The reversed Carnot cycle is the most efficient refrigerator (or heat pump) possible. However, when the refrigerant is a gas the Carnot cycle cannot be implemented in practice and the less efficient Brayton-Joule cycle had to be used. But, when the refrigerant is a vapour, then the reversed Carnot cycle can be made almost completely practical by operating in the liquid-vapour region.

The Ideal Cycle

The *ideal cycle* can be considered to be *one of two forms*.

(a) The most commonly seen form is as follows.



$$COP_c = \frac{q_{ref}}{w_{net}} = \frac{T_C}{T_H - T_C}$$

$$q_{ref} = T_C\left(s_1 - s_4\right)$$

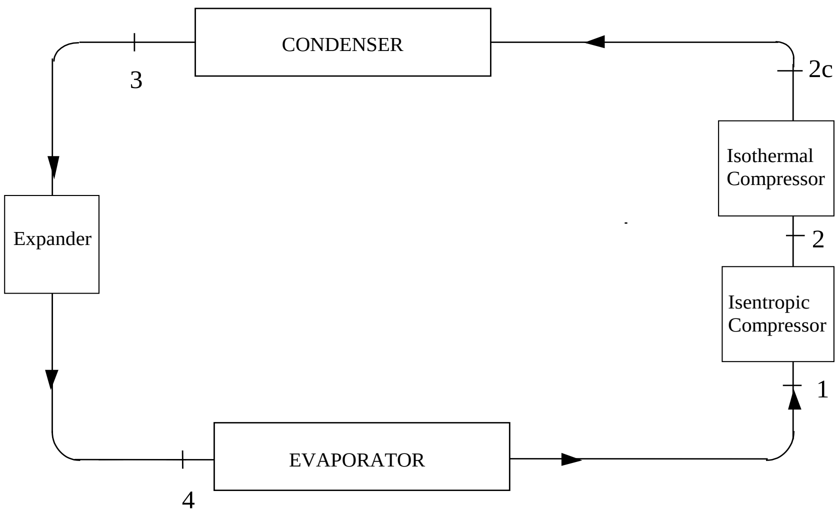
$$\begin{aligned} w_{net} &= q_{cond} - q_{ref} \\ &= \left(T_H - T_C\right)\left(s_1 - s_4\right) \end{aligned}$$

in which the processes are:

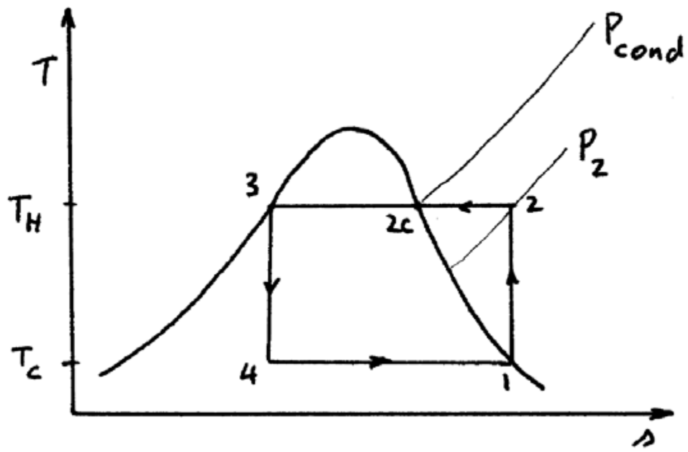
1-2 isentropic compression
2-3 condensation
3-4 isentropic expansion
4-1 evaporation

An alternate, but equally valid form is given in the following.

(b) where we now have two compressors, an isentropic compressor and an isothermal compressor (see the figure below).



The resulting T-s diagram is



As before, we have that

$$COP_c = \frac{q_{ref}}{w_{net}}$$

where $$q_{ref} = q_{41} = h_1 - h_4 \equiv T_c\left(s_1 - s_4\right)$$

and from the First Law of Thermodynamics,

$$w_{net} = q_{23} - q_{41}$$

i.e.
$$\begin{aligned} w_{net} &= T_H\left(s_2 - s_3\right) - T_C\left(s_1 - s_4\right) \\ &= \left(T_H - T_C\right)\left(s_1 - s_4\right) \end{aligned}$$

$\therefore$
$$COP_c = \frac{T_C}{T_H - T_C}$$

We obviously see that both are valid Carnot cycles, with exactly the same Coefficient of Performance, COP, but with different cooling capacities, *q*, and different power inputs.

Case (a) *is more realistic* in that *only one compressor* is used, whereas in case (b), there are two compressors.

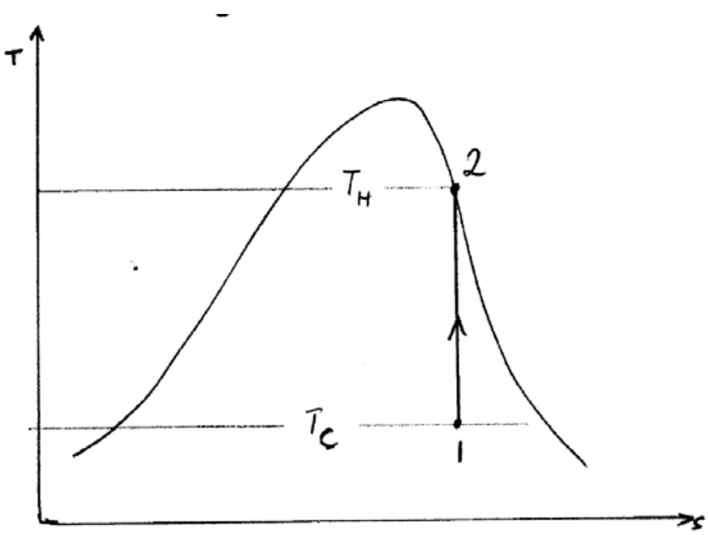
Case (b) *is more realistic* in regard to the *inlet conditions* at suction, i.e. point 1.

Theoretical (or Standard) Vapour Compression Cycle

While the cycle shown before offers a high coefficient of performance, practical considerations require certain revisions.

(i) Work of Compression

In *case (a)* we saw that the *compression* is *"wet"*, viz.



In this type of process "wet vapour" is taken from the evaporator and compressed isentropically until point 2 is obtained. Although this is possible and some *early* compressors did operate on this principle, there are however practical problems.

These problems are:

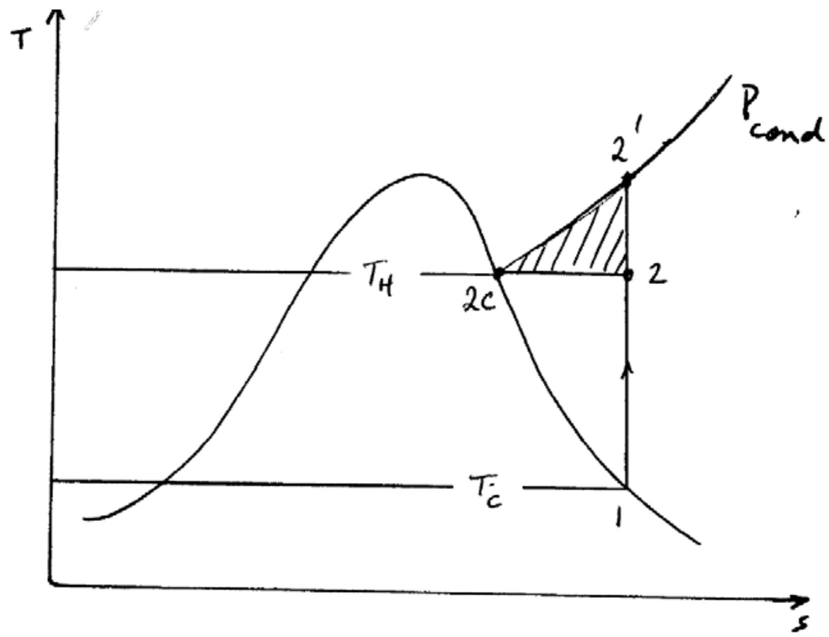
(1) During compression the droplets of liquid are vaporised by the internal heat transfer process which requires a finite amount of time. This time is not normally available and liquid droplets may become trapped in the head of the cylinder by the rising piston, possibly damaging the valves or cylinder head.

(2) The refrigerating effect is reduced and therefore a larger compressor may be required to circulate the refrigerant.

(3) The liquid droplets may wash away the lubricating oil from the cylinder walls, accelerating wear.

It is therefore normally undesirable to have "wet" compression, i.e. as per *case (a).*

Consider then the alternative of *case (b).*



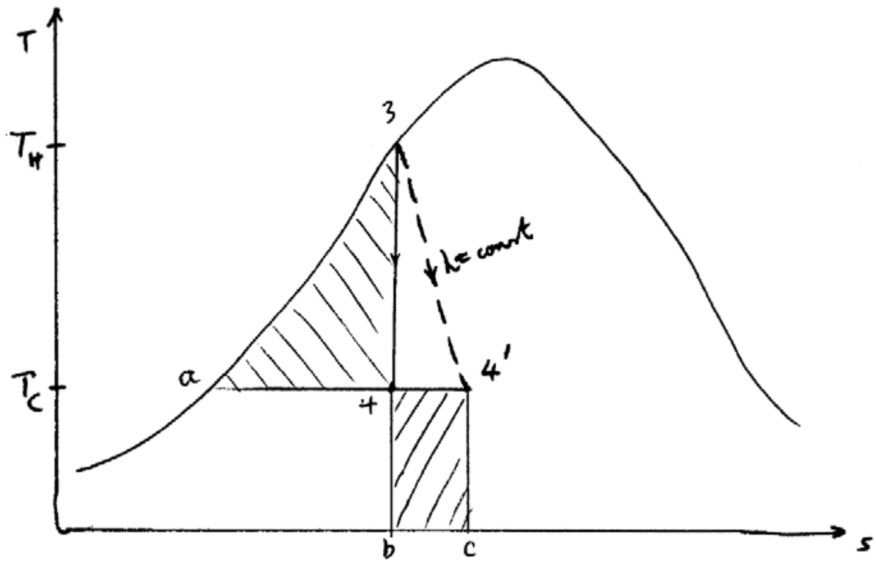
It is generally impractical to stop the isentropic compression in the “first” compressor at point 2 and then continue an isothermal compression in the “second” compressor to 2C. It is normal to continue the isentropic compression to point 2' , i.e. to the condenser pressure.

The shaded area represents the additional work required as a result of this.

(ii) Expansion Process

Despite the theoretical advantages of using reversible expansion for the process 3- 4, it is hardly ever attempted in practice for ordinary vapour compression cycles, because the extra cost of the expansion engine or turbine is not worth "even" the theoretical gain; and when account is taken of inefficiencies of the expander the gain becomes very small.

The most common alternative is to replace the isentropic expansion by a simple throttling process. Let us look at this.



It can be easily seen that the *loss in refrigerating effect* due to the introduction of throttling is

$$q_{loss} = h_{4'} - h_4 \equiv T_c\left(s_{4'} - s_4\right)$$

However, we also lose the work that the turbine (or expander) would have developed, i.e.

$$w_{exp_{loss}} = h_3 - h_4$$

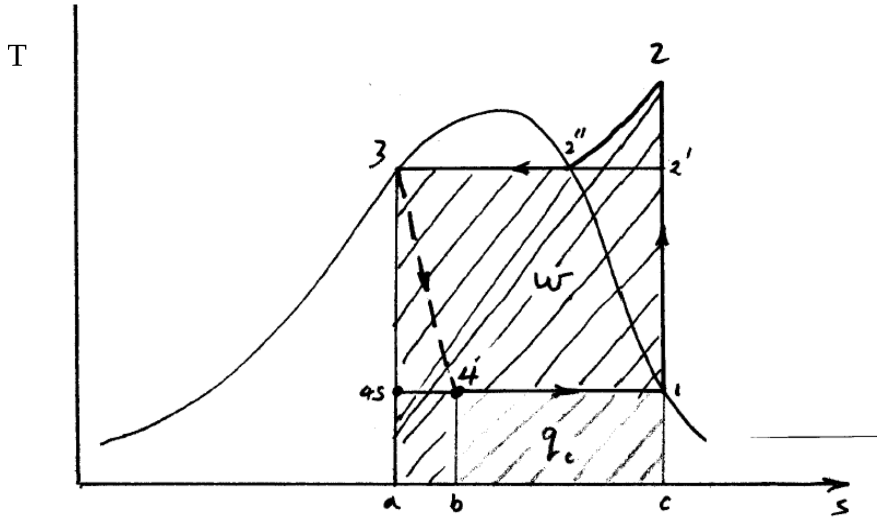
but $h_3 = h_{4'}$

$\therefore$ $$w_{exp_{loss}} = h_{4'} - h_4$$

Hence, the lost work is equal to the lost refrigerating effect.

The Theoretical Cycle

The cycle with the above two modifications is called the *Theoretical Cycle*.

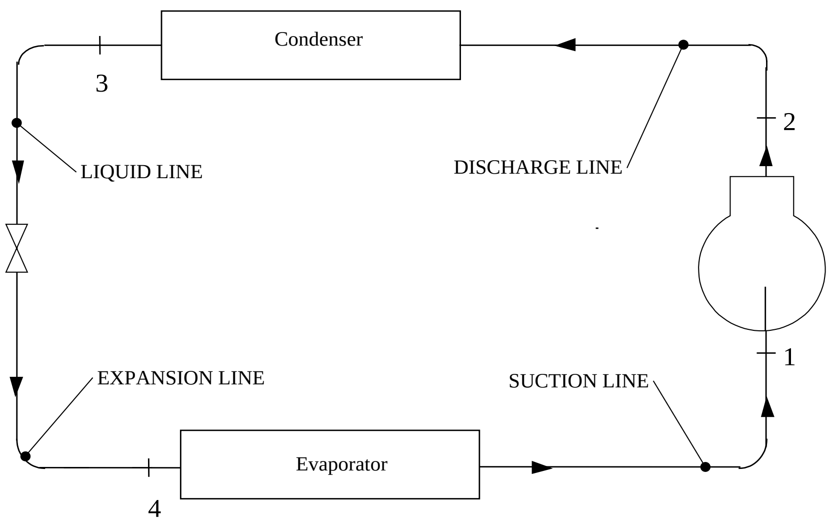


The cycle consists of,

1 - 2	isentropic compression (reversible adiabatic)
2 - 3	condensation at constant pressure
3 - 4	throttling (irreversible) constant enthalpy
4 - 1	evaporation at constant pressure

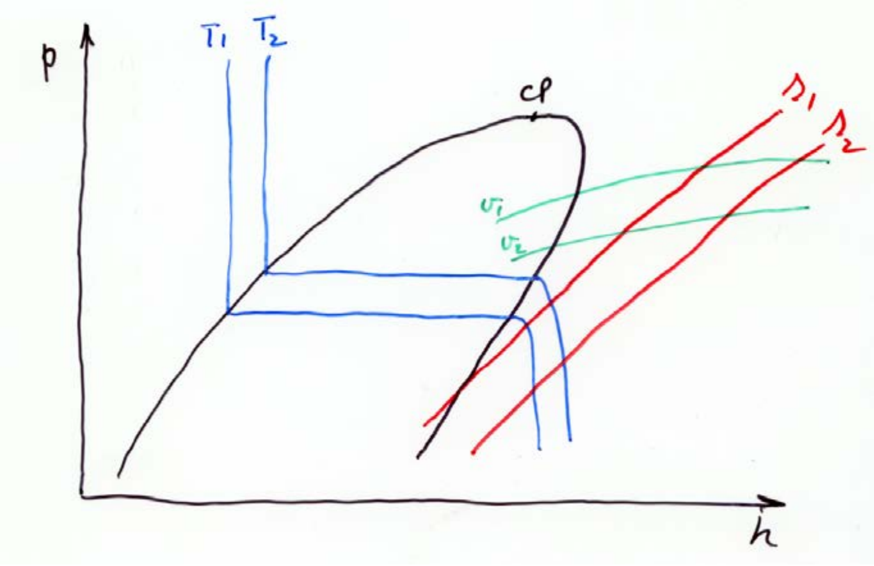
Note that the area 2'22" represents the increase in work required due to the superheat horn, the area 4s4ba represents the loss of refrigerating effect and equally the loss of expander work. All of these facts result is a lower COP for the theoretical cycle in comparison with the ideal cycle.

The plant is shown in the figure below. Common names for the lines interconnecting the equipment have been labelled.



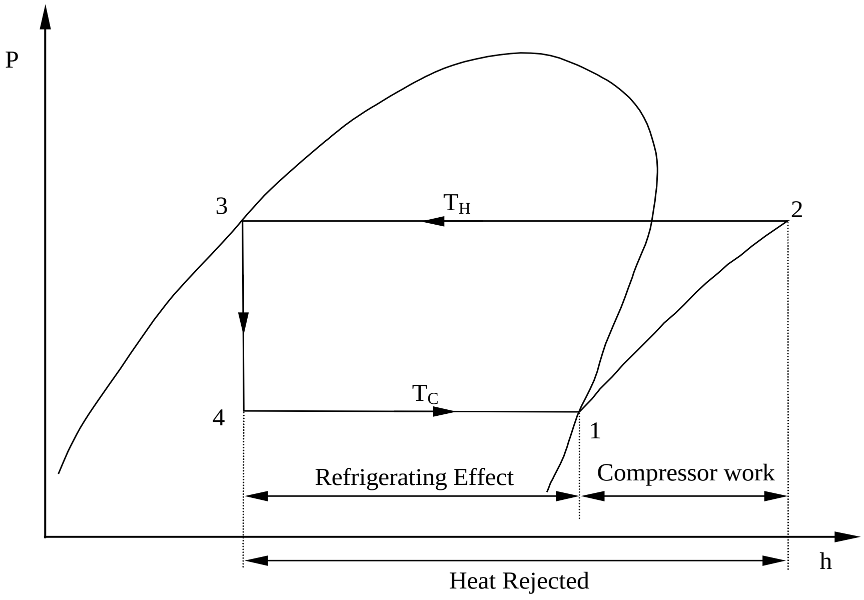
The Pressure-Enthalpy Diagram

Since our refrigeration cycle consists of two constant pressure processes and one constant enthalpy process, it has therefore been found convenient to represent the cycle on a pressure-enthalpy diagram.



The advantage of using the P-h diagram of the vapour-compression cycle analysis can best be illustrated by redrawing the theoretical cycle on the diagram.

Analysis of the Theoretical Cycle



Hence we can write,

$$\left.\begin{array}{lr} \text{Refrigerating effect} & q_{ref} = h_1 - h_4 \\ \text{Heat rejected in condenser} & q_{cond} = h_2 - h_3 \\ \text{Compressor work} & w = h_2 - h_1 \end{array}\right\} \text{per unit mass}$$

For the theoretical cycle

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

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

It is worth noting that the Australian Standard AS/NZ 3823, now uses the terminology Energy Efficiency Ratio (EER) for the cooling COP, and COP for the PF (or heating COP)!

Refrigerant circulation rate

$$\dot{m} = \frac{\dot{Q}_{ref}}{h_1 - h_4}$$

Another important parameter is the *refrigerant volumetric flow rate*, often called the theoretical piston displacement.

This is defined as

$$\dot{V} = \dot{m} v_{suction(or\ 1)}$$

This should *not* be confused with the *actual piston displacement*, or as often called, the *swept volume*. The swept volume, for example for a reciprocating compressor, is

$$\dot{V}_{swept} = \left(\frac{\pi D^2}{4} s \right) nN$$

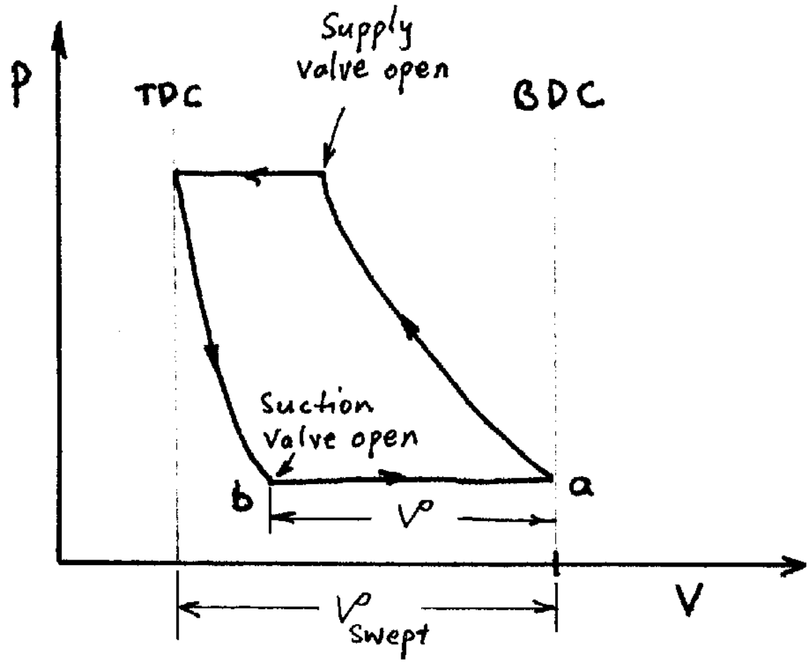
where D = diameter of piston (m)
s = stroke (m)
n = number of cylinders
N = speed (rev/sec)

It is also useful at this stage to *define the volumetric efficiency*.

$$\eta_v = \frac{volume\ flow rate}{swept\, volume\ flow rate}$$

which from the figure below clearly becomes,

$$\eta_v = \frac{V_a - V_b}{V_{swept}} = \frac{\dot{V}}{\dot{V}_{swept}}$$



and hence we can write that,

$$\dot{V} = \eta_v \dot{V}_{swept}$$

Note that the swept volume flow rate $\dot{V}_{swept} = f\left(compressor\ geometry\ only\right)$

Example 1:

A refrigeration system using R22 operates on a standard vapour-compression cycle in which the evaporating temperature is -8ºC and the condensing temperature is 42ºC. The compressor used in this system is a six-cylinder compressor operating at 1740 rpm. The cylinder bore is 67 mm, the piston stroke is 57 mm and for the given operating conditions the volumetric efficiency of the compressor is 77%.

Determine the

(a) volume flow rate measured at compressor suction;

(b) mass flow rate of refrigerant;

(c) refrigerating capacity of the system;

(d) power consumption;

(e) heat rejected from the condenser

(f) COP, and compare this with COP_c.

Solution:

(a)
$$\dot{V} = \eta_v \dot{V}_{swept}$$

and we can calculate,

$$\dot{V}_{swept} = \left(\frac{\pi D^2}{4}\right) snN = \left(\frac{\pi \times .067^2}{4}\right) 0.057 \times 6 \times \frac{1740}{60}$$

$$\dot{V}_{swept} = 0.035 m^3/s$$

Hence

$$\begin{aligned}\dot{V} &= \eta_v \times \dot{V}_{swept} \\ &= 0.77 \times 0.035 \\ &= 0.0269 m^3/s \quad (26.9 L/s)\end{aligned}$$

(b) the mass flow rate is given by,

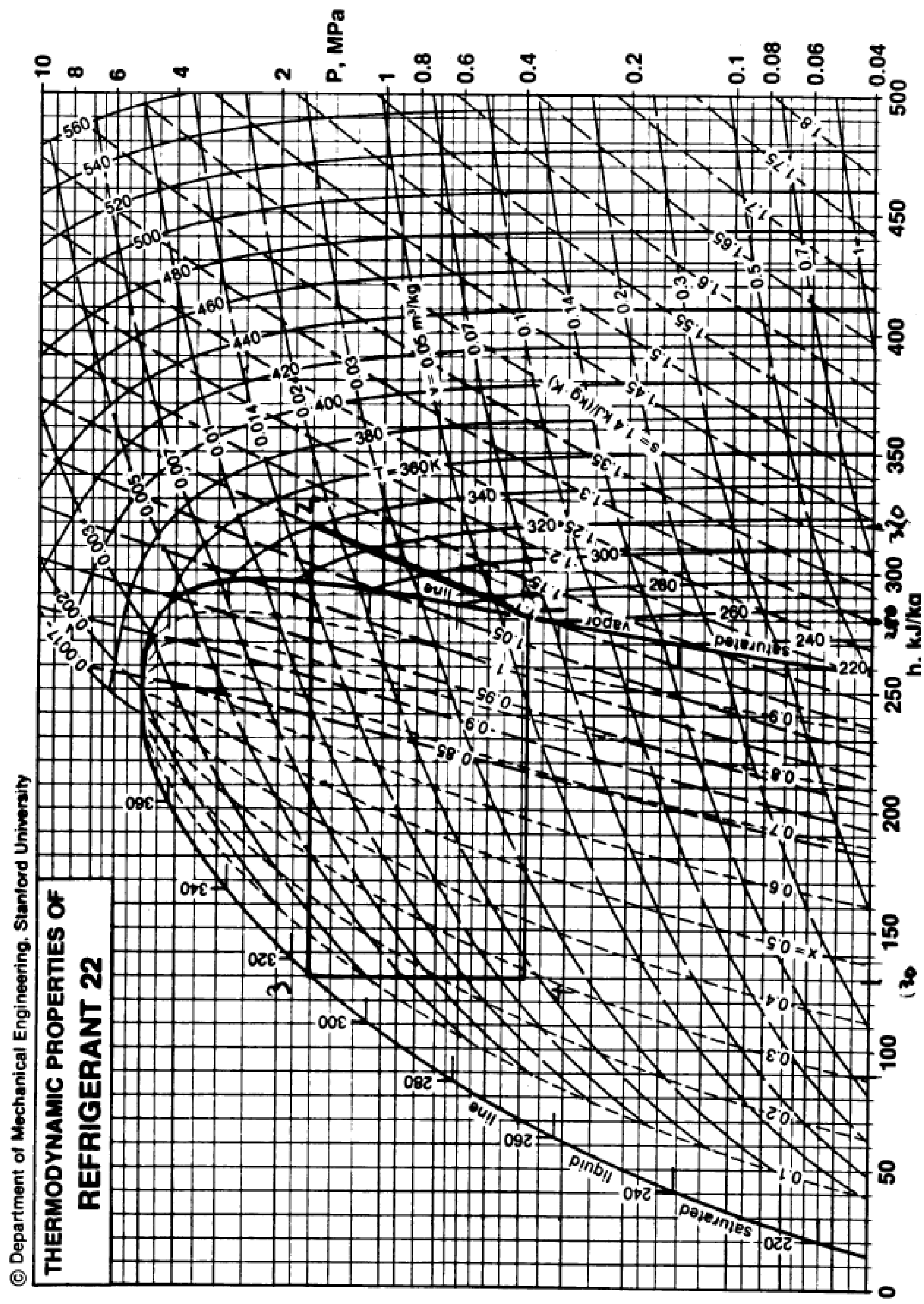
$$\dot{m} = \frac{\dot{V}}{v_1}$$

where $v_{1(suction)} = v_g(-8^oC) = 0.0615 m^3/kg$ (from refrigerant tables)

therefore,

$$\begin{aligned}\dot{m} &= \frac{0.0269}{0.0615} \\ &= 0.438 kg/s\end{aligned}$$

(c) The easiest way to answer the rest of the question is to draw the cycle on a P-h diagram. This is done on the next page.



$$\dot{Q}_{ref} = \dot{m}\left(h_1 - h_4\right)$$

Hence (from the P-h diagram or tables),

$$h_1 = 280 kJ/kg \quad \text{and} \quad h_4 = 130 kJ/kg = h_3$$

$$\dot{Q}_{ref} = 0.438\left(280 - 130\right) = 65.7 kW(R)$$

(d) The power required can be calculated from,

$$\dot{W} = \dot{m}\left(h_2 - h_1\right)$$

hence from the chart we have,

$$h_2 = 320\ kJ/kg \quad \text{and} \quad h_1 = 280 kJ/kg$$

and therefore,

$$\dot{W} = 17.5 kW$$

(e) The heat rejected from the condenser is given by,

$$\dot{Q}_{cond} = \dot{m}\left(h_2 - h_3\right) = 83.2 kW$$

note that it is also $\dot{Q}_{cond} = \dot{Q}_{ref} + \dot{W}$

(f) Finally the COP is calculated from,

$$COP = \frac{\dot{Q}_{ref}}{\dot{W}} = \frac{65.7}{17.5} = 3.75$$

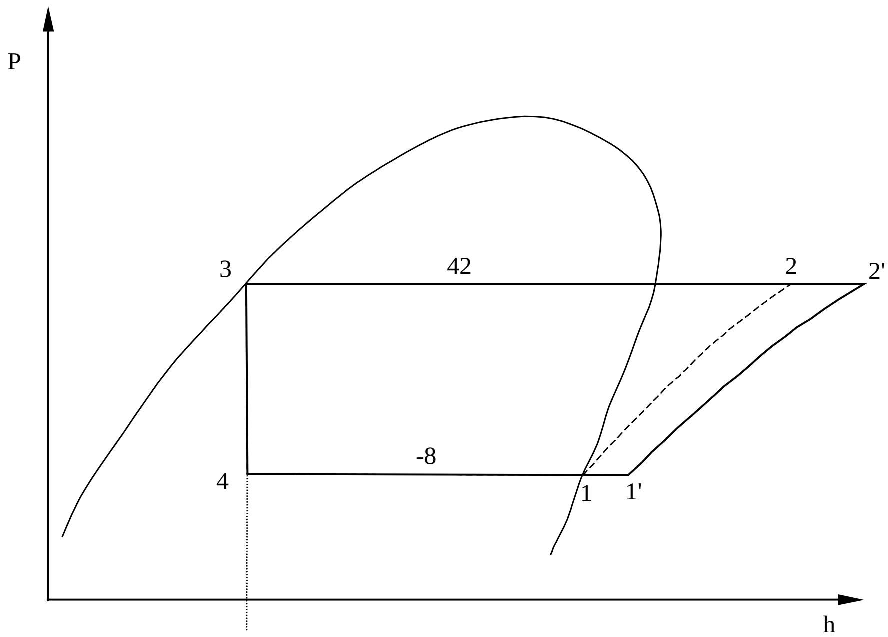
this compares with the Carnot COP of,

$$COP_{carnot} = \frac{T_c}{T_H - T_c} = \frac{\left(273 - 8\right)}{42 - -8} = 5.3$$

Do "Specific Properties" Tell the Whole Story?

In nearly every book on refrigeration and air conditioning, the effect of various changes to the cycle is discussed in terms of specify properties such as the specific refrigerating capacity (i.e. refrigerating effect) and the specific work. But do these properties really tell the whole story?

Let us consider the vapour compression system presented in Example 1. The same hardware, except that we now adjust the controls such that the vapour is now superheated when it enters the compressor.



The compressor is the same as before with the same volumetric efficiency and hence $\dot{V} = 26.9 L/s$.

Two questions are posed:

(1) has the refrigerating capacity of the system increased, decreased or remained the same compared with the original cycle?

(2) is more, less or the same power required than before?

Let us write down the specific quantities.

(a) original system

$$q_{ref} = q_{41} = 150kJ/kg$$

$$w = w_{12} = 40kJ/kg$$

(b) for the new system

from chart $h_{1'} = 290kJ/kg$ and $h_{2'} = 330kJ/kg$

therefore $q_{ref} = q_{41'} = 290 - 130 = 160kJ/kg$

Similarly $w = w_{1'2'} = 330 - 290 = 40kJ/kg$

Well! What do you think the answers are? Let us check.

(1) Since from the chart $v_{1'} = 0.066m^3/kg$ therefore

$$\dot{Q}_{ref} = \left(\frac{\dot{V}}{v_{1'}}\right) q_{41'} = \left(\frac{0.0269}{0.066}\right) \times 160 = 65.2kW$$

compared with the original value of $65.7kW$

Therefore, the refrigerating capacity has gone down slightly (not up, as the refrigerating effect would have us believe). Why is this so? The reason is quite simple. The specific volume has also changed (from 0.0615 to 0.066), resulting in a decrease in the mass flow rate. Therefore, although $\Delta h \uparrow$ $\dot{m} \downarrow$, where $\dot{m}$ goes down more than Δh has gone up.

2) $\dot{W} = \left(\frac{\dot{V}}{v_{1'}}\right) w_{1'2'} = \left(\frac{0.0269}{0.066}\right) \times 40 = 16.3\ kW$

compared with the original value of $17.5kW$

Therefore, the power has gone down (and not remained the same).

Therefore, the specific properties do ***not*** tell the whole story! In fact, they sometimes give a wrong indication, as in this case.

Can we introduce new properties which will give a true indication of trend? Yes we can!

"Volumic" Properties

Recall the equation for refrigerating capacity, i.e.

$$\begin{aligned}\dot{Q}_{ref} &= \dot{m}\left(h_1 - h_4\right) \\ &= \frac{\dot{V}}{v_1}\left(h_1 - h_4\right) \\ &= \dot{V}\left(\frac{h_1 - h_4}{v_1}\right)\end{aligned}$$

where $\dot{V} = \eta_v \dot{V}_{swept}$

Since $\dot{V}$ is constant for a given machine, then we can define

$$\text{Volumic refrigerating effect } = \frac{h_1 - h_4}{v_1}$$

This property has the units of *J/m³* and gives a truer indication of trends.

Similarly,

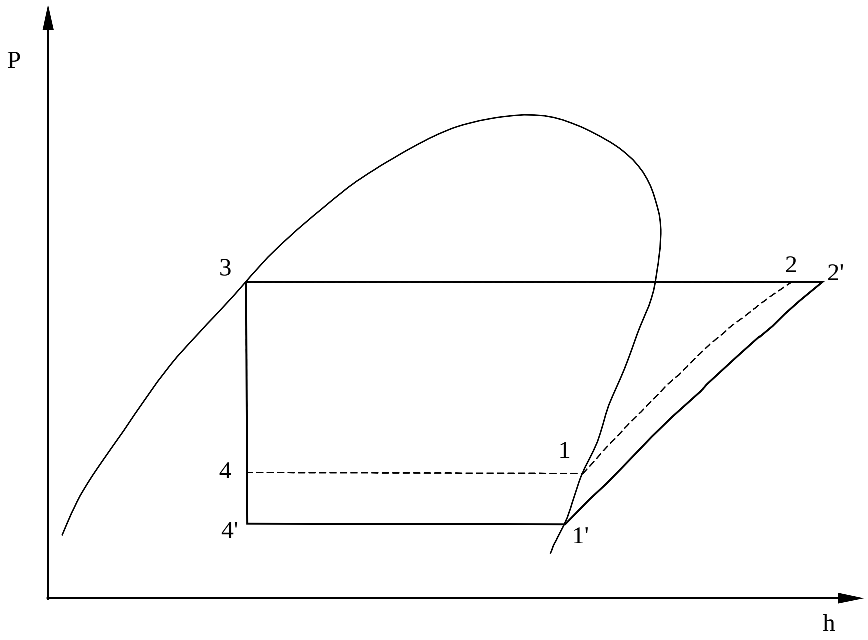
$$\begin{aligned}\dot{W} &= \dot{m}\left(h_2 - h_1\right) \\ &= \dot{V}\left(\frac{h_2 - h_1}{v_1}\right)\end{aligned}$$

Therefore

$$\text{Volumic work of adiabatic compression } = \frac{h_2 - h_1}{v_1}$$

Effect of Operating Conditions

Effect of Evaporator Pressure



Let us consider a system for which we *lower* the evaporator temperature (and hence pressure). What effect will this have on the capacity of the system?

Remember that,

$$\dot{Q}_{ref} = \dot{V}_{swept}\eta_v\left(\frac{h_1 - h_4}{v_1}\right)$$

where $\dot{V}_{swept}$ is fixed for a particular machine.

Similarly

$$\dot{W}_{ad} = \dot{V}_{swept}\eta_v\left(\frac{h_2 - h_1}{v_1}\right)$$

Let us see how these quantities vary with evaporator temperature.

The effect of dropping the evaporator temperature results in,

(i) a decrease in volumic refrigerating effect, since

$$\left(h_{1'} - h_{4'}\right) < \left(h_1 - h_4\right)$$

and also

$$v_{1'} > v_1$$

therefore

$$\left(\frac{h_{1'} - h_{4'}}{v_{1'}}\right) << \left(\frac{h_1 - h_4}{v_1}\right)$$

(ii) a drop in volumetric efficiency η_v due to the higher pressure ratio. Therefore, the refrigerating capacity drops significantly;

(iii) although there is an increase in the specific work required, i.e.

$$\left(h_{2'} - h_{1'}\right) > \left(h_2 - h_1\right)$$

the specific volume also increases. Therefore, depending on the relative rate of increase of $h_{2'} - h_{1'}$ and $v_{1'}$ will determine whether the volumic work of adiabatic compression increases or decreases.

Hence

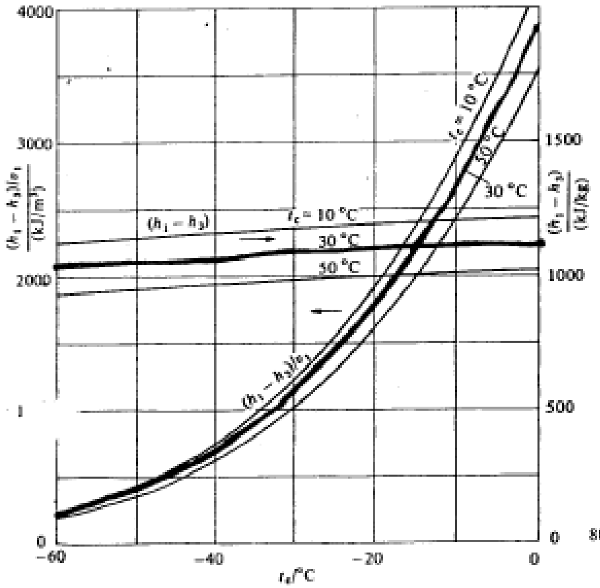
$$\left(\frac{h_{2'} - h_{1'}}{v_{1'}}\right) \begin{matrix} > \\ \text{or} \\ < \\ \text{or} \\ = \end{matrix} \left(\frac{h_2 - h_1}{v_1}\right)$$

Therefore, the power requirements may increase, decrease or remain the same.

Since the decreases in volumic refrigerating effect is always larger than variations in power requirements, the ***COP will always drop*** when the evaporating temperature is reduced.

Effect of varying Evaporator Temperature

Fig. 1. The specific refrigerating effect $(h_1 - h_3)$ and the volumic refrigerating effect $(h_1 - h_3)/v_1$ for ammonia as functions of evaporating temperature at three condensing temperatures.



Refrigerant is Ammonia

Fig 2. The specific work of isentropic compression $(h_2 - h_1)_s$ and the volumic work of isentropic compression $(h_2 - h_1)_s/v_1$ for ammonia as functions of evaporating temperature at three condensing temperatures.

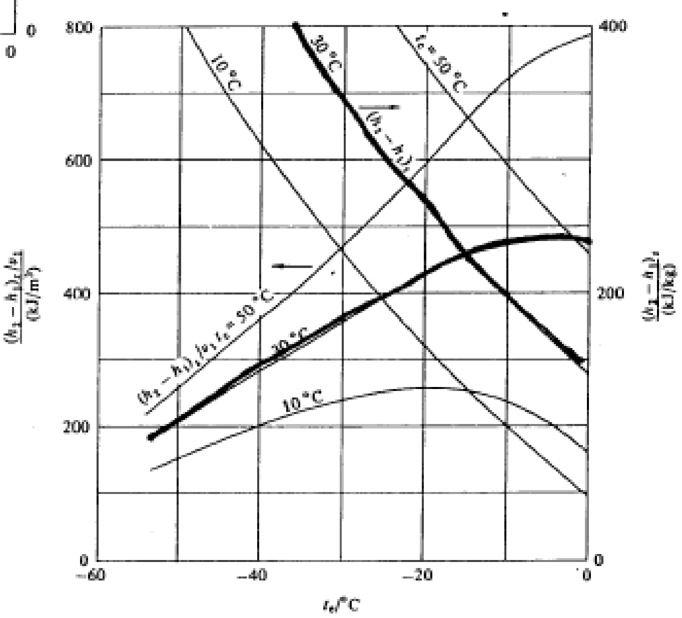
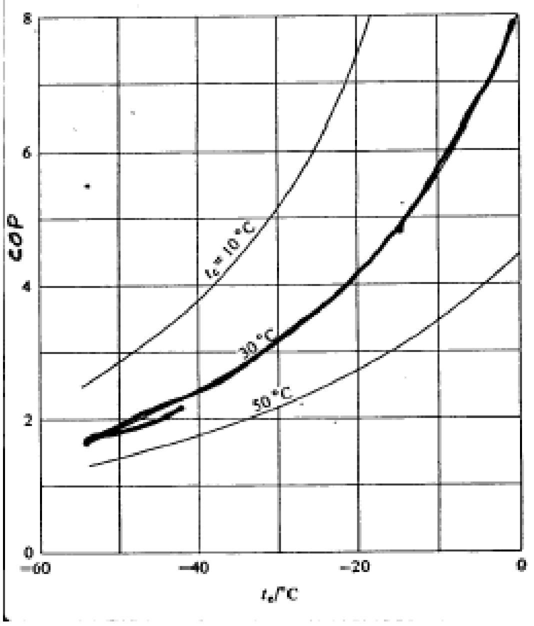
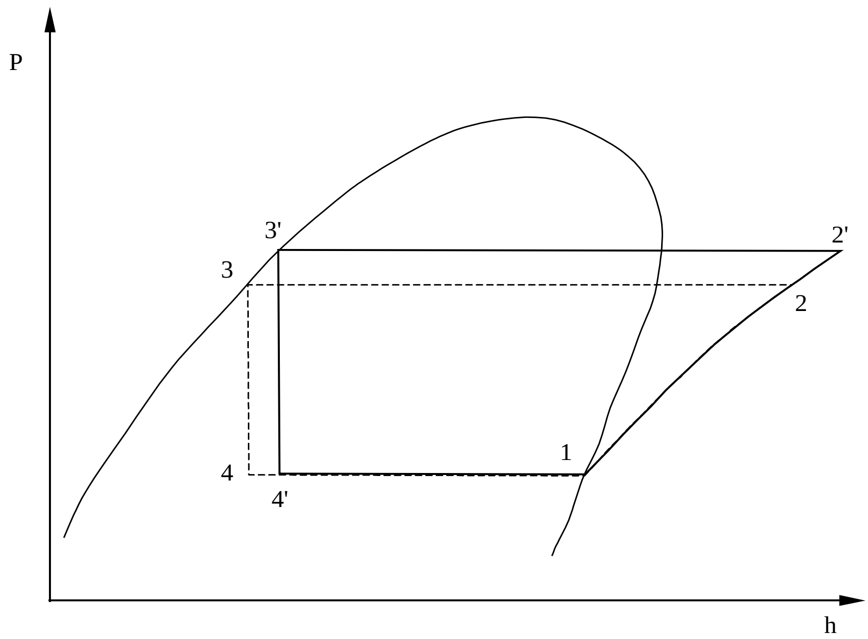


Fig 3. The isentropic coefficient of performance for ammonia as a function of evaporating temperature at three condensing temperatures.



Effect of Condenser Pressure



The effect of increasing the condenser temperature results in,

(i) decrease in refrigerating effect and hence a drop in volumic refrigerating effect;

(ii) decrease in volumetric efficiency η_v due to the higher pressure ratios;

(iii) increase in specific work and hence in the volumic work of adiabatic compression.

All these factors contribute to a decrease in refrigerating capacity and an increase in power consumption.

It can be seen that the effect of variation in T_c is not as severe on the capacity, but, as might be expected, have a severe effect on the power consumption.

Effect of varying Condenser Temperature

Fig. 1. The specific refrigerating effect $(h_1 - h_3)$ and the volumic refrigerating effect $(h_1 - h_3)/v_1$ for ammonia as functions of evaporating temperature at three condensing temperatures.

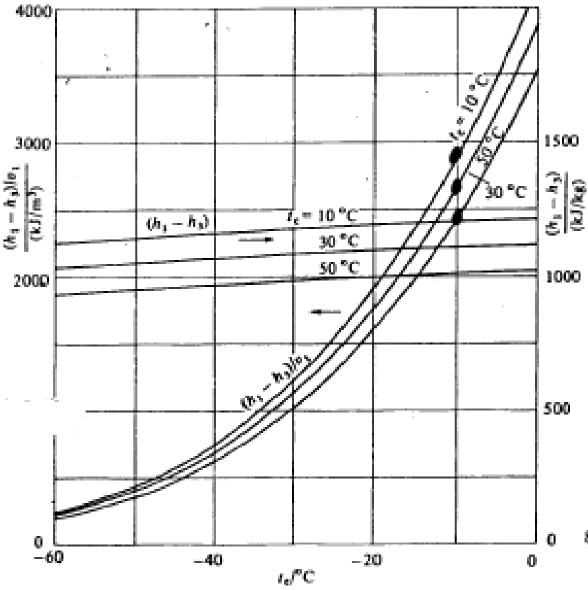


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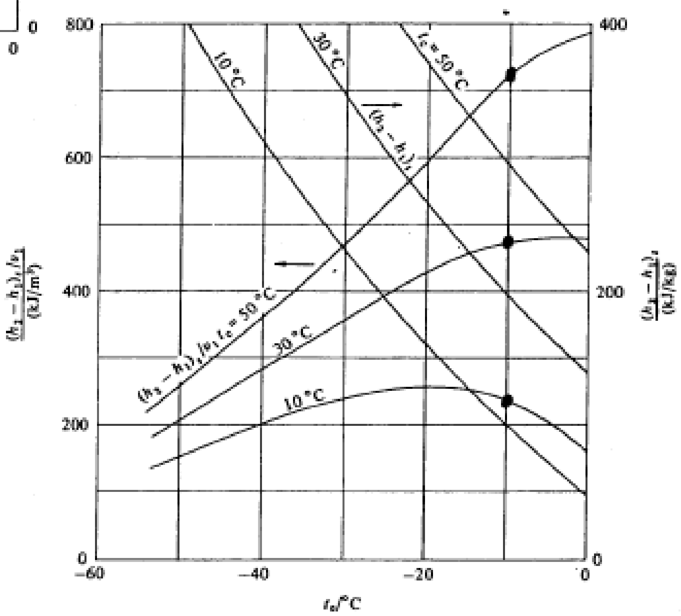
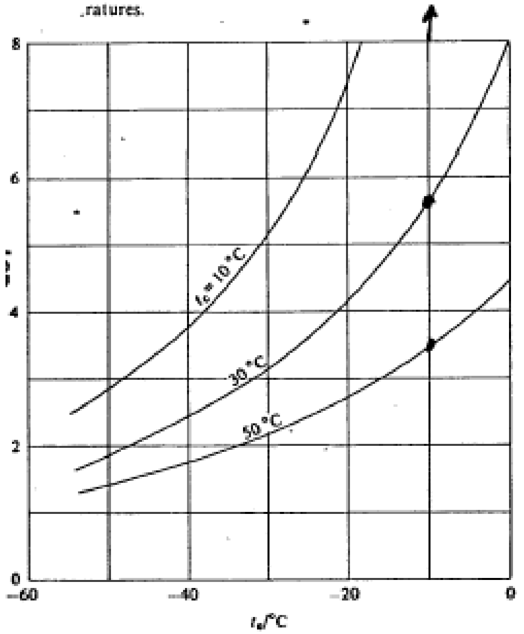
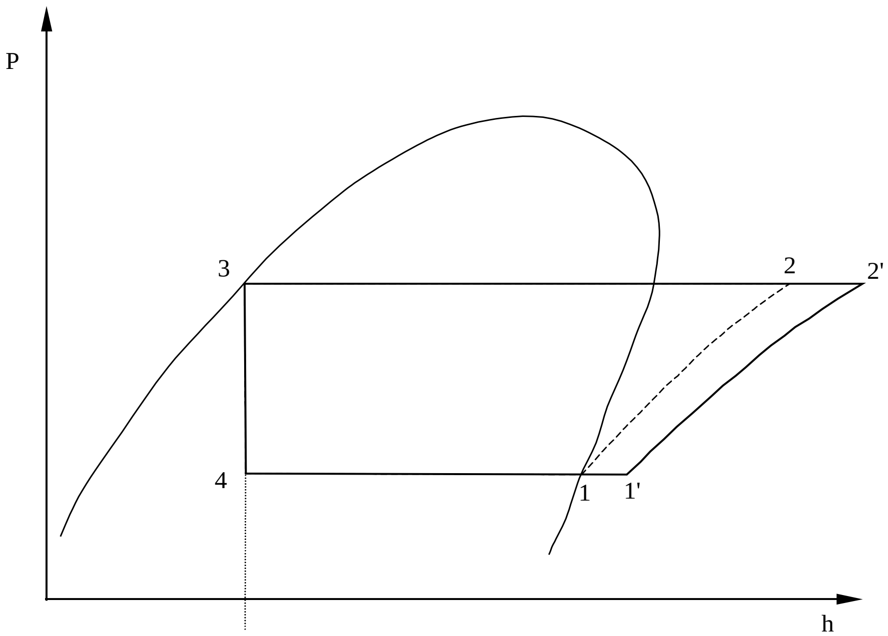
Fig 2. The specific work of isentropic compression $(h_2 - h_1)_s$ and the volumic work of isentropic compression $(h_2 - h_1)_s/v_1$ for ammonia as functions of evaporating temperature at three condensing temperatures.



3. The isentropic coefficient of performance for ammonia as a function of evaporating temperature at three condensing temperatures.



Effect of Suction Vapour Superheat



Superheating of the suction vapour is advisable in practice for reciprocating compressors because it ensures complete vaporisation of the liquid in the evaporator before it enters the compressor.

Increasing the temperature of the vapour entering the compressor from t_1 to $t_{1'}$ results in,

(i) an increase in the refrigerating effect from $(h_1 - h_4)$ to $(h_{1'} - h_4)$ whilst the specific volume increases from v_1 to $v_{1'}$. Hence, whether

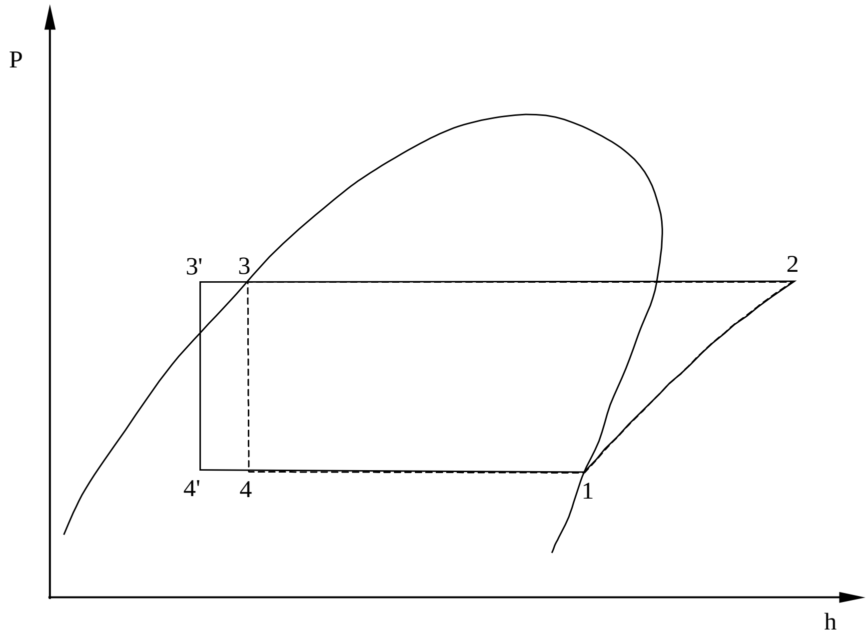
$$\left(\frac{h_{1'} - h_4}{v_{1'}}\right) \overset{>}{\underset{<}{or}} \left(\frac{h_1 - h_4}{v_1}\right)$$

depends on the relative rates of increase of $(h_{1'1} - h_4)$ and $v_{1'}$.

(ii) Similarly, the volumic work may increase or decrease.

The net effect is that the COP may increase or decrease.

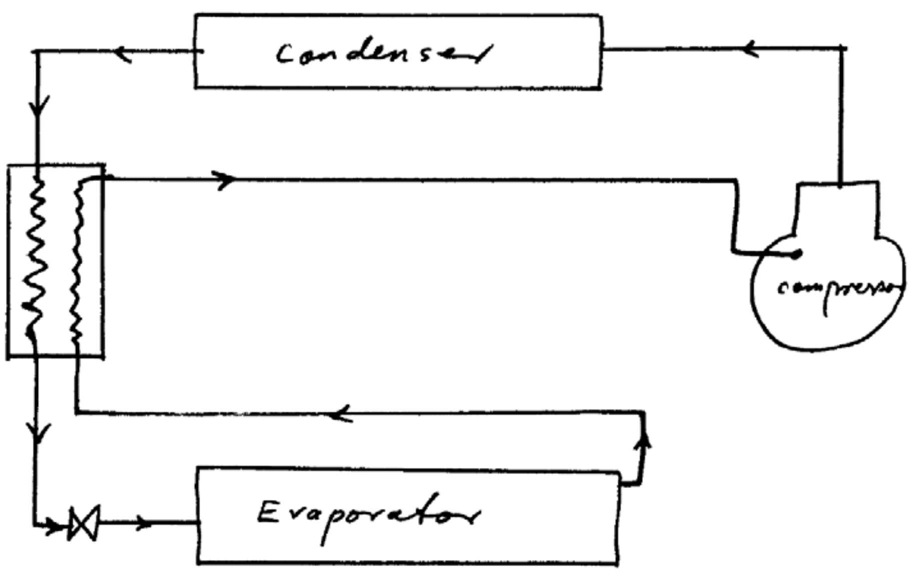
Effect of Liquid Subcooling



It can be seen that liquid subcooling results in a greater refrigerating effect for the same compressor work.

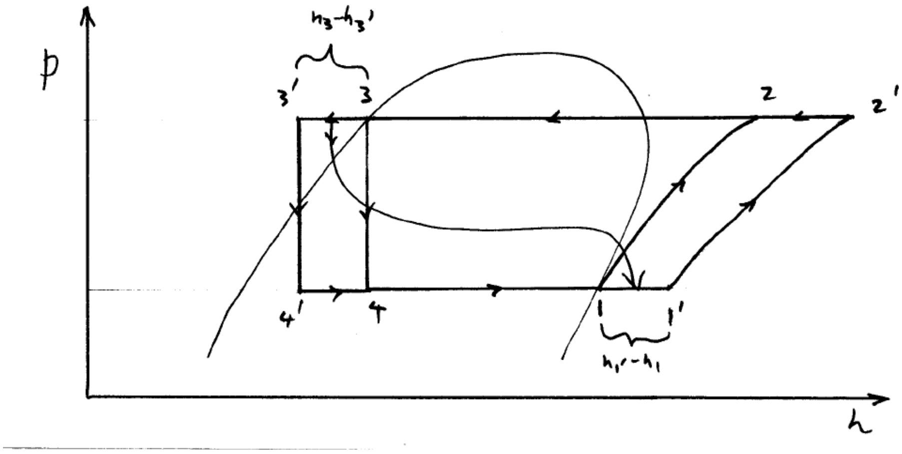
The only disadvantage is that either a subcooler or additional cooling is required at the condenser.

(i) Using a Liquid-Vapour Regenerative Heat Exchanger



A liquid-vapour heat exchanger may be installed as shown in the above figure. In this the refrigerated vapour from the evaporator is superheated while the liquid from the condenser is subcooled.

The effect on the thermodynamic cycle is as shown.



Since the mass flow rate of the liquid and vapour is the same, we have for the heat exchanger

$$q_{HE} = h_{1'} - h_1 + h_3 - h_{3'}$$

However, the heat exchanger is definitely justified in situations where the vapour entering the compressor must be superheated to ensure that no liquid enters the compressor.

Another practical reason for using the heat exchanger is to sub-cool the liquid from impeding the flow of refrigerant through the expansion valve.

An example of a typical liquid-to-suction heat exchanger is shown below.

Figure 10-14. A liquid-to-suction heat exchanger before enclosure with outer housing. *(Refrigeration Research, Inc.)*

Example 2:

A manufacturer's table for a certain compressor on R12 gives the refrigerating capacity as 9.65 kW and the shaft power as 3.82 kW at the following conditions

t_e = -20ºC

t_c = 30ºC

temperature of liquid leaving condenser = 25ºC

temperature of vapour entering compressor = 18ºC

(a) Determine the mass flow rate at these conditions and the isentropic efficiency.

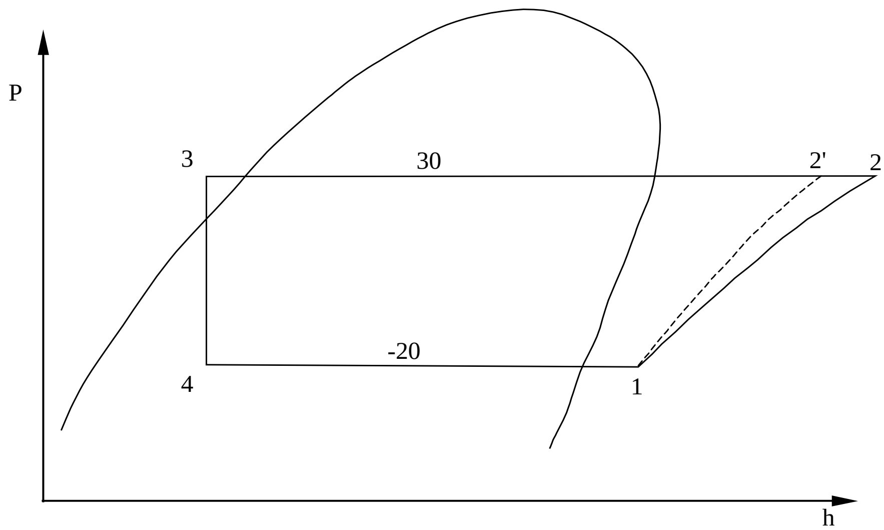
(b) Supposing that the compressor operates an evaporator with liquid feed via a TX valve giving a superheat at the outlet (of the evaporator) of 5 K and the superheating to 18ºC is done in a liquid to suction heat exchanger. Determine,

(i) the temperature of the liquid reaching the expansion valve;

(ii) the surface area required in the het exchanger based on an overall coefficient of heat transfer of 50 W/m^2K.

Solution:

(a)



(i) we know that,

$$\dot{m} = \frac{\dot{Q}_{ref}}{h_1 - h_4}$$

from the tables we have $h_4 = h_3 = h_f(25^0C) = 59.7\ kJ/kg$

and $h_1 = h\left(-20^oC, 38K_{superheat}\right) = 201.86 kJ / kg$

hence $h_1 - h_4 = 142.16 kJ / kg$

and therefore we have,

$$\dot{m} = \frac{9.65}{142.16}$$
$$= 0.0679 kg / s$$

(ii) to determine he isentropic efficiency we use,

$$\eta_{1s} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

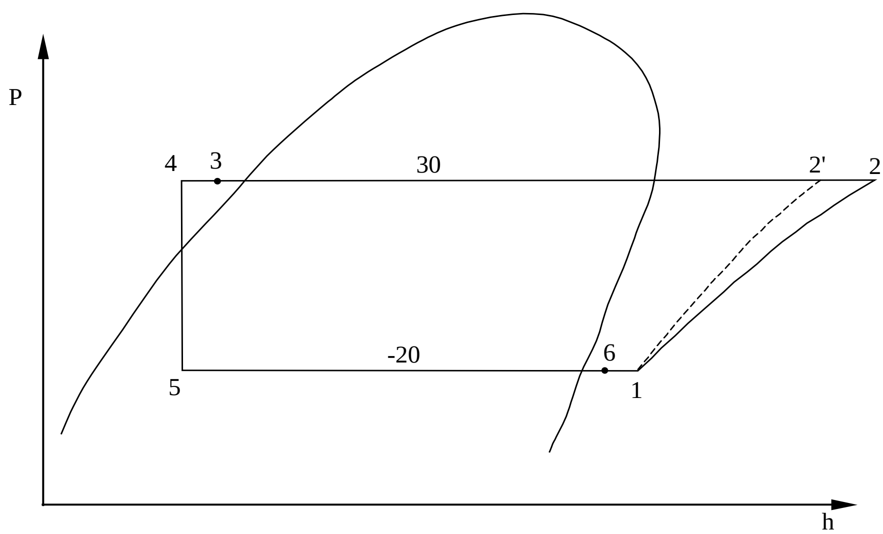
but we can calculate,

$$h_2 - h_1 = w = \frac{\dot{W}}{\dot{m}} = \frac{3.82}{0.0679} = 56.27 kJ / kg$$

to find h_{2s}, we use $s_{2s} = s_1 = 0.7938 kJ / kgK$ hence from tables (at $t_c = 30^o C$), $t_{2s} = 79^oC$ and $h_{2s} = 234.8 kJ / kg$

$$\therefore \qquad \eta_{1s} = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{234.8 - 201.86}{56.27} = 59\%$$

(b)



From tables $h_6 = h\left(-20^oC\, evap, 5K\, superheat\right) = 181.7 kJ/kg$

But now for the heat exchanger

$$h_1 - h_6 = h_3 - h_4$$

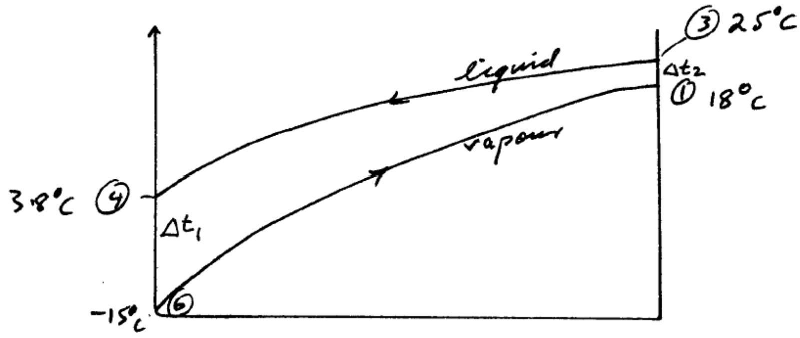
but $h_3 = 59.7 Jk/kg$ and $h_1 = 201.86 kJ/kg$

$\therefore$ $h_4 = 59.7 - 201.86 + 181.7 = 39.54 kJ/kg$

From tables at (30ºC condensing)

$h_4 = h_f\left(t_4\right)$ therefore $t_4 = 3.8^o C$

For the heat exchanger



Since,

$$\dot{Q}_{HE} = UA\Delta T_{LMTD}$$

where

$$\Delta T_{LMTD} = \frac{\Delta t_1 - \Delta t_2}{ln\left(\dfrac{\Delta t_1}{\Delta t_2}\right)}$$

$$= \frac{18.8 - 7}{ln\left(\dfrac{18.8}{7}\right)}$$

$$= 11.94$$

But we also know that

$$\dot{Q}_{HE} = \dot{m}\left(h_1 - h_6\right)$$

$$= \left(0.0679\right)\left(201.86 - 181.7\right)$$

$$= 1.369 kW$$

Therefore we can calculate the require heat transfer area,

$$A = \frac{\dot{Q}_{HE}}{U \Delta T_{LMTD}}$$

given $U = 50W / m^2K$ yields,

$$A = \frac{1.369 \times 10^3}{50 \times 11.94}$$
$$= 2.29m^2$$

Effects of Heat Transfers and Pressure Drops

Usually because of heat transfer and pressure drop the state of the fluid changes in the connecting pipes. Let us consider their effects.

(a) Suction Pipe

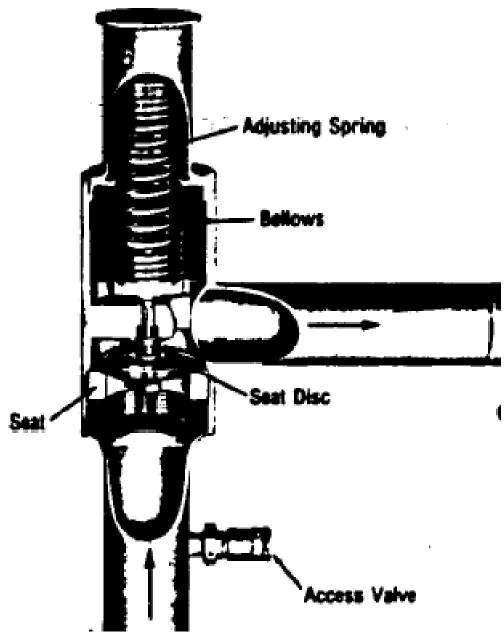
The most important changes are those in the suction pipe because they modify the state of the vapour at the entry to the compressor and cause significant changes in performance.

Heat transfer to the vapour in the suction pipe is useless refrigeration and, in general, is minimised by "insulation of the pipe". Also $\left(v\uparrow \therefore \dot{m}\downarrow \therefore \dot{Q}_{ref}\downarrow\right)$.

Pressure drop in the suction pipe is always harmful because it increases v and also increases the pressure ratio across the compressor. The volumic refrigerating effect is reduced, η_v is reduced and specific work of compression is increased, resulting is a reduction in COP.

Pressure drop is sometimes caused deliberately by a throttling valve as a simple but not very efficient means of reducing the refrigeration capacity.

Values used for suction throttling are called *Evaporator pressure regulators (EPR)* (see figure below).



These close when the pressure in the evaporator falls below a set value, thus maintaining a set evaporator pressure (or temperature). It is common to use the evaporator pressure regulator to control the evaporator temperature and the thermostatic expansion valve to control the mass flow rate of refrigerant.

(b) Delivery Line

Pressure drop between the compressor and condenser is harmful because it increases the work required, (i.e. a higher compressor discharge pressure to give the required condensing pressure) and because by raising the pressure the ratio the volumetric efficiency drops.

(c) Liquid Line

Heat loss from the liquid line going to the receiver and then to the expansion valve provides useful sub-cooling and increases the refrigerating capacity without any increase in power. However, occasionally the heat transfer is a heat gain. Then there is a danger that the liquid refrigerant will be partially vaporised.

This not only reduces the refrigeration capacity of the system but, more importantly, upsets the operation of the expansion valve which is designed to take only liquid. When this happens the refrigerating capacity of the plant falls dramatically.

Pressure drop in the liquid line has a harmful effect since it may also result in the partial vaporisation of the liquid refrigerant.

The most usual cause of pressure drop in liquid lines is not, however, fluid friction, but rather change in height, when the liquid flows upwards from a receiver to an expansion valve and evaporator at a higher level.

The degree of sub-cooling required to prevent vaporisation within a given lift can be determined by applying the steady state energy equation, viz.

$$0 = \left(h_2 - h_1\right) + g\left(z_2 - z_1\right)$$

However, since we are assuming the temperature remains constant and only the pressure changes, then from definition of h

$$h_2 - h_1 \approx v_f\left(p_2 - p_1\right) \qquad \text{since} \quad \mathrm{u}_1 \approx \mathrm{u}_2$$

$$p_1 - p_2 = \frac{g\left(z_2 - z_1\right)}{v_f}$$

where p_1 is the pressure at the bottom of the riser; p_2 is the saturation pressure of the liquid at temperature (t_1) and v_f is the specific volume determined at t_1.

Example 3:

Ammonia liquid leaves a receiver at a pressure of 1.35 MPa and a temperature of 34ºC and flows adiabatically up a vertical pipe. At what height above the inlet will it become saturated liquid?

Solution:

From the ammonia tables, at 1.35 MPa, t = 35ºC. Therefore, the liquid is sub-cooled 1K.

The saturation pressure at 34ºC is 1312 kPa.

Therefore

$$p_1 - p_2 = \frac{g(z_2 - z_1)}{v_f}$$

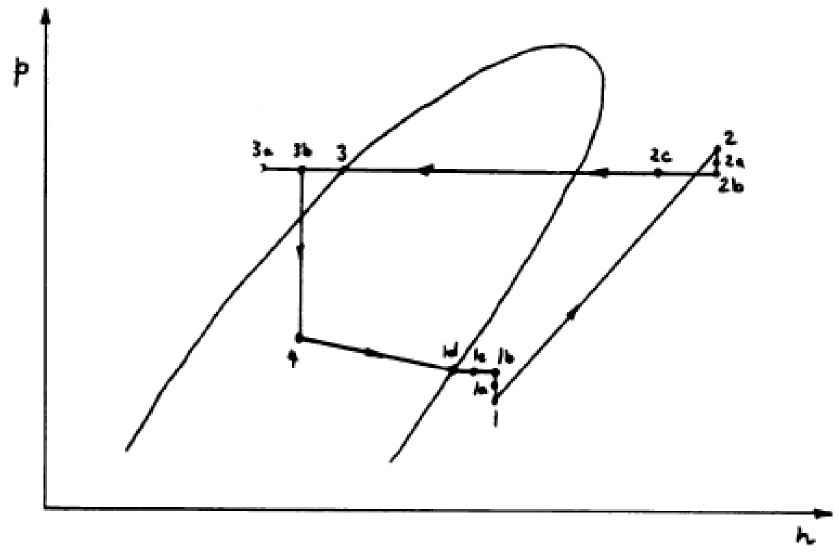
$v_f(t_1) = 0.0017$ and hence

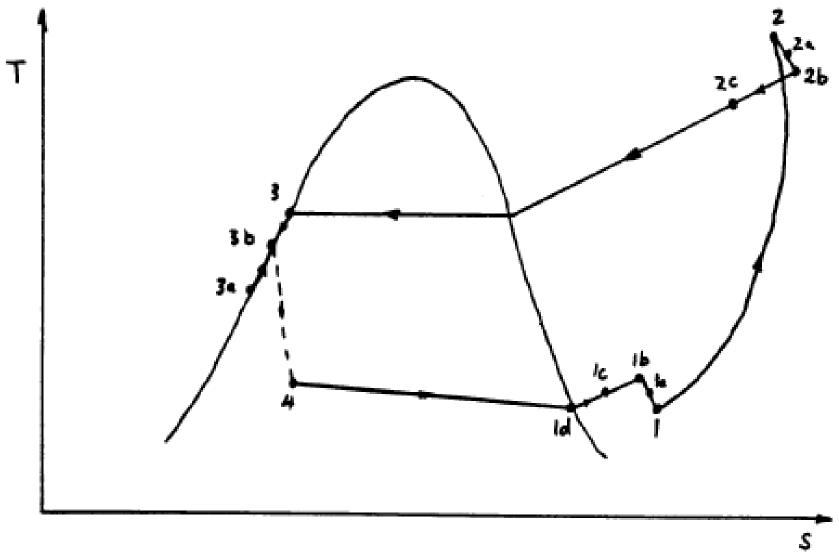
$$p_1 - p_2 = 1350 - 1312 = \frac{9.81(\Delta z)}{0.0017}$$

therefore

$$\Delta z = 6.6\text{m}$$

Actual Vapour Compression Cycle





Due to the flow of the refrigerant through the condenser, evaporator and piping, there will be drops in pressure. In addition, there will be heat losses or gains depending on the temperature different between the refrigerant and the surroundings. Further, compression will be polytropic with heat transfer and friction instead of isentropic.

The actual vapour compression cycle may have some or all of the items of departure from the simple cycle as shown in the p-h and T-s diagrams.

The cycle consists of

(i) Superheating of the vapour in the evaporator, 1d-ic.

(ii) Heat gain and superheating of the vapour in suction line, 1c-1b.

(iii) Pressure drop in the suction line, 1b-1a.

(iv) Pressure drop at the compressor-suction valve, 1a-1.

(v) Polytropic compression with friction and heat transfer to the surroundings instead of isentropic compression, 1-2.

(vi) Pressure drop at the compressor discharge valve, 2-2a.

(vii) Pressure drop in the delivery line, 2a-2b.

(viii) Heat loss and desuperheating of the vapour in the delivery line, 2b-2c.

(ix) Pressure drop in the condenser, 2c-3.

(x) Subcooling of the liquid in the condenser or subcooler, 3-3a.

(xi) Heat gain in the liquid line, 3a-3b.

(xii) Pressure drop in the evaporator, 4-1d.

Note that a larger pressure drop occurs in the evaporator than the condenser due to the (momentum pressure drop) change from liquid to vapour in the pipe, resulting in an increase in kinetic energy and hence drop in pressure.