THERMODYNAMIC ANALYSIS OF R134A – DMAC VAPOR ABSORPTION REFRIGERATION (VAR) SYSTEM

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Abstract
This study primarily focuses on the thermodynamic analysis of single stage vapor absorption refrigeration system using R134a – DMAC solution as the working fluid. Variations in the performance parameters of the system are studied against various operating temperatures of generator and absorber. The result of this theoretical study show that coefficient of performance (COP) value can be improved by elevating generator temperature up to certain level and lowering absorber temperature. At such elevated generator temperature, value of circulation ratio (CR) is lowered. The scope of this study is limited to the system with 1kW evaporator capacity and effectiveness of solution heat exchanger (SHX) as 0.8. For the proposed condition of source and sink temperature 120˚C and 40˚C respectively the maximum value of COP was found to be 0.41 and corresponding CR value as 3.90.

Keywords: Thermodynamics analysis, absorption refrigeration, generator temperature, absorber temperature, R134a–DMAC

Nomenclature

Symbols

COP	Coefficient of Performance
CR	Circulation Ratio
DMAC	Dimethyl Acetamide
h	Enthalpy (kJ/kg)
m	Mass flow rate (kg/s)
P	Pressure (bar)
Q	Heat duty (kW)
R	Universal Gas Constant (8.3145 J/molK)
T	Temperature (˚C)
v	Specific volume (m^3/kg)
V	Volume (m^3)
W	Work supplied (kW)
X	Mass fraction (kg/kg)

Subscripts

a	Absorber
c	Condenser
d	DMAC
e	Evaporator
f	Fluid
g	Generator
p	Pump
ref	Refrigerant
s	Strong solution
sol	Solution
w	Weak solution
E	Excess
SHX	Solution heat exchanger

Greek letters

ε	Effectiveness of SHX
ρ	Density of solution (kg/m^3)

Introduction
Vapor compression refrigeration is the most commonly used commercial system for refrigeration wherein, a compressor is used to compress refrigerant. Compression utilizes high-grade energy as electricity. Alternatively, cooling may be achieved by means of an absorption refrigeration system, wherein the compressor of the compression system is replaced by a generator and pump combination. A low-grade heat source such as solar energy, industrial waste heat is supplied to generator and the system only needs less pumping power compared to a mechanical compression system. Most of the heat produced during industrial process to produce steam or heat is ejected to surrounding as waste after completion of process. This waste heat can be converted to useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle [1]. The thermal driven absorption system is noticed as the alternative to the vapor compression system which may cause environmental problems such as global warming and ozone layer depletion [2].

The most common binary fluid for absorption system is H_2O/LiBr and NH_3/H_2O. Lithium Bromide and water is mostly popular but the evaporator temperature is limited to minimum of 5˚C. Ammonia water combination can be used for evaporator temperature below 0˚C but this system exhibits a relatively lower COP, and therefore efforts are being made to search for better refrigerant-absorbent pairs that can improve system performance. The search for new working fluids has centered on the halo hydrocarbon group of fluids commercially known as Freon [3]. A comparison of R134a, R22 and R124 fluorocarbon refrigerants with organic absorbents has been carried out in single- and double-stage absorption heat pumps [4]. These fluids meet most of the requirements for the desirable refrigerant. In a comparative experimental study of performance of R134a and R22 based vapor absorption refrigeration systems was performed by Songara et al. [5] who found that DMAC as a very promising solvent for R22 in VARS applications, and suggested further study on performance of R134a – DMAC solution in VARS. Some researchers have considered and reported data for R134a – DMAC as the refrigerant – absorbent pair for VARS and proved its feasibility in the absorption system. R-134a – DMAC working fluid pair is having zero ozone depletion potential and negligible global warming potential (0.25) with a comparatively lower heat source temperature [6].

Arivalagan et al. [7] performed a simulation studies on R134a – DMAC based half effect absorption refrigeration and found COP of the cycle to be 0.4, with an evaporator temperature of 5˚C, generator temperature of 70˚C and condenser temperature of 25˚C. This system is mostly applicable in industries, where low temperature waste heat (70˚C) is available. Muthu [8] performed experimental studies on performance of R134a – DMAC based absorption cooling system using low potential thermal sources. The study proved that R134a – DMAC based absorption cooling system yielded an optimum COP of 0.4, when the heat source temperature is 70˚C. Crepinsek et al. [9] performed a simulation studies on a half effect vapor absorption refrigeration cycle for solar energy based cold storage system using R134a – DMAC as working fluids showing that the COP of this cycle is about 0.35 - 0.46 for an evaporating temperature of -5 to 5˚C, heat input at 70˚C and a condensing temperature at 25˚C. COP improved up to 13% using a condensate pre-cooler. Muthu et al. [3] performed an experimental study on R134a – DMAC hot water based vapor absorption refrigeration systems of 1kw capacity. They found out the COP of the system to be 0.25 – 0.45 for sink and source temperatures of 30 and 80˚C respectively and a typical heat input of 4kW.

The heat-driven auto-cascade absorption refrigeration cycle has been analyzed by Yijian He and Guangming Chen [10] for low temperature applications. They used mixture of R23 + R32 +R134a/DMF as working pair and its characteristic study is carried out under different operational conditions.

Various studies in R134a – DMAC have been conducted in the past and COP has been identified around 0.2 to 0.4 for typical operating conditions with heat input at 70 to 80˚C, evaporating temperature at -5 to 5˚C and 25 to 30˚C for condenser as well as absorber. For absorption refrigeration system, COP, rich and poor solution concentrations are functions of generator temperature. [11] Further reducing the cooling water temperature from 30 ºC to 25 ºC, the absorber heat load and refrigerant mass absorption rate increases by 16.3% and 15 % respectively [12]. No literature is available on analysis of R134a – DMAC at high generator temperature and also theoretical prediction of maximum COP with respect various operating conditions are not dealt. Hence, this study is focused on the thermodynamic investigations on R134a – DMAC based absorption refrigeration systems for the typical operating conditions with source temperature from 90 to 140˚C, evaporator temperature of 5˚C, absorber temperature from 30˚C to 50˚C along with condenser temperature of 40˚C.

System description

Figure 1 shows the schematic of a simple vapor absorption refrigeration system. The liquid refrigerant from valve V1 (9) evaporates after absorbing Q_e amount of heat in evaporator. The vapor refrigerant from evaporator (10) travels to absorber where it is absorbed by the weak solution (solution containing lower concentration of refrigerant and higher concentration of absorbent) coming from generator after expanding through valve V2 (6) forming a strong solution in Absorber. During absorption process, Q_a amount of heat of mixing rejected from the system. Thus formed strong solution (1) is now pumped to generator (3) using a circulating pump. Before reaching generator, the strong solution (2) absorbs some amount of heat from the weak solution coming from generator in solution heat exchanger (SHX) which is supposed to improve performance of the system. Further, Q_g amount of heat is supplied to the strong solution in generator through external heat source so as to separate refrigerant from the refrigerant-absorbent solution. Vapor refrigerant (7) moves to condenser leaving weak solution in the generator. The weak solution (4) passes through SHX. After losing some heat in SHX the weak solution (5) is allowed to expand through valve V2. Further heat is removed from vapor refrigerant in condenser. The condensed refrigerant (8) then passes through valve V1 which liquefies the refrigerant and is supplied to evaporator for the next cycle.

Correlations

Thermodynamic and thermophysical properties such as bubble point, dew point, vapor pressure, liquid and vapor mass fractions, enthalpy and specific heat are obtained from the literature [6,14,15]. The heat transfer coefficient for R134a – DMAC is in the range of 100 – 400 W/m^2K [13].

The density of R134a – DMAC solution with respect to temperature and concentration at saturation condition is Given by : [6]

$$\rho = \sum_{j=0}^{3}\sum_{i=0}^{3} A_{i,j} X^i T^j \qquad (1)$$

where *A* is a constant.

Enthalpy of R134a – DMAC solution with respect to the enthalpies of individual component and excess enthalpy of solution is expressed as: [6]

Enthalpy of solution is given by

$$h_{sol} = h_{ref} + h_d + h_E \qquad (2)$$

Enthalpy of R134a with respect to temperature at saturated condition is given by

$$h_{ref} = \left(0.0009T^2 + 0.32T + 99.636\right) \times 4.187 \qquad (3)$$

The enthalpy of DMAC with respect to temperature at saturated condition is given by

$$h_d = \left(100 + AT + 0.5BT^2\right) \times 4.187 \qquad (4)$$

Excess Enthalpy of R134a – DMAC solution with respect to temperature and mole fraction of R134a at equilibrium condition is given by

$$h_E = \sum_{j=0}^{7}\sum_{i=0}^{7} C_{i,j} X^i T^j \qquad (5)$$

where *C* is a constant.

The vapor enthalpy of R134a leaving from the generator for a given pressure and temperature is given by following correlation: [14]

$$h_g = h_f + h_{fg} = h_o + h_g \text{ (Refrigerant vapor)} \qquad (6)$$

$$h_f + h_{fg} = h_o + \left[\int \sum_{i=1}^{N} C_{pi}{}^{o} x_i dT + \left(\frac{a}{b} - \frac{T}{b}\frac{da}{dT}\right)\ln\frac{V}{V+b} + RT\left(\frac{pV}{RT} + 1\right)\right] \qquad (7)$$

where h_o is an arbitrary constant, which is calculated at 273K and which is fixed even for varying the temperature and pressure of R134a vapor.

The pressure at vapor-liquid equilibrium with respect to the temperature and concentration of R134a in the solution is given by Borde [6]:

$$P = \sum_{j=0}^{7}\sum_{i=0}^{7} C_{i,j} X^i T^j \qquad (8)$$

where *C* is a constant.

Thermodynamic analysis

Thermodynamic analysis in this study is carried out for 1 kW refrigerating capacity. The temperatures of evaporator and condenser are kept constant at 5°C and 40°C respectively. The mass and energy balance equations for various parts of the system are as follows:

The mass flow rate of refrigerant, weak and strong solution based on steady flow analysis of evaporator is determined as follows:

$$m_{ref} = \frac{Q_e}{(h_{10} - h_9)} \tag{9}$$

$$m_w = m_{ref} \frac{(X_{ref} - X_s)}{(X_s - X_w)} \tag{10}$$

$$m_s = m_w + m_{ref} \tag{11}$$

The energy balance in absorber yields

$$Q_a = m_{ref} h_{10} + m_w h_6 - m_s h_1 \tag{12}$$

For solution heat exchanger, the energy balance is as follows

$$T_5 = T_4 - \varepsilon(T_4 - T_2) \tag{13}$$

$$h_3 = \frac{m_w}{m_s}(h_4 - h_5) + h_2 \tag{14}$$

$$T_3 = \varepsilon \frac{m_w}{m_s}(T_4 - T_2) + T_2 \tag{15}$$

In Generator, the energy balance provides

$$Q_g = m_{ref} h_7 + m_w h_4 - m_s h_3 \tag{16}$$

Circulation Ratio

$$CR = \frac{m_s}{m_r} \tag{17}$$

The pump power is deduced as

$$v_{sol} = \frac{1}{\rho} \tag{18}$$

$$w_p = CP \times m_s v_{sol}(P_c - P_e) \tag{19}$$

Energy balance in the condenser yields

$$Q_c = m_{ref}(h_7 - h_8) \tag{20}$$

Coefficient of performance is given by

$$COP = \frac{Q_e}{Q_g + w_p} \tag{21}$$

Results and Discussion

Figure 2 shows the variation of COP of system with the generator temperature (T_g) for three different absorber temperatures. It is found that for a fixed absorber temperature, as the generator temperature increases, a rapid increase in the value of COP is observed up to certain temperature followed by a gradual drop after the particular generator temperature is crossed. For 40˚C absorber temperature the maximum COP of 0.41 is obtained at 120˚C generator temperature. As the generator temperature exceeds 120˚C the COP curve starts declining. Similar effect can be seen for 30˚C and 50˚C absorber temperature. This effect on COP can be explained using figure 3 and figure 4.

As the heat load to evaporator is fixed in this study, the complete effect seen on COP is due to the generator heat load. In figure 3 starting from 100˚C up to 140˚C the energy for strong solution (H3) and weak solution (H4) are decreasing whereas energy of refrigerant is almost constant. Generator heat load is the sum of weak solution and refrigerant energy deducted by strong solution energy. Now, taking the case of 40˚C absorber temperature, from figure 3 as the generator temperature increases up to 120˚C the generator load decreases. Then further increasing the generator temperature beyond 120˚C, the generator heat load gradually increases. Since, COP is inverse of generator load (as Q_e is 1kW for this study and neglecting small pump work) this decreasing and increasing phenomenon of Q_g causes the COP of the system to form a hill type curve of first increasing and then decreasing with increasing generator temperature as shown in figure 4.

In the figure 2, the performance of the system is more significant in case of lower absorber temperature i.e. for a particular generator temperature the value of COP increases as the absorber temperature decreases. For example, when T_g is fixed to 140˚C, the COP is 5 % higher for 30˚C absorber temperature than for 40˚C and 15 % higher than that of 50˚C which clearly shows that performance of system is improved by maintaining lower the absorber temperature. Lowering the absorber temperature from 40˚C to 30˚C the maximum COP value shifts from 0.41 at 120˚C T_g to 0.46 at 100˚C T_g i.e. improving the maximum COP value by 12%.

Different minimum generator temperature exists for different absorber temperatures. This minimum point can be considered as a Minimum Critical Point i.e. below this minimum temperature inaccurate COP value is obtained. This is because below this minimum generator temperature, the concentration of the weak solution becomes higher than that of strong solution which is not a desirable condition for absorption system. This effect has been plotted in figure 5. For example; the minimum generator temperature is 87.81˚C when absorber temperature is 40˚C. Similarly, unique minimum generator temperature exists. This value of minimum generator temperature is lowered as absorber temperature is lowered.

Figure 5 shows the curve drawn between generator temperature and concentration of strong as well as weak solution. Strong solution concentration is unaffected by change in generator temperature. In other hand, weak solution concentration keeps on decreasing. This is because large volume of refrigerant gets evaporated from the strong solution when the heat Q_g is supplied to the generator. On the other side, there is no change in weak solution concentration with respect to T_a whereas; strong solution concentration is decreasing as T_a increases.

Critical Point (1, 2 & 3) can be seen in the figure 5 which can be considered as the minimum operating generator temperature for a particular absorber temperature. Critical Point is the point where the curves for strong and weak solution intersect each other (i.e. for a particular absorber temperature the concentration of weak as well as strong solution is equal). For a system to work effectively, the concentration of strong solution should always be greater than that of weak solution. But for generator temperature below the critical temperature, concentration of weak solution is greater than that of strong solution hence forming an undesirable effect on the absorbing process i.e. the absorbing capacity of the solution is reduced as the solution in the absorber is already saturated and cannot absorb further amount of vapor refrigerant coming from evaporator. For example, the minimum generator temperature is 87.81˚C when absorber temperature is 40˚C.

Figure 6 shows the effect of generator temperature on the Circulation Ratio (CR) which is the ratio of mass flow rate of strong solution to mass flow rate of refrigerant. Refrigerant mass flow rate is unaffected by any change in the generator temperature whereas flow rate of strong solution decreases with increase in generator temperature as shown in figure 7. Thus, this results a decline in the value of CR. Analogous to value of Q_g, for a fixed absorber temperature the value of circulation ratio falls continuously with rise in generator temperature. For 40˚C absorber temperature the value of CR is 3.9 at 120˚C generator temperature.

As seen from graphs between COP and CR as well as from the mathematical relation between two (i.e. Q_g is directly proportional to CR and Q_g is inversely proportional to COP), these two terms seem to have an inverse relation between each other i.e. lower values of CR will represent higher COP. Same can be observed in figure 8 i.e. the value of CR decreasing and COP is increasing up to 120˚C generator temperature (for T_a = 40˚C). Beyond it, value of both COP and CR declines.

A combined plot for COP ad CR is shown in the figure 8 for T_a = 40˚C. The inverse relation between COP and CR is clearly illustrated in the figure. For a fixed generator temperature of 120˚C as the absorber temperature reduces from 50˚C to 30˚C the value of CR drops from 4.8 to 2.2 which is around 54 % drop. For the same conditions of generator and absorber temperature from figure 2, COP is increased by 15 % when absorber temperature drops to 30˚C from 50˚C. Hence, this proves that lower value of absorber temperature is preferable for an absorption system.

In figure 9 various heat duties like generator, condenser, absorber and solution heat exchanger have been plotted against the generator temperature. As we have already discussed the behavior of Q_g against generator temperature for a particular absorber temperature similar effect is seen in SHX and absorber. But the Q_c varies linearly with change in generator temperature as Q_c is dependent on mass of refrigerant and enthalpy only. The change in mass flow of refrigerant is negligible with change in generator temperature. With increase in generator temperature Q_c also increases linearly whereas Q_a, Q_g and Q_{SHX} first decreases and then increases after certain generator temperature. This effect in absorber and SHX heat duty can be explained similar to that of generator heat duty.

Figure 10 is the other form of figure 2 where COP of the system has been plotted against the absorber temperature for different generator temperature. For all generator temperature, the COP seems to be better for lower absorber temperatures. Another important point this graph reveals is that with increasing generator temperature the stability in COP of system is increased i.e. the drop in COP with increasing T_a for higher generator temperature is small relative to the lower generator temperature.

Conclusions

Vapor absorption refrigeration system with R134a – DMAC as the working fluid have been studied and analyzed theoretically. From the analysis following outcomes were obtained:

1. The value of COP is increased when generator temperature is elevated up to certain level and later declines slightly forming a hill shaped curve. For 40˚C absorber temperature, the value of maximum COP obtained is 0.41 at 120˚C generator temperature.

2. For all absorber temperatures, the shape of the COP curve is identical (i.e. hill shaped, first increasing and then decreasing). The performance of the system is improved as the absorber temperature is lowered. By lowering the absorber temperature to 30˚C from 40˚C, the maximum COP of the system is increased to 0.46 (at 100˚C generator temperature) which means 12% improvement of the maximum COP.

3. As the generator temperature is elevated or the absorber temperature is lowered, value of CR is lowered. For a fixed generator temperature of 120˚C as the absorber temperature reduces from 50˚C to 30˚C the value of CR drops from 4.8 to 2.4 which is around 54 % drop. For a fixed absorber temperature of 40˚C, the value of CR reduces from 34.66 at T_g = 90˚C to 7.56 at T_g = 100˚C and finally 3.05 at T_g = 140˚C.

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Illustrations

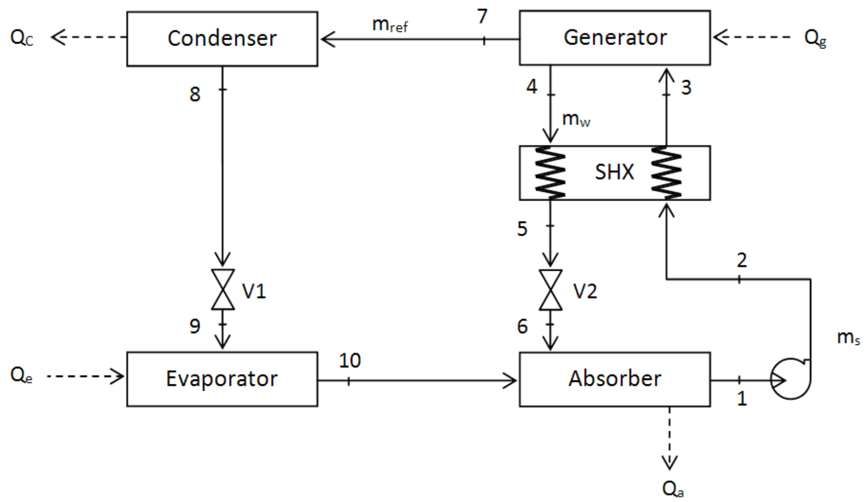

Figure 1: Simple Schematic for Vapour Absorption Refrigeration System

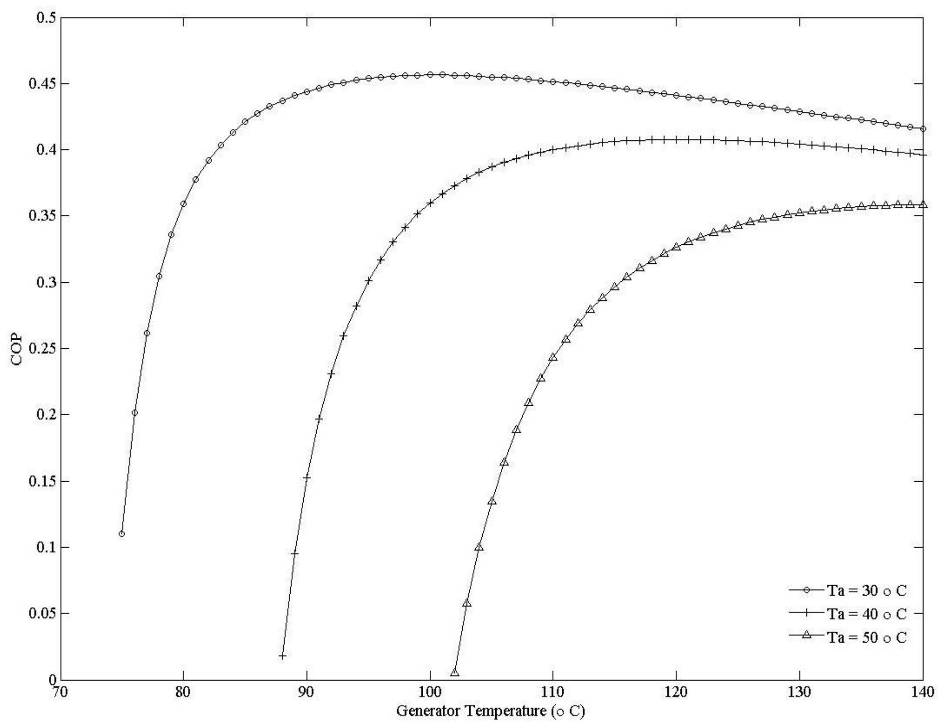

Figure 2: Variation of COP with generator temperature

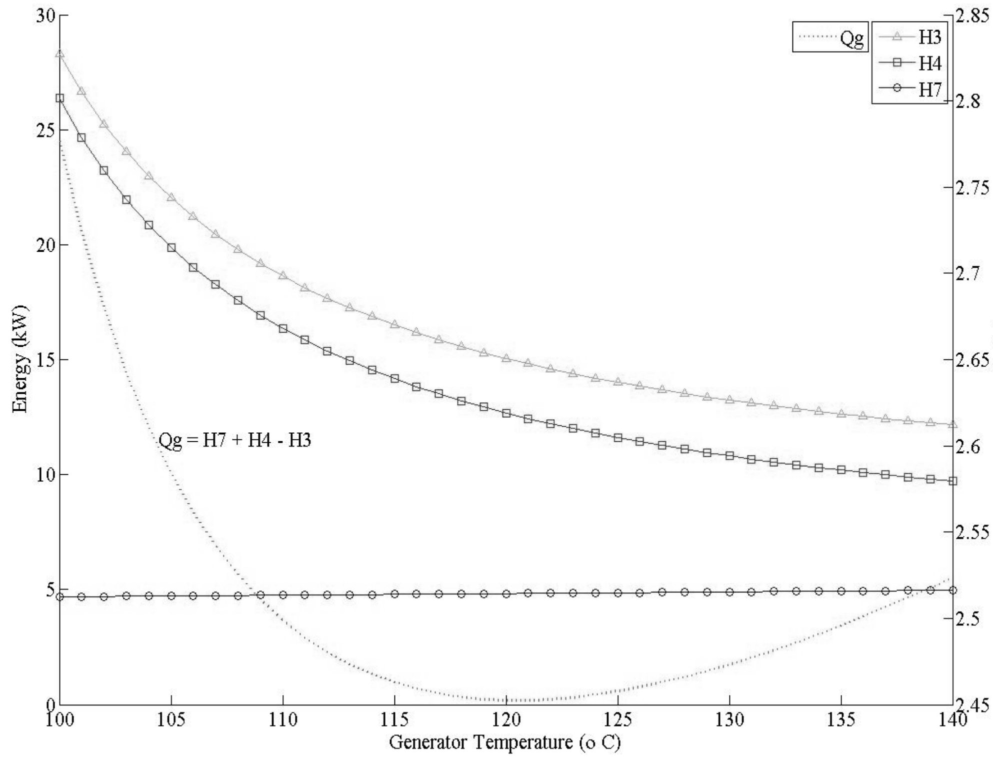

Figure 3: Variation of Energy & Qg with generator temperature for Ta = 40˚C

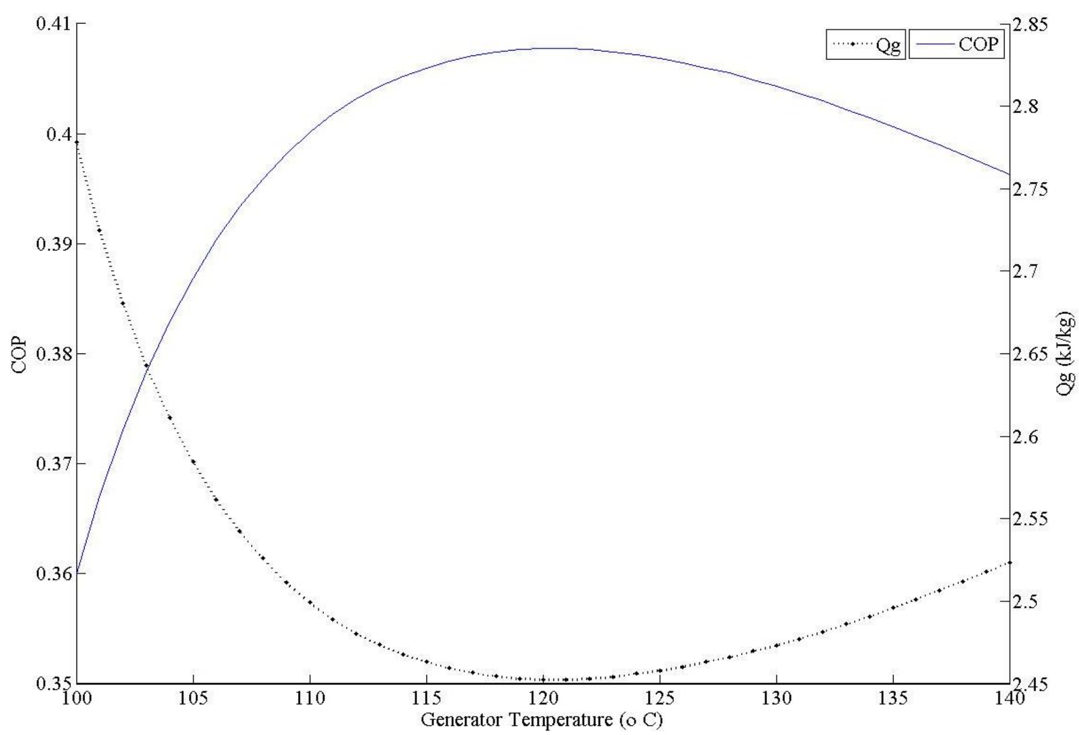

Figure 4: Variation of Qg & COP with generator temperature for Ta = 40˚C

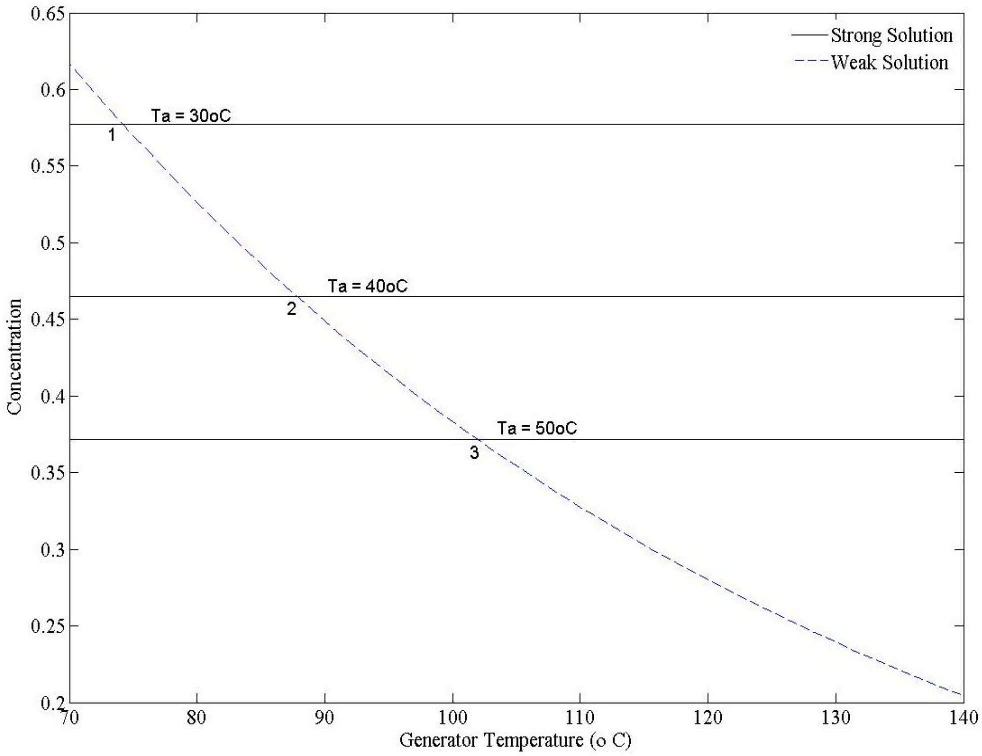

Figure 5: Variation of Solution Concentration with generator temperature for Ta = 40˚C, 50˚C & 60˚C

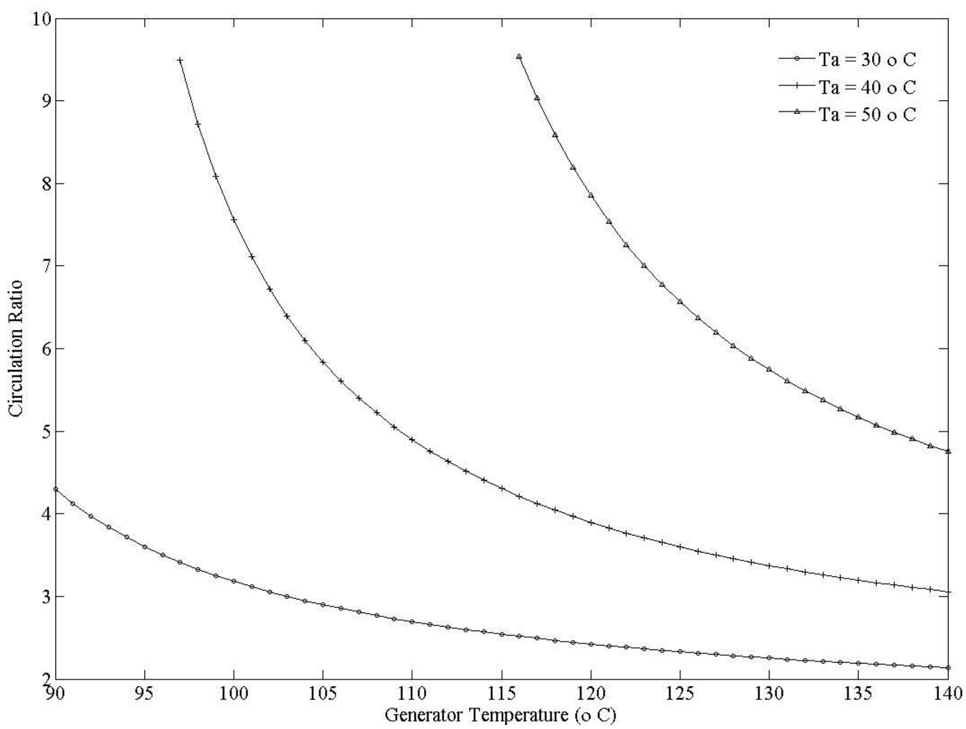

Figure 6: Variation of Circulation Ratio with generator temperature

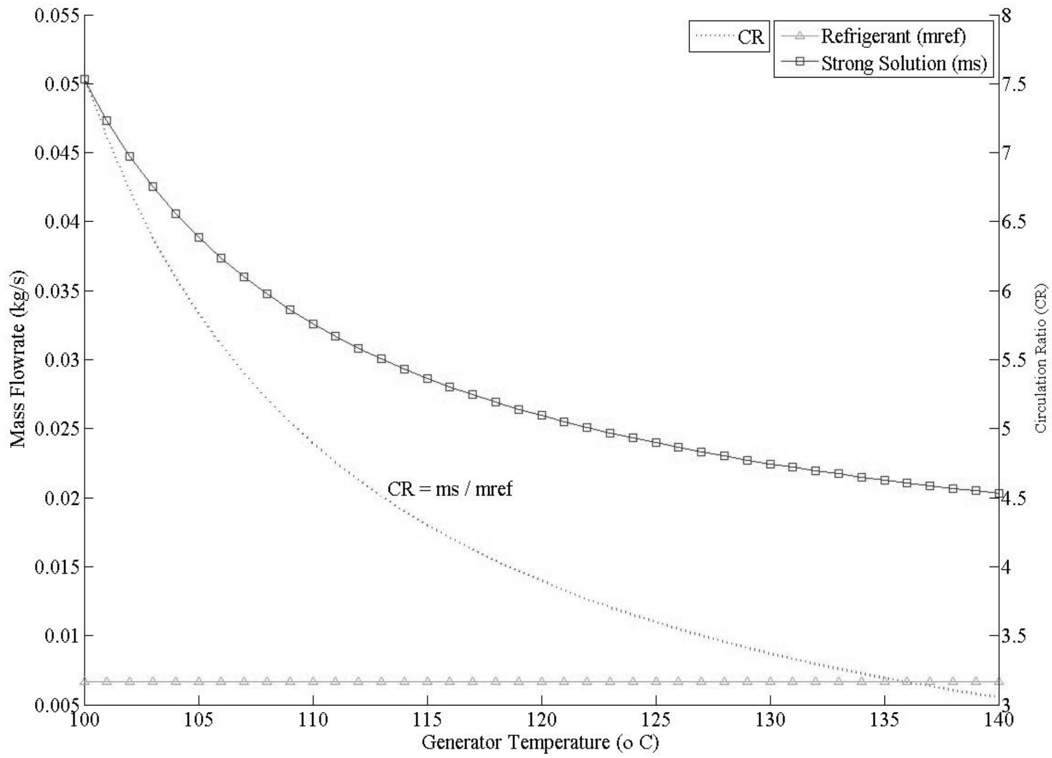

Figure 7: Variation of mass flow rate and CR with generator temperature for Ta = 40˚C

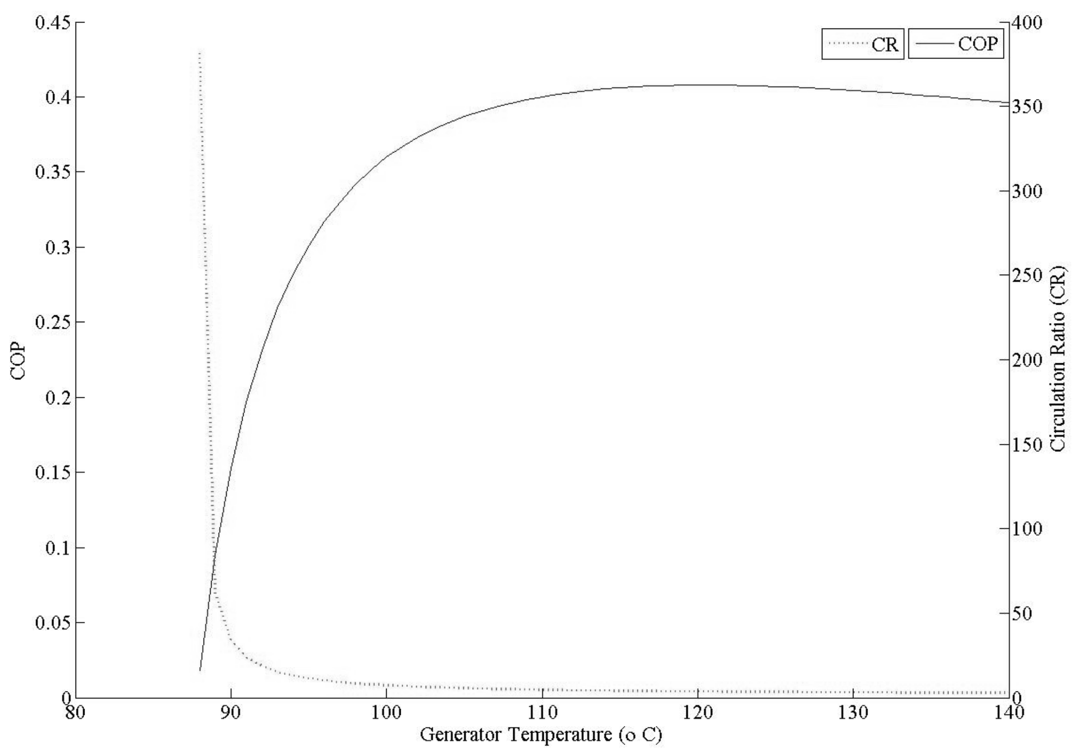

Figure 8: Variation of COP & CR with generator temperature for Ta = 40˚C

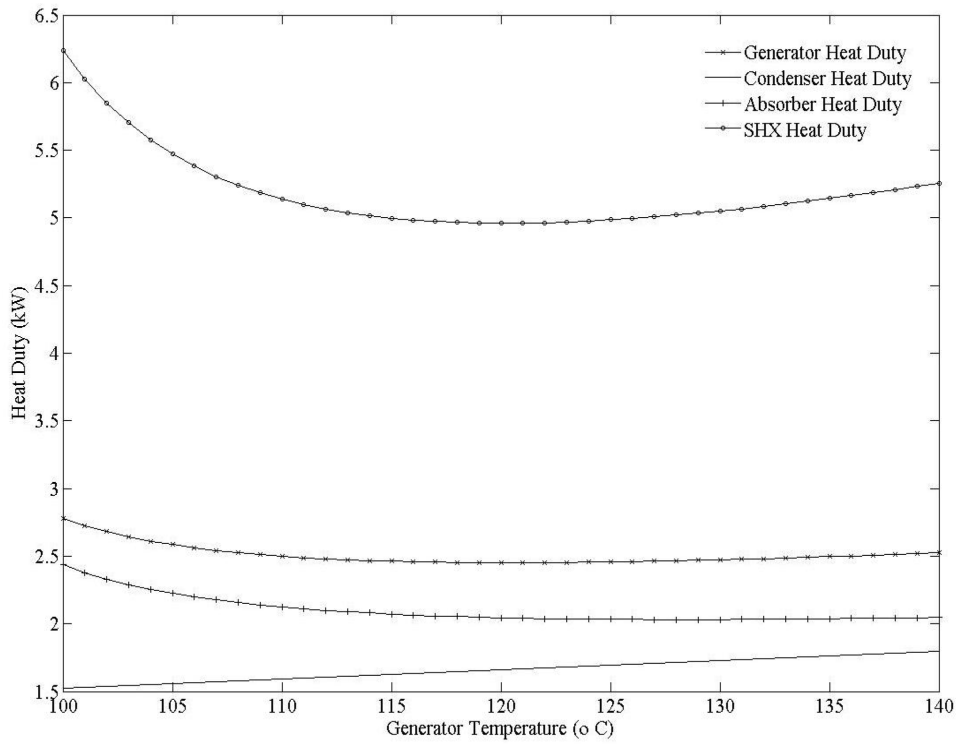

Figure 9: Variation of various heat duties with generator temperature at Ta = 40˚C

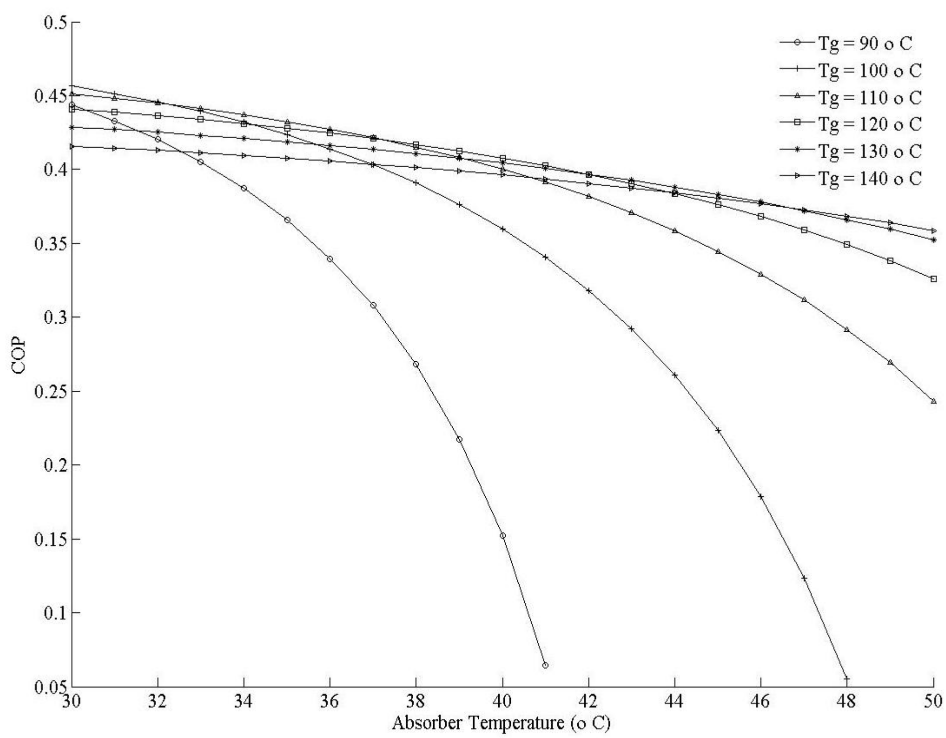

Figure 10: Variation of COP with Absorber Temperature for different generator temperature