Refrigeration Cycle Experiment

Abstract:

The aim of this experiment is to construct the Vapour Compression Refrigeration Cycle on the Pressure-Enthalpy property diagram, and determine the Coefficient of Performance by Measuring the temperatures and pressures of the liquid and gas phases of the refrigerant at various locations. Also to identify the components of the mechanical refrigeration apparatus and the thermodynamic processes occurring in these components and analyse these processes in condenser, evaporator, compressor and throttle/expansion valve

Methodology:

First the cooling water supply and mains supply was switched on to the unit. After that, the valves have been checked due Normal Operation which allowed vapour to be drawn from the evaporator by the compressor and for condensed liquid to return to the evaporator from the condenser. The third step was to let the water supply to the unit and to set the condenser cooling water flow-rate about 6 g/s. Then, the evaporator water flow has been sited to approximately 10 g/s. After that, the main switch of the compressor has been turned on in order to supply energy. At that point, the refrigerant flow rate has been adjusted about 1 g/s. the unit has been allowed to run for approximately 10-15 minutes to stabilise for the surrounding environment. From the data, the vapour-compression cycle (the large P-h diagram) can be plotted.

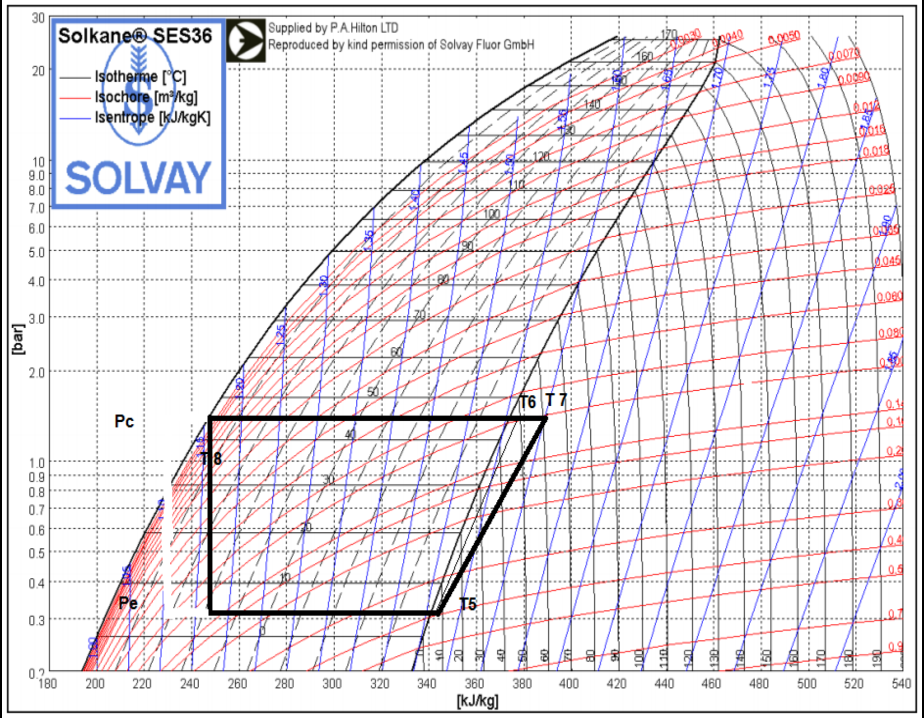

Figure 1-pressure-enthalpy diagram for refrigerant solkane SES36

Theory and Analysis:

1. The refrigeration cycle has been plotted and it is dependent on the temperatures (t_5, t_6, t_7, t_8) and the absolute pressure of the evaporator and condenser.
2. The temperatures are given.
3. The absolute pressure has been calculated from the evaporator and condenser gauge relatively to the local atmospheric pressure.
   Absolute pressure of condenser: $P_{abs} = P_{atms} + P_{gauge}$=1.026+0.45=1.476 bar
   Absolute pressure of evaporator: $P_{abs} = P_{atms}$-P_{vac}=1.026-0.7=0.326 bar
4. The pressures had been converted to bars according to the graph given.
5. two horizontal lines has been drawn which match the size of the plot and also the state of each process.
6. Each temperature has been pointed in its own location and also to find the specific enthalpy.
7. The rate of heat transfer from water to evaporator $\dot{Q}_e$= $\dot{m}_e C_P(t_2 - t_1) = 7.75 \times 10^{-3} \frac{j}{s}\, or\ w.$
   Direct substitution from the table
8. The rate of heat transfer from water to condenser $\dot{Q}_c$= $\dot{m}_c C_P(t_3 - t_4) = 2.97 \times 10^{-4} \frac{j}{s}\, or\ w.$
   Direct substitution from the table
9. Delivered pressure ratio= $\frac{Absolute\ pressure\ of\ condenser}{Absolute\ pressure\ of\ evaporator}$=$\frac{1.476}{0.326} = 4.52$
10. The coefficient of performance can be found through the temperatures (t_5, t_7, t_8) which the specific enthalpy can be found from.
11. $t_5 = h_5 = h_1\ ,\ t_7 = h_7 = h_2\ ,\ t_8 = h_8 = h_4 \therefore COP_{ref} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{345-233}{390-345} = 2.5$
12. Isentropic efficiency can be determined from pointing a point from point (t_1) which is parallel to the blue line (isentrope) it could be impossible but it can be approximated. This efficiency describes the competence of the isentropic work which is the difference between the predicted point and point 1 over the actual work done by the compressor.
13. Isentropic efficiency =$\frac{h_{7'} - h_5}{h_7 - h_5}$=$\frac{371-345}{390-345} = 0.57\ ,\ h_7 = h_2, h_5 = h_1$
14. Power input to the refrigerant: $P_{comp} = \dot{m}_{ref}(h_2 - h_1)$ = 45 , direct substitution from the table and graph.

By using the pressure-enthalpy diagram shown for the refrigerant the values for (h) can be obtained, and the coefficient of performance of the refrigerant (COP_{ref}) can therefore be calculated to be 2.5. In general it is said that the reverse Carnot cycle is the most efficient refrigeration cycle operating between lower temperature zone, and higher temperature zone. However it is not a suitable model for refrigeration cycles since processes 2-3 and 4-1 are not practical. In process 2-3 the condenser must be able to handle refrigerants in two phases (gas and liquid). And process 4-1 involves the expansion of a high moisture-content refrigerant in a turbine.also, it appear that The refrigerant evaporates at a lower temperature and low pressure drawing in heat,The refrigerant condenses at a higher temperature and higher pressure radiating heat,The cycle is not the most practical because it involves condensing refrigerants of 2 phases in one component, It also involves expansion of the refrigerant in a turbine which can lead to damage.

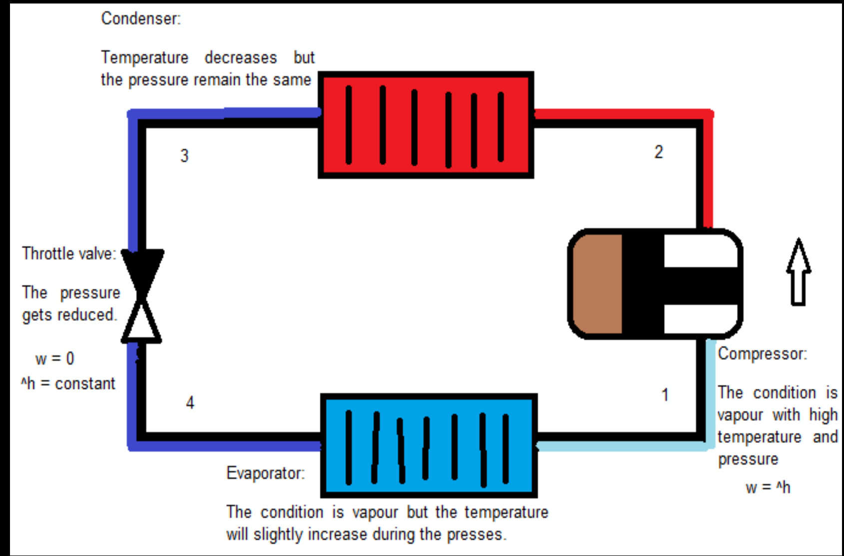

Figure 2- the main four components in the system

Table 1 – investigation of system performance

Data	value
Evaporator Gauge Pressure, Peg (kN/m2)	-70
Evaporator Absolute Pressure, Pea (bar)	0.326
Evaporator Inlet Water Temperature, t1 (°C)	23.3
Evaporator Outlet Water Temperature, t2 (°C)	20.2
Evaporator Refrigerant Temperature, t5 (°C)	8.5
Evaporator water flow-rate, me (g/s)	10
Condenser Gauge Pressure, Pcg (kN/m2)	45
Condenser Absolute Pressure, Pca (par)	1.476
Condenser Inlet Water Temperature, t4 (°C)	23.7
Condenser Outlet Water Temperature, t3 (°C)	30.8
Condenser Refrigerant Temperature t6 (°C)	38.7
Condenser water flow-rate, mc (g/s)	6
Compressor Discharge Temp, t7 (°C)	59.7
Compressor Power Input, Pmech (Watts)	170
Refrigerant mass flowrate mref (g/s)	1.0
Throttle inlet temperature t8 (°C)	31.5
Rate of Heat Transfer from Water Qe (w)	7.7*10^-3
Rate of Heat Transfer from Water Qc (w)	2.9*10^-4
Delivered Pressure Ratio	4.52
Coefficient of Performance (COP)	2.5
Isentropic efficiency of compressor	0.57
Power input to the refrigerant (W)	45

Reference:

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