THERMAL SYSTEMS DESIGN PROJECT

Optimized design of a vapor compression refrigeration system

Problem statement

Vapor compression refrigeration systems are used to remove heat from a low temperature (T_L) environment and reject it to a high-temperature (T_H) reservoir (typically ambient air). They are generally designed to operate optimally at a particular maximum cooling load condition, which is termed as the 'design point' of the system. The schematic of a simple vapor compression refrigeration system is shown in Fig. 1. Based on the constitutive relationships between pressure, temperature, enthalpy, etc., the system shown in Fig. 1 has four degrees of freedom. The two pressures in the system (P_1 and P_2), the level of superheat at the compressor outlet (T_2) and the mass flow rate of the refrigerant (R-12 in this case) uniquely define the remaining thermodynamic states at each of the transition points (denoted 1,2,3,4 in Fig. 1) given the assumptions for an ideal vapor compression refrigeration system are valid. An optimized combination of these variables will result in an efficient system operating at the design point.

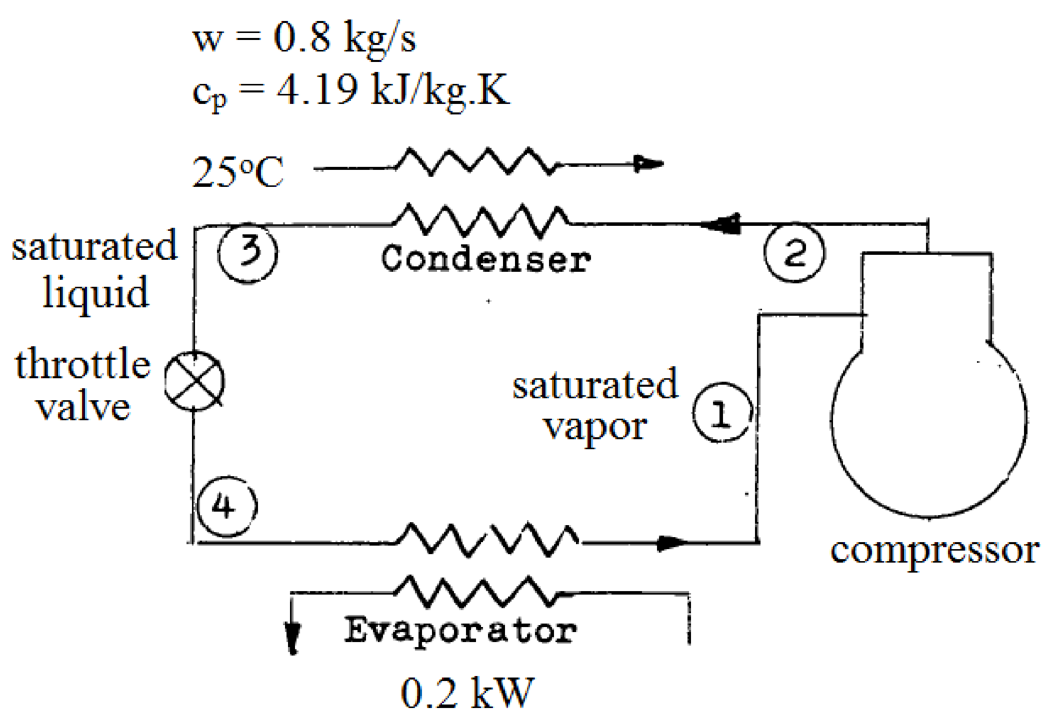

Figure 1. Schematic of the proposed refrigeration system

For the refrigeration system shown, the following data is given for the evaporator and condenser:

UA = 0.05 kW/K (evaporator), UA = 0.05 kW/K (condenser)

The condenser is water cooled, water enters at 25°C with a flow rate of 0.8 kg/s. The refrigerant, R-12 is saturated liquid at point 3. Consider specific heat of water and R-12 to be 4.19 kJ/kg.K and 0.9 kJ/kg.K respectively.

The design cooling load at the evaporator is 0.2 kW $[=\dot{m}(h_4 - h_1)]$. The refrigerant is saturated vapor at point 1.

The compressor has adjustable capacity which is regulated to provide the specified refrigeration rate in the evaporator.

The T-s and p-h diagrams for the vapor compression refrigeration cycle shown in Figure 1 are given below:

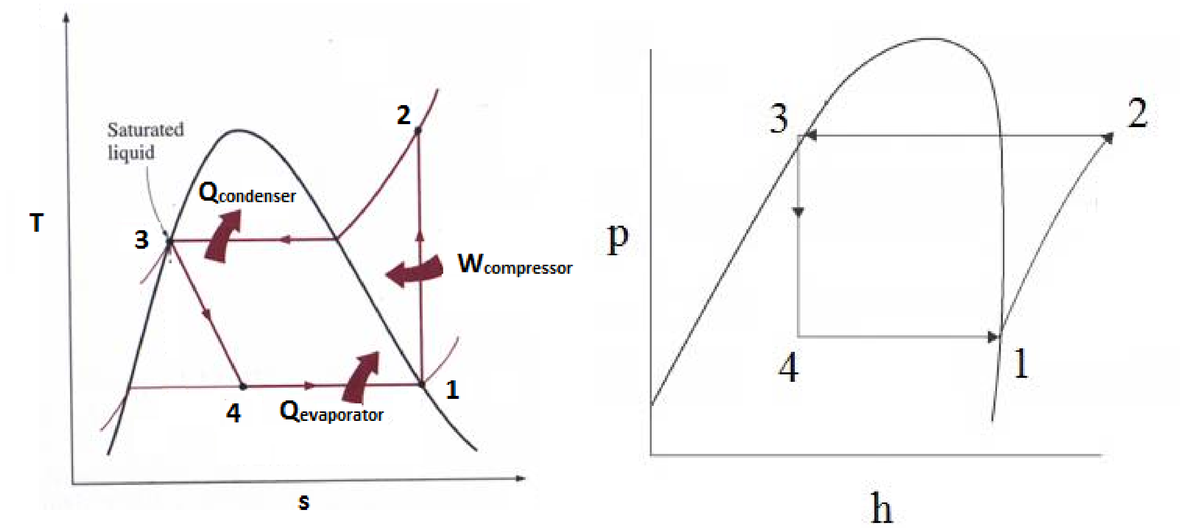

Figure 2. The T-s and p-h diagrams for the cycle shown in Figure1

For saturated pressure-temperature relation for R-12 is as follows –

$$\ln p = 14.861 - 2498.3/(t + 273)$$

Where, p =pressure (kPa)
t = temperature in °C

For enthalpy of saturated liquid:

$$h_f = 200 + 0.925t + 0.00081t^2$$

Where, h_f is the enthalpy of saturated liquid, kJ/kg *t* is temperature in ^oC

For enthalpy of saturated vapor

$$h_g = 351.1 + 0.428t - 0.00071t^2$$

Where, h_g is the enthalpy of saturated vapor, kJ/kg
t is temperature in ^oC

The specific work of compression for compressing R-12 from pressure p_1 to p_2 can be expressed as –

$$\Delta h = h_2 - h_1 = 188(1 - \frac{14.861 - \ln p_2}{14.861 - \ln p_1})$$

Where, Δh = specific work of compression, kJ/kg
p_2 is discharge pressure, kPa
p_1 is suction pressure, kPa

Assignment:
Determine the mass flow rate of refrigerant R-12 along with the other unknowns in the problem that results in minimum power requirements at the compressor.

HINT:
In order to proceed, obtain first, the mathematical statement of the objective function in the form:

minimize $y = f(x_1, x_2, \ldots x_n)$

where y is the work of compression (in this problem), and x_1, x_2, …..x_n represent the adjustable parameters

Then using conservation principles, rate equations and other data given above, develop a set of constraint equations set in the form:

$\phi_1(x_1, x_2, \ldots x_n) = 0$
$\phi_2(x_1, x_2, \ldots x_n) = 0$
……………
$\phi_m(x_1, x_2, \ldots x_n) = 0$
where $m < n$

Use a suitable optimization scheme to design y^*, x_1^*, x_2^*, ……… x_n^*