THERMODYNAMICS

TUTORIAL 5

HEAT PUMPS AND REFRIGERATION

On completion of this tutorial you should be able to do the following.

- Discuss the merits of different refrigerants.
- Use thermodynamic tables for common refrigerants.
- Define a reversed heat engine.
- Define a refrigerator and heat pump.
- Define the coefficient of performance for a refrigerator and heat pump.
- Explain the vapour compression cycle.
- Explain modifications to the basic cycle.
- Sketch cycles on a pressure - enthalpy diagram.
- Sketch cycles on a temperature - entropy diagram.
- Solve problems involving isentropic efficiency.
- Explain the cycle of a reciprocating compressor.
- Define the volumetric efficiency of a reciprocating compressor.
- Solve problems involving reciprocating compressors in refrigeration.
- Explain the ammonia vapour absorption cycle.

1. INTRODUCTION

It is possible to lower the temperature of a body by use of the thermo-electric affect (reversed thermo-couple or Peltier effect). This has yet to be developed as a serious refrigeration method so refrigerators still rely on a fluid or refrigerant which is used in a reversed heat engine cycle as follows.

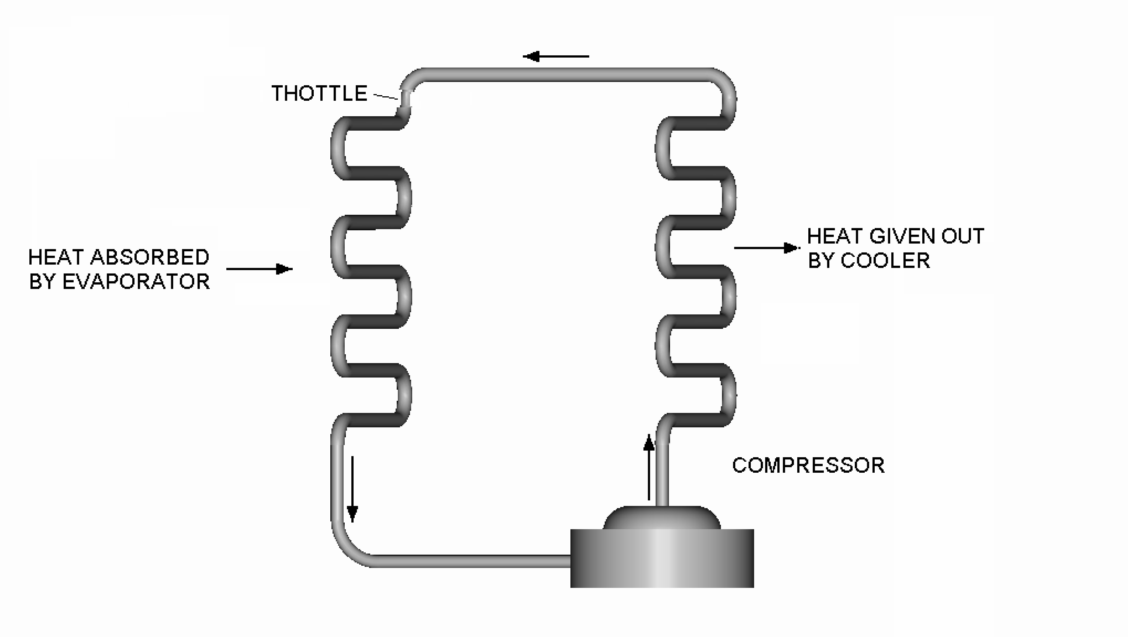

Figure 1

Heat is absorbed into a fluid (this is usually an evaporator) lowering the temperature of the surroundings. The fluid is then compressed and this raises the temperature and pressure. At the higher temperature the fluid is cooled to normal temperature (this is usually a condenser). The fluid then experiences a drop in pressure which makes it go cold (this is usually a throttle valve) and able to absorb heat at a cold temperature. The cycle is then repeated.

Various fluids or refrigerants are used in the reversed thermodynamic cycle. Refrigerants such as air, water and carbon dioxide are used but most refrigerants are those designed for vapour compression cycles. These refrigerants will evaporate at cold temperatures and so the heat absorbed is in the form of latent energy. Let's look at the properties of these and other refrigerants.

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2. REFRIGERANTS

Refrigerants are given R numbers. Carbon dioxide, for example is R744. Some of them are dangerous if released because they are either explosive or toxic. Toxic refrigerants are placed in categories. Sulphur dioxide, for example, is classed as toxic group 1 which means that death occurs after breathing it for 5 minutes.

In the past the most popular fluids have been ammonia (R717),fluorocarbons and halo-carbons. The most popular of these is R12 or dichlorodifluoromethane (CF_2Cl_2).

The type of refrigerant used in a cycle is largely governed by the evaporation temperature required and its latent capacity. Below is a list of some of them.

Refrigerant	R number	Evaporation temp. at 1.013 bar.(^oC)	Toxic group
$C\,Cl_3\,F$	R11	24	5
$C\,Cl_2F_2$	R12	-30	6
$C\,ClF_3$	R13	-82	6
$C\,F_4$	R14	-128	6
$CH\,Cl_2F$	R21	9	4
$CH\,Cl\,F_2$	R22	-40	5
$CH\,F_3$	R23	-84	5
$C\,Cl_2\,F\,C\,Cl\,F_2$	R113	47	4
$C\,Cl_2\,F\,C\,F_3$	R114A	3	6
$C\,Cl_2\,F_2\,C\,Cl\,F_2$	R114	3	6
$C\,Cl_2\,F_2\,C\,F_3$	R115	-39	

All the above are Halo-Carbons and Fluro-carbons which are non-flammable and may be detected by a halide torch or electric cell sensor. Other refrigerants are shown below.

Ammonia is flammable and detected by going white in the presence of sulphur dioxide. It has a strong characteristic pungent smell. Death occurs when breathed for 30 minutes.

NH_3	R717	-33	2

Carbon Dioxide is safe and non-toxic but it can suffocate.

CO_2	R744	-78

Sulphur Dioxide is highly toxic and does not burn.

Other refrigerants are in the Hydro-Carbon groups such as Propane, Butane and Ethane. These are explosive. Because of the problems with damage to the ozone layer, new refrigerants such as R134a have been developed and are now included in the thermodynamic tables.

Now let's look at the use of thermodynamic tables for refrigerants.

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3. TABLES

The section of the fluid tables devoted to refrigerants is very concise and contains only two superheat temperatures. The layout of the tables is shown below.

							15K		30 K	
t	p_s	v_g	h_f	h_g	s_f	s_g	h	s	h	s

t is the actual temperature in degrees Celsius.
p_s is the saturation pressure corresponding to the temperature.
It follows that if the refrigerant is wet or dry saturated, it must be at temperature t and pressure p_s. If the refrigerant has 15 degrees of superheat, then the actual temperature is t+15 and the properties are found under the 15 K heading. Similarly if it has 30 K of superheat, its actual temperature is t+30.

For example, R12 at 2.191 bar and 20°C must have 30 K of superheat since its saturation temperature would is -10°C. From the 30 K columns we find that h=201.97 kJ/kg and s = 0.7695 kJ/kg K.

When dealing with liquid refrigerant, take the properties as h_f and s_f at the given temperatures. The pressures are never very high so the pressure term will not cause much error.

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4. VAPOUR COMPRESSION CYCLES

4.1 THE BASIC CYCLE

Refrigeration/heat pump cycles are similar to heat engine cycles but they work in reverse and are known as reversed heat engine cycles. A basic vapour cycle consists of isentropic compression, constant pressure cooling, isentropic expansion and constant pressure heating. You may recognise this as a reverse of the Rankine cycle or even the reverse of a Carnot cycle. The heating and cooling will involve evaporation and condensing. Let's consider the cycle first conducted entirely with wet vapour.

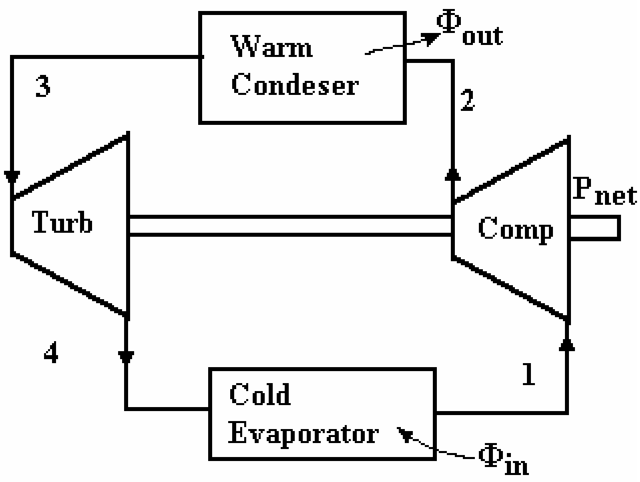

Figure 2

The basic principle is that the wet vapour is compressed and becomes dryer and warmer in the process. It is then cooled and condensed into a wetter vapour at the higher pressure. The vapour is then expanded. Because of the cooling, the expansion back to the original pressure produces a fluid which is much colder and wetter than it was before compression. The fluid is then able to absorb heat at the cold temperature becoming dryer in the process and is returned to the original state and compressed again. The net result is that heat is absorbed at a cold temperature and rejected at a higher temperature. Work is needed to drive the compressor but some of it is returned by the turbine.

The thermodynamic cycle for refrigerators is often shown on a pressure – enthalpy diagram (p – h) and professional charts are available but not used in the Engineering Council exams. Figure 3 shows the basic cycle.

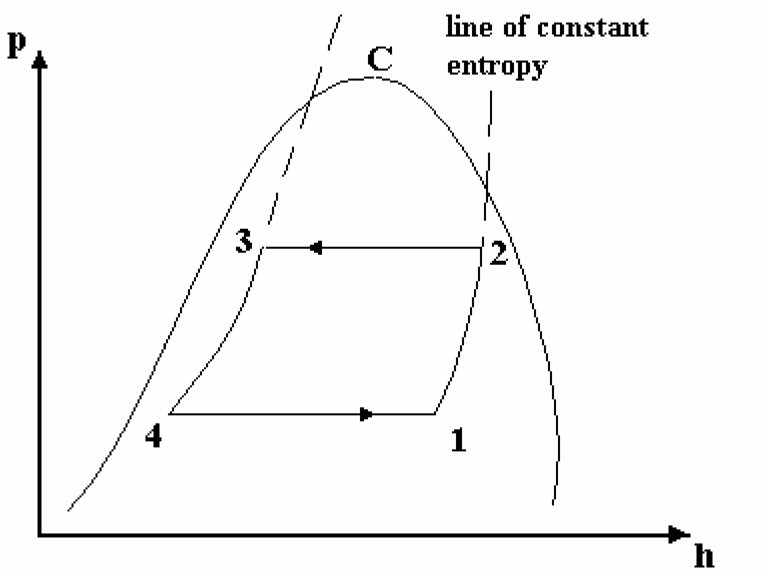

Figure 3

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The four thermodynamic processes are

1 - 2 Isentropic compression. $P(in) = m_r(h_2 - h_1)$
2 - 3 Constant pressure cooling. $\Phi(out) = m_r(h_3 - h_2)$
3 - 4 Isentropic expansion. $P(out) = m_r(h_3 - h_4)$
4 - 3 Constant pressure heating. $\Phi(in) = m_r(h_4 - h_1)$
m_r = mass flow rate of refrigerant.

In practice wet vapour is difficult to compress and expand so the refrigerant is usually dry before compression and superheated after. The cooling process may produce anything from wet vapour to under-cooled liquid. The expansion of a liquid in a turbine is impractical and so a throttle is used instead.

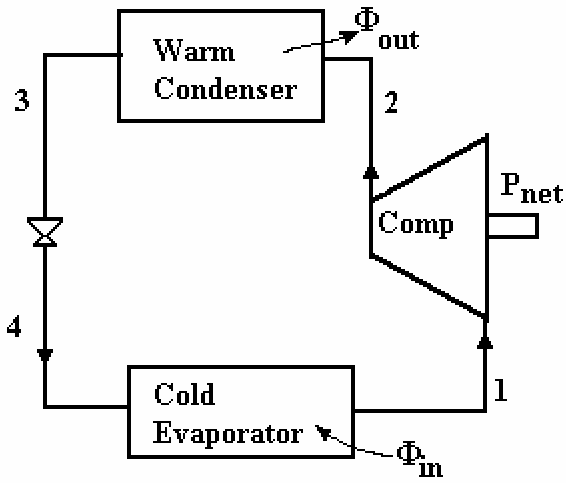

Figure 4

A throttle produces no useful work but it converts the pressure into internal energy. This makes the liquid evaporate and since the saturation temperature goes down it ends up cold. The enthalpy before and after a throttle are the same. The entropy increases over a throttle.

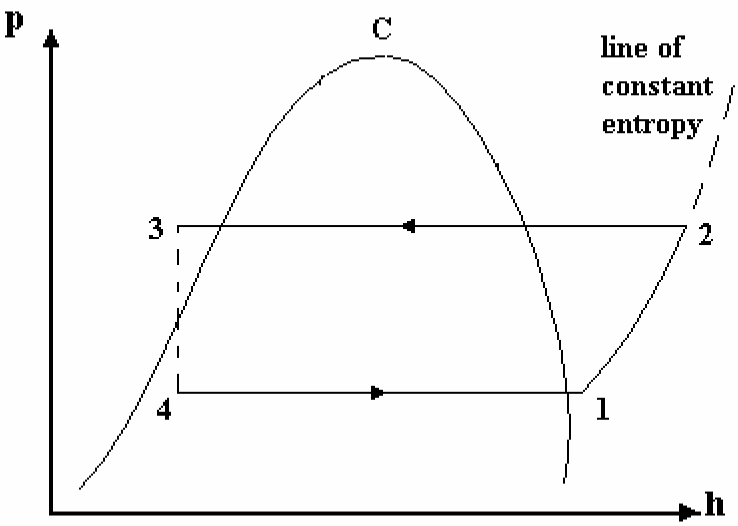

Figure 5

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The cycle may also be drawn on a temperature entropy diagram as shown. The conditions shown are wet at (1), superheated at (2) and under-cooled at (3). These conditions vary.

The four thermodynamic processes are

1 - 2 Isentropic compression.
$P(in) = m_r(h_2 - h_1)$

2 - 3 Constant pressure cooling.
$\Phi(out) = m_r(h_3 - h_2)$

3 - 4 Throttle ($h_3 = h_4$)

4 - 3 Constant pressure heating.
$\Phi(in) = m_r(h_4 - h_1)$

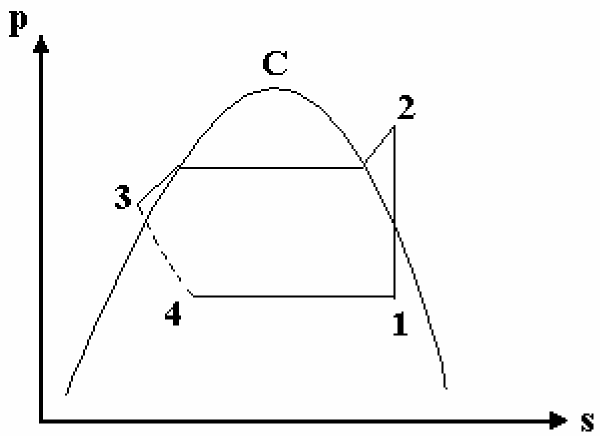

Figure 6

4.2 COEFFICIENT OF PERFORMANCE

The second law of thermodynamics tells us that no heat engine may be 100% efficient. In the reversed cycle, the reverse logic applies and it will be found that more energy is given out at the condenser and more absorbed in the evaporator, than is needed to drive the compressor. The ratio of heat transfer to work input is not called the efficiency, but the coefficient of performance or advantage.

There are two coefficients of performance for such a cycle, one for the refrigeration effect and one for the heat pump effect.

4.2.1 REFRIGERATOR

A refrigerator is a device for removing heat at a cold temperature so we are interested in the heat absorbed in the evaporator Φ(in). The coefficient of performance is also called the advantage and is defined as

$$\textbf{C.O.P.} = \Phi(in)/P(in)$$

The heat absorbed is called the refrigeration affect.

4.2.1 HEAT PUMP

A heat pump is a device for producing heat so we are interested in the heat given out in the cooler Φ(out). The coefficient of performance is defined as

$$\textbf{C.O.P.} = \Phi(out)/P(in)$$

It is usual to find a convenient source of low grade heat for the evaporator such as the atmosphere or a river. The heat is removed from this source and upgraded to higher temperature by the compressor. Both the work and the heat absorbed are given out at the higher temperature from the cooler.

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5. MODEL REVERSED HEAT ENGINE

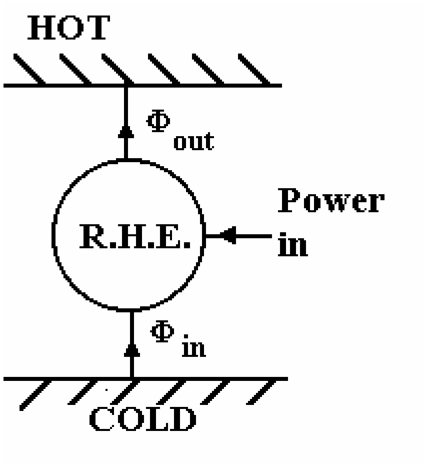

Figure 7

The ideal model pumps heat from a cold source to a hot place. The 1st. Law of Thermodynamics applies so

$$\Phi(in) + P(in) = \Phi(out)$$

$$\text{C.O.P. (refrigerator)} = \Phi(in)/P(in) = \Phi(in)/\{\Phi(out) - \Phi(in)\}$$

If the heat transfers are reversible and isothermal at temperatures $T_{(hot)}$ and $T_{(cold)}$ then

$$\text{C.O.P. (refrigerator)} = T_{(cold)}/\{T_{(hot)} - T_{(cold)}\}$$

$$\text{C.O.P. (heat pump)} = \Phi(out))/P(in)$$

Again for reversible isothermal heat transfers this reduces to

$$\text{C.O.P. (heat pump)} = T_{(hot)}/\{T_{(hot)} - T_{(cold)}\}$$

This is the inverse of the Carnot efficiency expression for heat engines.

$$\text{C.O.P. (heat pump)} = \{\Phi(in) + P(in)\}/P(in)$$

$$\text{C.O.P. (heat pump)} = \Phi(in)/P(in) + 1$$

$$\text{C.O.P. (heat pump)} = \text{C.O.P. (refrigerator)} + 1$$

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WORKED EXAMPLE No.1

A heat pump uses a vapour compression cycle with refrigerant 12. The compressor is driven by a heat engine with a thermal efficiency of 40%. Heat removed from the engine in the cooling system is recovered. This amounts to 40% of the energy supplied in the fuel.

The heat pump cycle uses an ideal cycle with an evaporator at 5°C and a condenser at 12.19 bar. The vapour is dry saturated at inlet to the compressor. The condenser produces liquid at 45°C.

Calculate the thermal advantage (Coefficient of Performance) for the heat pump. Compare it with a boiler running at 90% thermal efficiency.

The plant is to deliver 40 kW of heat. Determine the mass flow rate of refrigerant.

SOLUTION

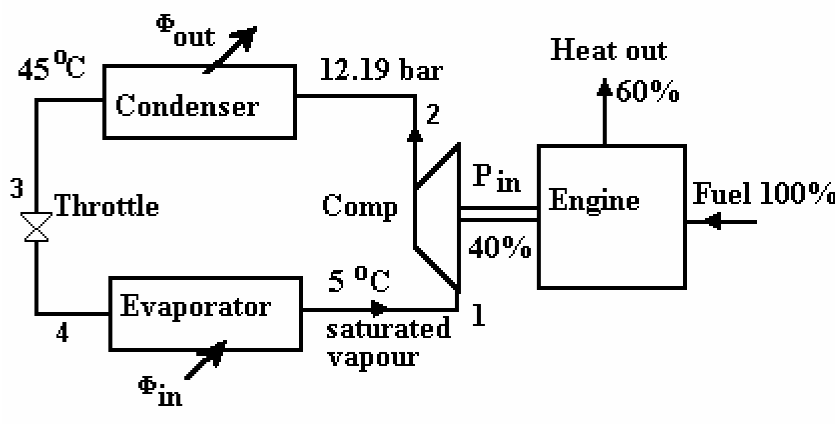

Figure 8

From the R12 tables we find

$h_1 = h_g$ @ 5°C = 189.66 kJ/kg

$s_1 = s_g$ @ 5°C = 0.6943 kJ/kg K = s_2 @ 12.19 bar

If a p - h chart was available, h_2 could be found easily. We must use the tables and we can see that 0.6943 occurs between 0 K of superheat and 15 K of superheat. Using linear interpolation we may find the enthalpy as follows.

	0 K	θ	15 K	
s	0.6797	0.6943	0.7166	0.7166 - 0.6797 =0.0369
h	206.45	h_2	218.64	0.6943 - 0.6797 =0.0146
				218.64 -206.45 =12.19

h_2 = 206.45 + (0.0146/0.0369)12.19 = 211.27 kJ/kg

$h_3 = h_4 = h_f$ at 45°C = 79.71 kJ/kg

h_1 =189.66 kJ/kg

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P(in) = $h_2 - h_1$= 21.67 kJ/kg = 40% of fuel energy

Φ(out)= $h_2 - h_3$= 131.56 kJ/kg

Φ(in)= $h_1 - h_4$= 109.95 kJ/kg

C.O.P (condenser). = 131.56/21.67 = 6.07

C.O.P (evaporator). = 109.95/21.67 = 5.07

Φ(out)= 40% x 10.43 = 417.2% of fuel power

Total heat from system = 242.8 % + 40% = 282.8 % of fuel energy.

Compared to a boiler which gives 90% this is 192.8 % more.

Total heat output = 40 kW = 282.8 % of fuel power.

Fuel Power = 14.14 kW

40 kW = Φ(out) + energy recovered from cooling water

40 = $m_r(h_2 - h_3)$ + 40% x 14.14 = m_r(131.56) + 5.656

m_r=0.261 kg/s

Φ(in) = $m_r(h_1 - h_4)$= 28.7 kW

Φ(out) = $m_r(h_2 - h_3)$ = 34.3 kW

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6. ISENTROPIC EFFICIENCY

When the compression is not reversible and isentropic then the isentropic efficiency is used in the usual way for a compression process.

η_{is} = ideal enthalpy change/actual enthalpy change

WORKED EXAMPLE No.2

The power input to the compressor of an ammonia vapour compression plant is 8.2 kW. The mechanical efficiency is 85%. The ammonia is dry saturated at -6°C at inlet to the compressor. After compression the vapour is at 11.67 bar. The compression has an isentropic efficiency of 90%. The condenser produces saturated liquid.

Calculate the following.

i. The flow rate.
ii. The coefficient of performance for the refrigerator.
iii. The coefficient of performance for the heat pump.

SOLUTION

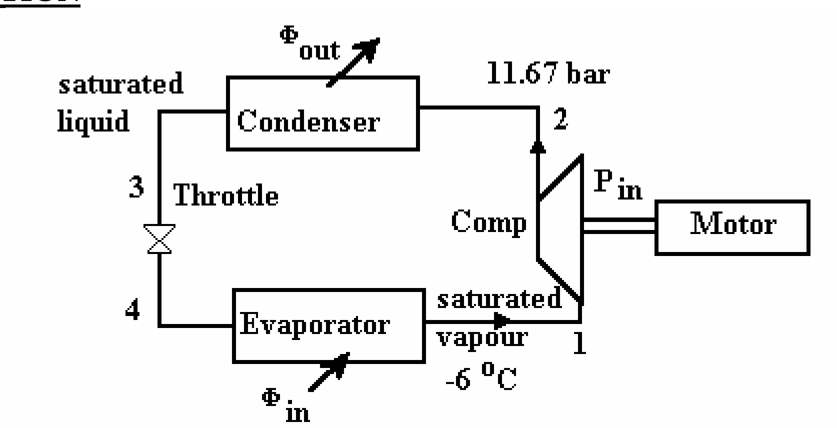

Figure 9

$h_1 = h_f$ at -6°C =1437.6 kJ/kg $h_3 = h_4 = h_f$ @ 11.67 bar = 323.1 kJ/kg
$s_1 = s_f$ at -6°C = 5.419 kJ/kg K Ideally $s_2 = s_1 =$ 5.419 kJ/kg K

From the tables at 11.67 bar we see that the specific entropy is 5.417 kJ/kg K when there is 50 K of superheat. This is the ideal condition after compression and the corresponding enthalpy is 1610.5 kJ/kg K.

Ideal change in enthalpy = 1610.5 - 1437.6 = 172.9 kJ/kg.
Actual change = 172.9/90% = 192.1 kJ/kg.
Actual value of h_2 = 1437.6 + 192.1 = 1629.7 kJ/kg.
Power input to cycle=8.2 kW x 85% =6.97 kW $=m_r$(1629.7 - 1437.6)
m_r= 6.97/192.1 = 0.0363 kg/s
Heat input to evaporator = $m_r(h_1 - h_4)$ = 40.44 kW
Coefficient of performance (refrigerator) = 40.44/6.97 = 5.8
Heat output at condenser = $m_r(h_2 - h_3)$ = 470 kW
Coefficient of performance (heat pump) = 47.4/6.97 = 6.8

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SELF ASSESSMENT EXERCISE No.1

1. A simple vapour compression refrigerator comprises an evaporator, compressor, condenser and throttle. The condition at the 4 points in the cycle are as shown.

Point	Pressure	Temperature
After evaporator	0.8071 bar	-20°C
After compressor	5.673 bar	50°C
After condenser	5.673 bar	15°C
After throttle	0.8071 bar	-35°C

The refrigerant is R12 which flows at 0.05 kg/s. The power input to the compressor is 2 kW. Compression is reversible and adiabatic.

Calculate the following.

i. The theoretical power input to the compressor. (1.815 kW)
ii. The heat transfer to the evaporator. (6.517 kW)
iii. The coefficient of performance based answer (i.)(3.59)
iv. The mechanical efficiency of the compressor. (90.7%)
v. The coefficient of performance based on the true power input. (3.26)

Is the compression process isentropic?

2. A vapour compression cycle uses R12. the vapour is saturated at -20°C at entry to the compressor. At exit from the compressor it is at 10.84 bar and 75°C. The condenser produces saturated liquid at 10.84 bar. The liquid is throttled, evaporated and returned to the compressor.

Sketch the circuit and show the cycle on a p-h diagram.

Calculate the coefficient of performance of the refrigerator. (2.0)

Calculate the isentropic efficiency of the compressor. (71%)

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7 MODIFIED CYCLES

An improvement to the basic compression cycle is the use of a flash chamber instead of a throttle valve. The condensed high pressure liquid at point 7 is sprayed into the low pressure flash chamber. The drop in pressure has the same effect as throttling and the liquid partially evaporates and drops in temperature. The dry saturated vapour is drawn into the compressor and the saturated liquid is pumped to the evaporator. The principal difference is that the evaporator now operates at a higher pressure and so the liquid at point 2 is below the saturation temperature.

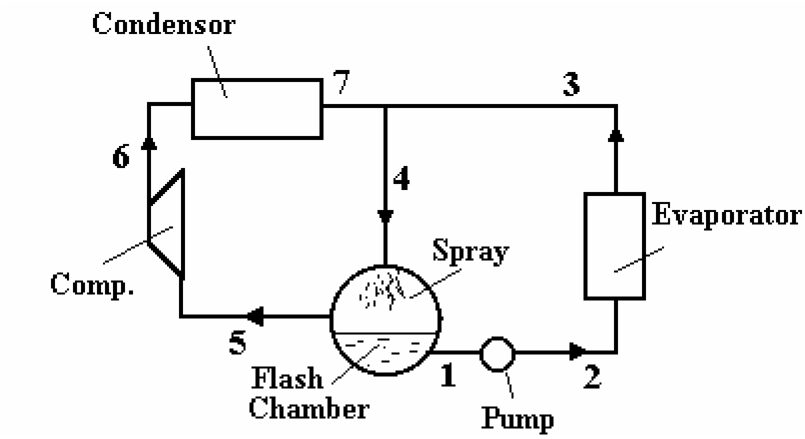

Figure 10

Further modifications may be made by compressing the vapour in two stages and mixing the vapour from the flash chamber at the inter-stage point. The output of the evaporator then goes to the input of the low pressure stage as shown in fig.10

These modifications require more hardware than the basic cycle so the extra cost must be justified by savings and increased capacity to refrigerate.

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WORKED EXAMPLE No.3

A refrigeration plant uses R12 in the cycle below. The evaporator temperature is -50°C and the condenser pressure is 50°C. The flash chamber is maintained at 0°C. Saturated vapour from the chamber is mixed with the compressed vapour at the inter-stage point (3). The liquid in the chamber is further throttled to -50°C in the evaporator. The vapour leaving the evaporator is dry saturated and compression is isentropic. Find the coefficient of performance for the refrigerator assuming 1 kg/s flow rate.

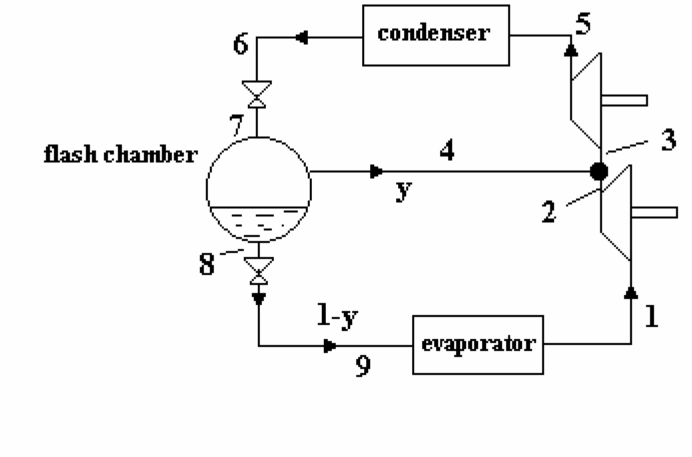

Figure 11

SOLUTION

At point (1) the vapour is dry saturated so $h_1 = h_g$ @ -50°C = 164.95 kJ/kg
Similarly $s_1 = s_g$ = 0.7401 kJ/kg K

The pressure at point 2/3 must be ps at 0°C = 3.08 bar

$s_1 = s_2$ = 0.7401 kJ/kg K @ 3.08 bar.

From the tables this is seen to be superheated so the degree of superheat and the enthalpy must be estimated by interpolation as follows.

$\Delta T/T = (0.7401 - 0.7311)/(0.7641 - 0.7311)$

$\Delta T = 4.09$ K

Now the enthalpy at point 2 may be estimated as follows.

$4.09/15 = (h_2 - 197.25)/(.7641 - .7311)$

h_2 = 199.9 kJ/kg

Energy balance on Flash Chamber

Assume saturated liquid at point (6). $h_6 = h_f$ @ 50°C = 84.94 kJ/kg
Since the enthalpy is the same after throttling then h_7 = 84.94 kJ/k

h_4 = hg at 0°C =187.53 kJ/kg

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Let the flow rate be 1 kg/s at (7) and (y) at point (4). Balancing enthalpy we have

$h_7 = y(h_4) + (1-y)h_8$

$h_8 = h_f$ @ 0^oC = 36.05 kJ/kg K

$y(187.53) + (1-y)(36.05)$

$y = 0.3227$ kg/s

Energy balance at mixing point.

$(1-y)h_1 + yh_4 = h_3$

$((1-0.3227)(199.9) + (0.3227)(187.53) = h_3 = 195.9$ kJ/kg

Power of LP Turbine

$P_1 = (1-y)$ kg/s $(h_2 - h_3) = (1-0.3227)(199.9 - 164.95) = 23.67$ kW

The vapour at point 3 is clearly superheated between 0 and 15 K. Interpolation gives the degree of superheat as follows.

$\Delta T/15 = (195.908-187.53)/(197.25-187.53)$

$\Delta T = 12.93$ K

We may now interpolate to find s_3

$12.93/15 = (s_3 - 0.6966)/(0.7311 - 0.6966)$

$s_3 = 0.7263$ kJ/kg K $= s_5$ @ 12.19 bar (p_s at 50^oC)

Again we must interpolate to find the degree of superheat at point 5 which is between 0 and 15 K again.

$\Delta T/15 = (0.7263 - 0.7166)/(0.7503-0.7166)$

$\Delta T = 4.33$ K

We may now interpolate with this degree of superheat to find h_5.

$4.33/15 = (h_5 - 218.64)/(230.33 - 218.64)$

$h_5 = 222$ kJ/kg

HP Turbine Power

$P_2 = 1$ kg/s$(h_5 - h_3) = (222-195) = 26.92$ kW

Total Power input = 50.59 kW

Evaporator Heat Input

$h_9 = h_8 = hf$ at 0^oC = 36.05 kJ/kg

$\Phi(in) = 0.6773$ kg/s $(h_1 - h_9)$

$\Phi(in) = 0.6773(164.95 - 36.05) = 87.3$ kW

Coefficient of Performance = 87.3/26.92 =1.73

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SELF ASSESSMENT EXERCISE No.2

1. A refrigerator operates with ammonia. The plant circuit is shown below. The conditions at the relevant points of the cycle are as follows.

1	saturated liquid at -30°C
3,4 and 7	saturated liquid at 10°C
5	saturated vapour at -30°C

The pump and compressor have an isentropic efficiency of 80%. there are no heat losses. The specific volume of ammonia liquid is 0.0015 m3/kg.

Determine the coefficient of performance and the mass flow rate if the refrigeration effect is 10 kW.

(Ans. 3.964 and 0.0635 kg/s)

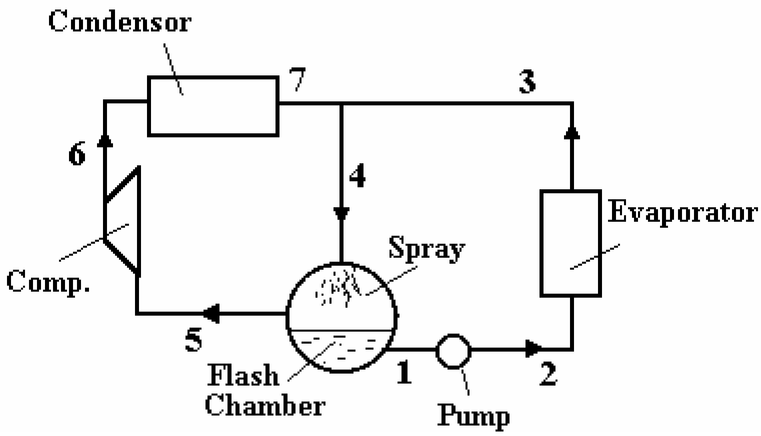

Figure 12

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2. A heat pump consists of a compressor, condenser, throttle, and evaporator. The refrigerant is R12. The refrigerant is at 0^oC at entry to the compressor and 80^oC at exit. The condenser produces saturated liquid at 50^oC. The throttle produces wet vapour at -10^oC. The mass flow rate is 0.02 kg/s. The indicated power to the compressor is 1 kW.

   Sketch the T - s diagram and p - h diagram for the cycle.

   Calculate the coefficient of performance for the heat pump
   (2.9 based on I.P.)

   Calculate the rate of heat loss from the compressor.
   (0.2 kW)

   Calculate the coefficient of performance again for when the refrigerant is sub cooled to 45^oC at exit from the condenser.
   (3 based on I.P.)

   Calculate the temperature at exit from the compressor if the compression is reversible and adiabatic.
   (68.7^oC)

3. A refrigeration cycle uses R12. The evaporator pressure is 1.826 bar and the condenser pressure is 10.84 bar. There is 5K of superheat at inlet to the compressor. The compressor has an isentropic efficiency of 90%. the condensed liquid is under-cooled by 5K and is throttled back to the evaporator.

   Sketch the cycle on a T-s and p-h diagram.

   Calculate the coefficient of performance. (3.04)

   Explain why throttles are used rather than an expansion engine.

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8. RECIPROCATING COMPRESSORS

This is covered in detail in tutorial 2. The knowledge of compressors is often required in refrigeration and heat pump studies so the basics are covered here along with example questions on compressors used in this area. The diagram shows the basic design of a reciprocating compressor. The piston reciprocates drawing in gas, compressing it and expelling it when the pressure inside the cylinder reaches the same level as the pressure in the delivery pipe.

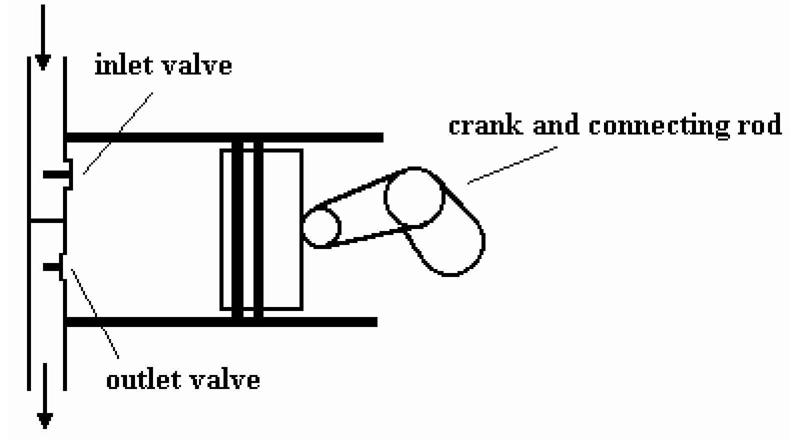

Figure 13

If the piston expels all the air and there is no restriction at the valves, the pressure - volume cycle is as shown below.

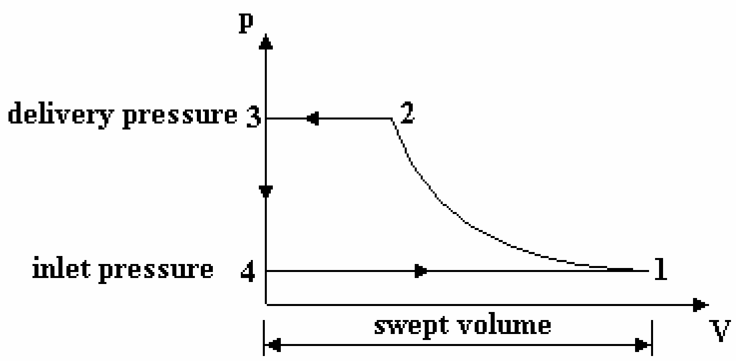

Figure 14

Gas is induced from 4 to 1 at the inlet pressure. It is then trapped inside the cylinder and compressed according the law $pV^n = C$. At point 2 the pressure reaches the same level as that in the delivery pipe and the outlet valve pops open. Air is then expelled at the delivery pressure. The delivery pressure might rise very slightly during expulsion if the gas is being compacted into a fixed storage volume. This is how pressure builds up from switch on.

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In reality, the piston cannot expel all the gas and a clearance volume is needed between the piston and the cylinder head. This means that a small volume of compressed gas is trapped in the cylinder at point 3. When the piston moves away from the cylinder head, the compressed gas expands by the law $pV^n = C$ until the pressure falls to the level of the inlet pressure. At point 4 the inlet valve opens and gas is drawn in. The volume drawn in from 4 to 1 is smaller than the swept volume because of this expansion.

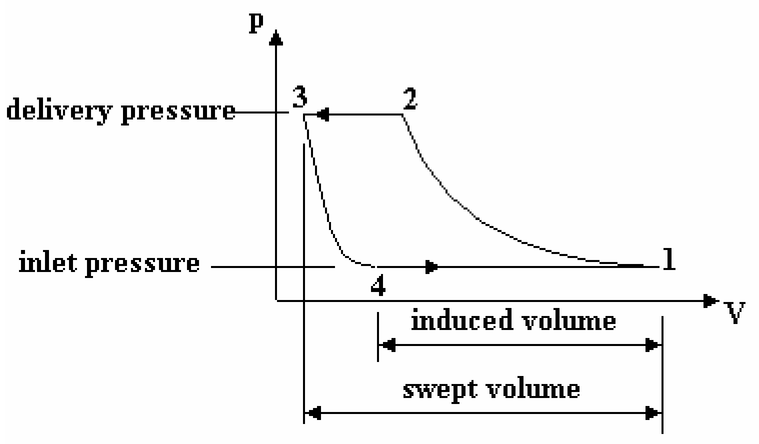

Figure 15

The volumetric efficiency is defined as

$$\eta_{vol} = \textbf{Induced Volume/Swept volume.}$$

This efficiency is made worse if leaks occur past the valves or piston.

In real compressors, the gas is restricted by the valves and the valves tend to move so the real cycle looks more like this.

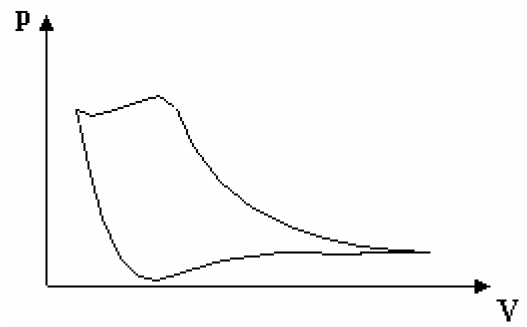

Figure 16

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WORKED EXAMPLE No.4

Dry saturated Refrigerant 12 vapour at 5°C is compressed in a reciprocating compressor to 12.19 bar at a rate of 0.274 kg/s. The clearance volume is 5% of the swept volume. The expansion part of the cycle follows the law $pV^{1.2}$ =C. The crank speed is 360 rev/min. Calculate the swept volume and the volumetric efficiency.

SOLUTION

Swept Volume = V Clearance volume = 0.05 V
Consider the expansion from 3 to 4 on the p-V diagram.
The inlet pressure must be p_s at 5°C hence p_4 = 3.626 bar.

$p_3V_3^{1.2} = p_4V_4^{1.2}$

$12.196(0.05V)^{1.2} = 3.626 (V_4^{1.2})$
$V_4 = 0.137V$ or 13.7% of V

$V_1 = V + 0.05V = 1.05V$
Induced volume = $V_1 - V_4 = 1.05V - 0.137V = 0.913V$

m_r = 0.274 kg/s

At inlet $v = v_g$ at 5°C = 0.0475 m³/kg.

Volume flow rate required = 0.274 x 0.0475 = 0.013 m³/s.

Induced volume = 0.013 = 0.913V
V = 0.013/0.913 = 0.0142 m³/s

Crank speed = 6 rev/s so the swept volume = 0.0142/6 = 0.00237 m³.

η_{vol} = Induced Volume/Swept volume.

η_{vol} = 0.913V/V = 91.3%

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SELF ASSESSMENT EXERCISE No. 3

1. Why is it preferable that vapour entering a compressor superheated?

A vapour compression refrigerator uses R12. The vapour is evaporated at -10^oC and condensed at 30^oC. The vapour has 15 K of superheat at entry to the compressor. Compression is isentropic. The condenser produces saturated liquid.

The compressor is a reciprocating type with double action. The bore is 250 mm and the stroke is 300 mm. The speed is 200 rev/min. The volumetric efficiency is 85%. You may treat superheated vapour as a perfect gas. Determine

i. the mass flow rate (0.956 kg/s)
ii. the coefficient of performance. (5.51)
iii. the refrigeration effect. (122.7 kW)

(Note that double acting means it pumps twice for each revolution. The molecular mass for R12 is given in the tables.)

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9. AMMONIA ABSORPTION CYCLE

The basic principle of the ammonia absorption cycle is similar to that of the vapour compression cycle. An evaporator is used to absorb heat at a low temperature and a condenser is used to reject the heat at a higher temperature. The difference is in the way the ammonia is passed from the evaporator to the condenser. In a compression cycle this is done with a compressor. In the absorption cycle it is done by absorbing the ammonia into water at the lower temperature. The water and ammonia is then pumped to a heater raising the pressure and temperature. The heater also separates the ammonia from the water and the ammonia vapour is driven off is at a higher pressure and temperature than it started at. The vapour is then condensed and throttled back to the evaporator.

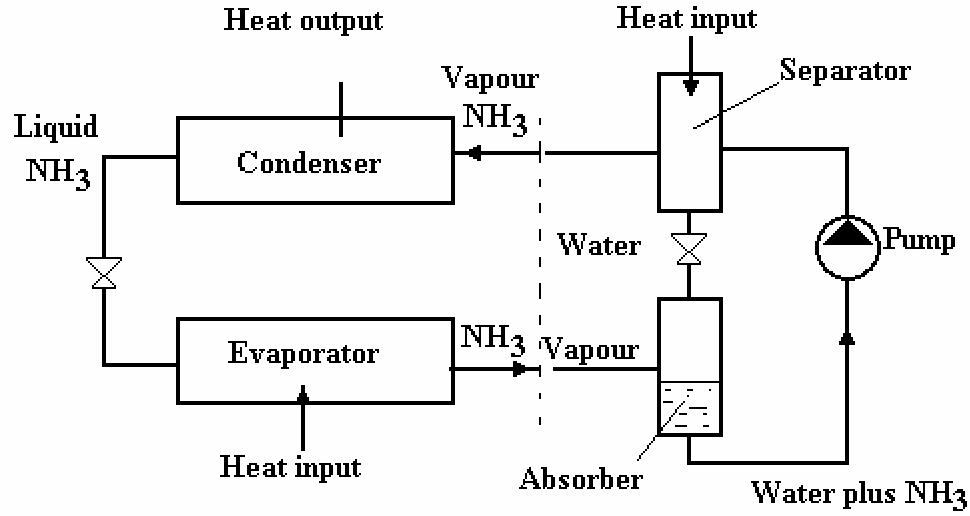

Figure 17

The advantage of this system is that a water pump replaces the vapour compressor. The pump may be done away with altogether by making use of the principle of partial pressures. When hydrogen is mixed with ammonia vapour, the total pressure of the mixture 'p' is the sum of the partial pressures such that

$$p = p(\text{ammonia}) + p(\text{hydrogen})$$

The mixing is done in the evaporator but the total pressure stays the same. The ammonia vapour hence experiences a drop in pressure when mixing occurs and the effect is the same as throttling so a throttle valve is not needed either. Mixing causes the vapour to cool and condense so that a cold wet vapour results.

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Evaporation dries it out and a mixture of ammonia and hydrogen gases leaves the evaporator. The mixture goes to the absorber where the ammonia is absorbed into the water leaving the hydrogen behind. The hydrogen goes back to the evaporator in a continuous cycle. When the ammonia and hydrogen separate out in the absorber, they both experiences a pressure rise back to p which is also the water pressure.

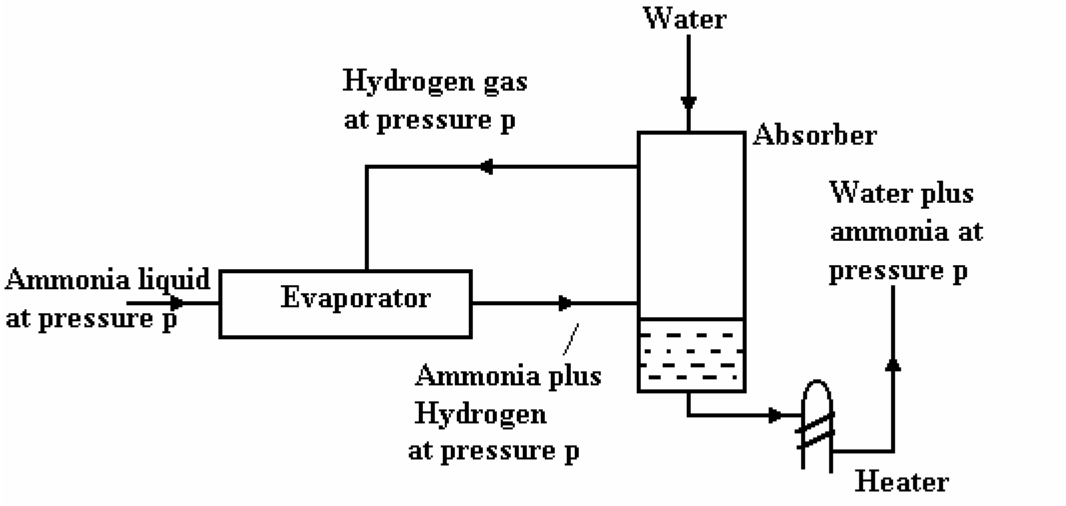

Figure 18

Since no pressure difference exists between the evaporator and condenser, circulation may be caused by a thermo-siphon which is induced by heating the water and ammonia. The p-h cycle is similar to that of a vapour compression cycle. Process 3 to 4 is due to the mixing in the evaporator. Process 1 to 2 is due to the absorption.

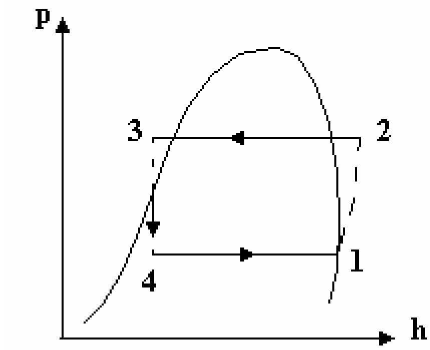

Figure 19

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Heat which is put into the separator in order to make the ammonia leave the water, is carried with the water back to the absorber. This heat can be used to operate the thermo-siphon by use of a heat exchanger. A schematic of a complete plant is shown below. The heat required to operate the system may be obtained from anywhere and is commonly a gas flame (Electrolux refrigeration system). This system is popular in caravan refrigerators.

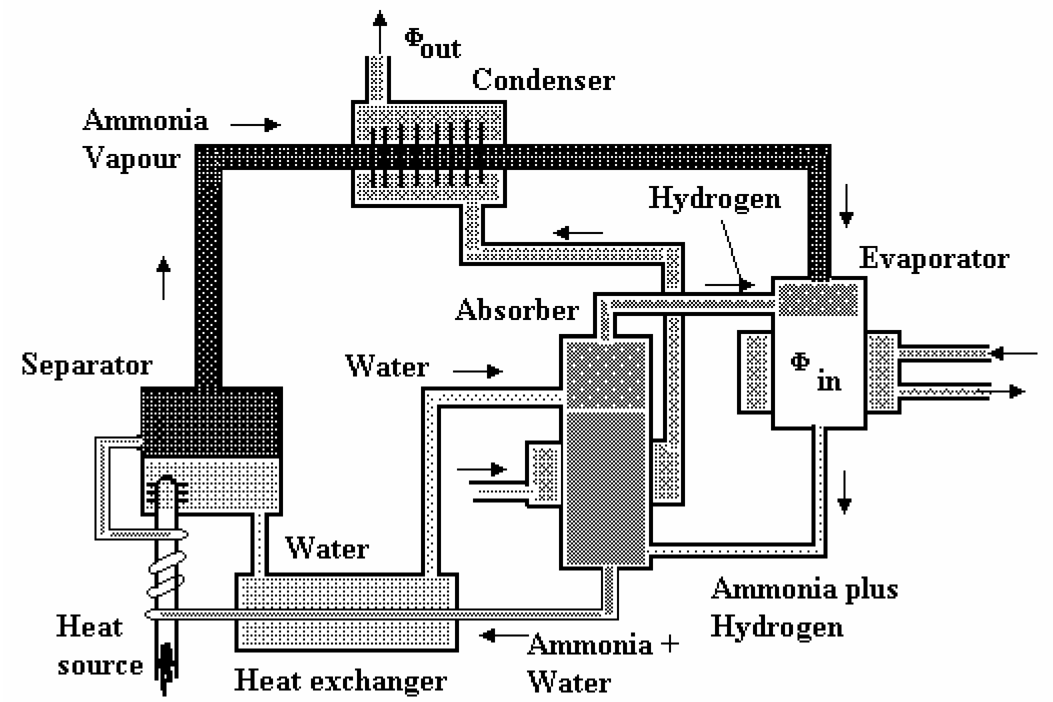

Figure 20

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