Parametric Study of single effect combined absorption-ejector cooling system

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Abstract: - I n this study, a thermodynamic study based on heat transfer characteristics of the ammonia /water solar combined ejector–absorption refrigeration systemis performed. The influences of the generator temperature ,heat recovery ratio, solution circulation ratio, areinvestigated in order to gauge the least amount of heat transfer area that is needed for a given heat duty and the coefficient of performance (COP). The results showed that the increases in the heat recovery ratio of the solution heat exchanger improved the coefficient of performance. It is also found that the lower solution circulation ratio is better for both the coefficient of performance (COP) and the heat transfer area .It is evident that the COP improves with increases heat transfer area of the system at fixed operation condition. The total heat transfer area required is7.764 m^2 at generator temperature is 100 ℃ to reach the COP nearly 0.6148 for system cooling capacity of 5 kW.

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Key-Words: -heat transfer characteristics,performance, absorption cycles

1 Introduction

Refrigating cycles represent the predominant compressors in current applications. However, absorption cycles represents a viable alternative to the compressors, due to their relatively lower electricity consumptions. This is achieved mostly by swapping out the compressor with a thermo-mechanical version. A thermo mechanical compressor will initiate the absorption of refrigeration vapor into the absorption liquid, and will also increase the pressure of the solution. The advantage of this over compressors are its low electricity consumptions (5 % of the cooling capacity) and theutilization of external thermal energy via renewable resources (solar radiation, biomass combustion) and the usage of ammonia as refrigerants. This will result in the reduction of the usage of hazardous refrigerants in compressors (green house gasses). Despite these purported advantages, it is not without disadvantages, among them requiring a complex assembly and larger sizes and area to function .This is the primar reason why absorption cycles are utilized in industrial settings more than anywhere else. Currently, the creation of smaller units is assisted by better comprehension of the inner workings of the smaller constituent parts that makes up the absorption cycles. It is also prudent that these smaller units maximize its usage of currently available energy in the system and produces the minimum amount of wasted energy. The previous decade has seen the resurgence of research and development in increasing COP of absorption systems for both heating and cooling applications[1,2]. Many researchers have investigated the performance of the single effect absorption cooling system in recent years [3-5]. In the literature, there are numerical models that are used to assess the performance of absorption chillers on the basis of heat and mass balances for each parts. Lostec et al.[6] conducted a numerical study the effect of parametric study of evaporator and desorber temperature on the absorption chiller's performance. Ventas et al.[7]numerically analyzed single-effect absorption cycles, with ammonia-lithium nitrate solution as the working pair. It was studied the effect of the massflow rate recirculated through the absorber and the performance of the system. It was found that the value of the recirculation ratio reached maximum performance for adiabatic absorbers is between four to six times. In an adiabatic configuration, the extent of the absorber subcoolercould be six times smaller compared to the diabatic arrangements. This corresponds to a heat transfer area that is six times smaller, and the adiabatic absorber configured with a plate heat exchanger yields a 2% smaller maximum COP, and a 15-20% smaller cooling power. Seara and Vázquez [8]numerically analyzed the optimal generator temperature (OGT) in single stage NH3-H2O Absorption Refrigeration Systems (AARS). The influence of design parameters and operational conditions upon OGT were duly analyzed. The results of this analyses showed that OGT can be increased by decreasing the efficiency of solution heat exchangers and increasing the drop of pressure between the evaporator and absorber. It was also successfully shown that changing solutions,precooler heat exchangers, efficiency of the pumps, and the concentration of the refrigerant on the OGT as ineffective. Analyzing the operational conditions have led us to conclude that OGT increases as the absorption or condenstaion temperature increases, and the evaporation temperature decreases. The simulations described in the literature took into account the operational variable such as the temperatures and flow rates of the cooling water, chilled water, and hot water upon the performance of the cycles. Despite this, analysis on the parameters of design such as factors on heat transfer, solution circulation ratio [9], heat recovery ratio on the COP, and the heat transfer area arealmost nonexistent.This study intends to analyze the steady–state cycle simulation on the effect of design parameters on the COP and the heat transfer area of a single effect combined absorption-ejector cooling system, using an ammonia – water solution as working fluid.

2- System Description

Fig.1 shows schematically the main part of single effect absorption cooling system . A solar–assisted combined absorption-ejector cooling system of an evacuated tube collector that is driven internally irreversible absorption system, which essentially includes a generator, ejector, condenser, flash tank, evaporator, and absorber. The basic working principles of the system are as follows: in the generator, ammonia-water solution is heated by the solar collectors in order to vaporize the refrigerant and disassociate it from the solution. The vapor formed from the refrigerant is then condensed to conduct heat to the cooling water. The condensed liquid passes through the flash tank, which reduces the pressure and separates the flash vapor ammonia from the liquid. Then, the liquid ammonia flows to the evaporator via the expansion valve. The evaporator is a place where the refrigerant is evaporated in order to induce a cooling effect. The vapor produced here will then be transported to the absorber,where it will be absorbed into a high concentrated solution originating from the generator while eschewing heat from the water being cooled. Finally, the solution wit the low concentration is transported to the generator, which completes the cycle. An ejector is integrated into the absorption refrigeration system to suck the vapor that produced from the flash tank.

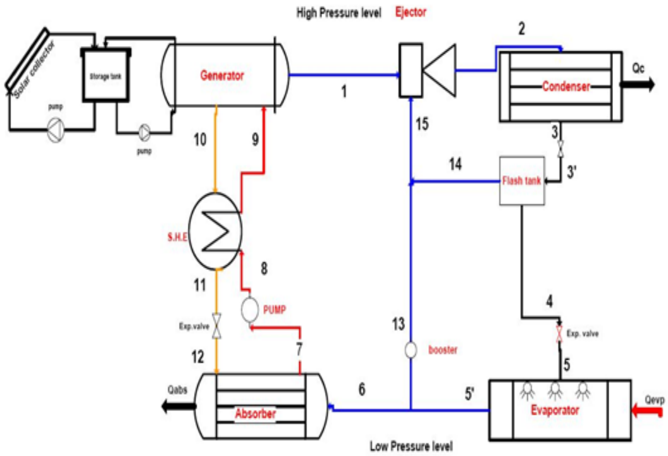

Fig .1. System schematic of single effect combined absorption-ejector cooling system**.**

3- Parametric analysis

There are several design parameters that needs to be taken into account in the analyses of their influence on the performance of a system. It should be noted that when one of them varied, the other parameters remain constant at a practical value.The range of the parameters and fixed valuesof the system are shown in Table 1 corresponds to a single-effect unit, with a 5kW capacity.The consequences of system parameters, such as the solution circulation ratio, the heat recovery ratios in the solution heat exchanger, heat transfer area of evaporator, and total heat transfer were analyzed in the context of theCOP .

Table 1. System parameters assumed for the simulation study

Input variable	Fixed value	Ranges
Capacity Q_E	5 kW	
Generator temperature T_g	100	60-120 °C
Evaporator temperature T_e	-5	
Absorber temperature T_a	25	25- 50°C
Condenser temperature T_c	30	30-50°C
Heat recovery ratio ,ε_{shx}	0.8	0-1
Inlet temperature of chilled water		16-26 °C
Inlet temperature of cooling water		22-30 °C
NH_3/H_2O solution	0.996	

concentration, X_l		
Hydraulic efficiency ,η_{hed}	0.7	
Electro-mechanical efficiency ,η_{mec}	0.7	

4- Theoretical calculations

The mass balances, energy balances, and the equations of state for the NH_3/H_2O solution, and the refrigerant in each part that are involved in the cycle are needed to simulate the system.Each of the heat exchanger in this case are of the single–pass counter-current type, and are defined by local UA and Δt_{lm}. They are shell and tube heat exchangers. The model took into account the differing single–or two-phase flow regions in the condenser, evaporator, and generator, and the solution heat exchanger is perceived in a similar manner, as boiling is possible in this part. For the external fluid, the heat exchangers utilized water. The equations on heat transfer for the regions in the components are [10]:The heat transfer area for each component's is :

$$A = Q / U_{lm}\Delta t_{lm} \tag{4.1}$$

and the log mean temperature difference

$$\Delta t_{lm} = \frac{[(T_{hot,\,i}-T_{cold,\,o})-(T_{hot,\,o}-T_{cold,\,i})]}{\ln[(T_{hot,\,i}-T_{cold,\,o})-(T_{hot,\,o}-T_{cold,\,i})]} \tag{4.2}$$

The overall heat transfer coefficient under fouling conditions can be expressed as :

$$1/U_{lm}=R_{rt}+1/h_o \tag{4.3}$$

$$R_{rt}=R_{fo}+(((1/h_i)+R_{fi})*(\frac{d_i}{d_o}))+((\frac{t_{wall}}{k_{wall}})*(\frac{d_o}{Dm})) \tag{4.4}$$

$$Dm=(d_o-d_i)/\ln(d_o/d_i) \tag{4.5}$$

$$t_{wall}=d_o-d_i \tag{4.6}$$

mass flow rate of water , G_w is calculated:

$$Q_e=G_w C_{Pw}(T_{in}-T_{out}) \tag{4.7}$$

The heat transfer coefficient (h_i) on the water side is calculated :

For laminar flow [11]:

$$Nu_i=1.86\left(\left(\frac{d_i}{L_{tub}}\right)Re_i Pr_w\right)^{1/3} \tag{4.8}$$

where

$$Nu_i = \frac{h_i\ d_i}{k_w} \tag{4.8.1}$$

For turbulent flow[12] :

$$Nu_i = \frac{[(f/2)Re_i Pr_w]}{\left[(1.07+12.7\left(\frac{f}{2}\right)^{0.5}\left(Pr_w^{2/3}-1\right)\right]} \tag{4.9}$$

where

$$f = (1.58 * \ln(Re_i)-3.28)^{-2} \tag{4.9.1}$$

The heat transfer coefficient of spray evaporator in shell side is given by Owens [13]:

$$\frac{h_o d_o}{k_r} = 0.185 * (\frac{H_{feed}}{d_o})^{0.1} * Re_o^{0.5} \tag{4.10}$$

The average heat transfer coefficient for the bundle of triangular staggered arrangement is:

$$h_{mn} = h_o(N)^{-1/6} \tag{4.11}$$

The heat transfer coefficient in the shell side of the absorber is suggested by the correlation of Cavallini and Zecchin [10] as:

$$h_o = 0.05 * Re_{eg}^{0.8} Pr_o^{0.33} (\frac{k_s}{d_o}) \tag{4.12}$$

The heat transfer coefficient in shell side of condenser is given by Butterworth [14].

$$h_o = \left(\left(1/2\, h_{sh}^2 + \left(\frac{1}{4} h_{sh}^4 + h_{mc}^4\right)^{0.5}\right)^{0.5} \left(N^{5/6}\right) - (N-1)^{5/6} \right) \tag{4.13}$$

$$h_{sh}=0.59\ (k_l/d_o)Re_o^{0.5} \tag{4.14}$$

$$h_{mc} = \frac{0.728 k_l/d_o\ ((\rho_l\ (\rho_l-\rho_g)\ h_{fg} d_o^3\ )}{(\mu_l(T_c-T_w)k_l)^{1/4}} \tag{4.15}$$

In order to assess the COP of the cycle, the governing equation required through each component [15].

- **Mass balance**

$$\sum_e \dot{m}_e = \sum_i \dot{m}_i \tag{4.16}$$

- **Energy balance**

$$Q_K=\sum \dot{m}_e h_e - \sum \dot{m}_i h_i \tag{4.17}$$

Where Q_K is the heat added to component K at temperature T_K .

- **Pump**

The electrical power needed for the pump

$$w_p = \frac{\dot{m}_7 v_7\ (P_8-P_7)}{\eta_{hed}\eta_{mec}} \tag{4.18}$$

$$w_p = \dot{m}_8 h_8 - \dot{m}_7 h_7 \tag{4.19}$$

- **Heat exchanger**

The heat recovery ratio in the solution heat exchanger is expressed by

$$\varepsilon_{shx} = \frac{h_{10}-h_{11}}{h_{10}-h_8} \tag{4.20}$$

- **Equation of state**

The thermodynamic properties of states (1)-(6),(13)-(15) in Fig .1 are determined by NH_3 and other properties at states (7)-(12) can be calculated based on the binary mixture of NH_3/H_2O which is given by the correlations of Sun [16].

The solution circulation ratio is calculated using

$$S_{ratio}=\frac{X_{str}}{X_{str}-X_{weak}} \tag{4.21}$$

The coefficient of performance COP is defined as

$$COP=\frac{Q_E}{Q_G+w_p} \quad (4.22)$$

Finally , the total heat transfer area is given by Xu et al. [17] as :

$$A_{total}= A_E+A_C+A_G+A_{SHE}+A_{abs} \quad (4.23)$$

5-Results and Discussion

The effects of important parameters such as generator temperature (Tg), solution circulation ratio (S),heat recovery ratio on both COP and the heat transfer area are examined and presented.

5.1 Effects of design parameters on COP

The influence of both solution circulation ratio (S) and the heat recovery ratio in the solution heat exchanger on the COP is shown in Fig. 2. It indicates that the system performance increases with an increase the heat recovery ratio in the solution heat exchanger. It should also be pointed out that the COP becomes rather weak as the recovery ratio increases. This is assumed to be due to the concentration of the concentrated solution being constant.The effect of operating temperature on the system performance and solution circulation ratio (S) of the combined absorption cycle are presented in Fig.2. The solution circulation ratio (S) an important design and optimizing parameter since it is directly related to the size and cost of the system components . Because of this ,it is observed from simulation are given in this figure that the value of COP increases as (S)decreases. This reveals that with increase the generator temperature the difference between the concentration of the strong solution and the weak solution becomes larger, and consequently, the system performance increases in tandem with the increase in generator's temperature The results agree with the work done by others researchers [17]. Fig .3.shows the variation os SHE outlet enthalpy with heat recovery ratio. As known , if the heat recovery increases, the heat transfer between the strong solution and the weak solution increases , and as a results of this, the enthalpy of the strong solution (h_{11}) decreases and that of the weak solution (h_9) increases . With an increase the enthalpy of the strong solution entering the generator , so that it required less energy . Similarity , by cools down the weak solution , so that it can absorb the ammonia at lower temperature, consequently, the heat rejected from the absorber also decreases .

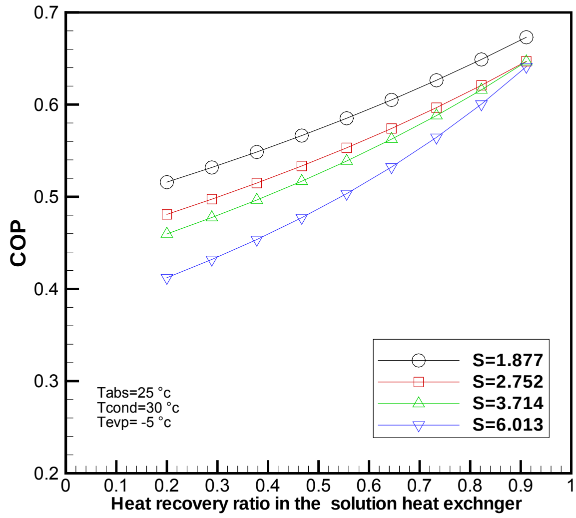

Fig.2. Variation of COP with heat recovery ratio of solution heat exchanger at different solution circulation ratio.

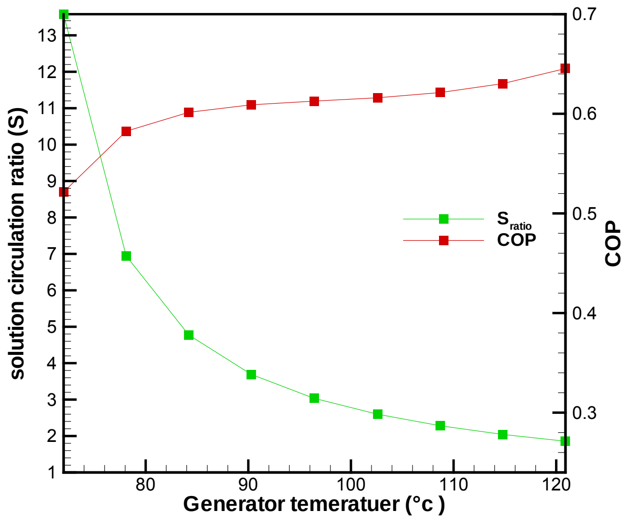

Fig.3. solution circulation ratio (S) (left axe) and the COP (right axe) as a function of Tg

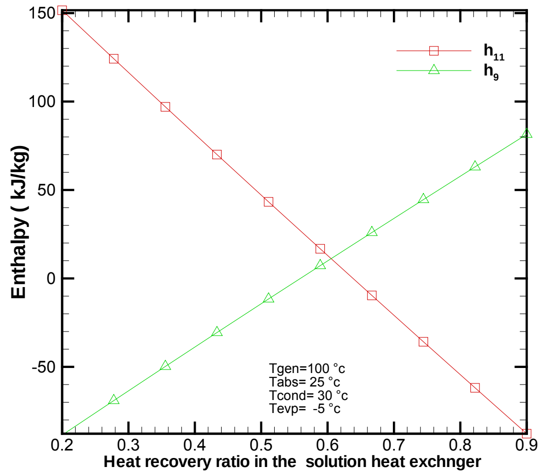

Fig.3. Variation of enthalpy transfer efficiency with heat recovery ratio of solution heat exchanger .

5.2 Effects of design parameters on the heat transfer area

The problem of size is concerned with determining the dimensions of the dimensions of the heat exchanger, choosing a suitable heat exchanger and confirming the dimensions to meet the design of the absorption cooling system. One of a tried and proven method of decreasing the heat transfer area is to improve the heat transfer technology. Fig. 4.shows the effect of heat transfer area on the performance of the system at operation condition (T_{abs}=25 ℃ , T_{evp}=-5 ℃ ,T_{cond}=30 ℃) and different generator temperature. It can be seen that the COP increases as total heat transfer area increases. It is also noted that the total heat transfer area decreases with increases the generator temperature. The strong affinity between the refrigerant and the absorbent requires more energy to evaporate them.In other word,isothermal heat interactions at the heat absorption and the heat rejection , which proceed at very slow rates necessitated large heat transfer areas.

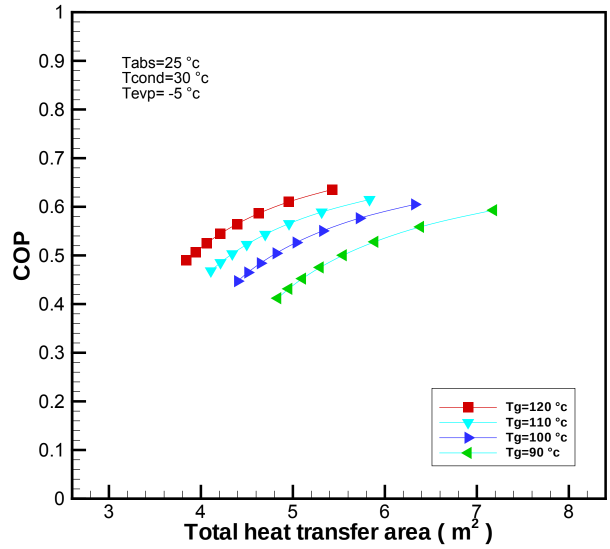

Fig.4. Effects of total heat transfer area on COP at different generator temperature (Tg) .

Fig.5.illustrates the effect of both solution circulation ration and the heat recovery ratio on the total heat transfer area. It is observed that the heat recovery ratio in the solution heat exchanger is directly proportional to the total heat transfer area to a point that is greater than 0.80.It is also noted that the total heat area is inversely proportional to (S). The results also indicated that both the heat recovery ratio and COP are more responsive towards lower(S). Moreover, the influence of the heat recovery ratio on both the system performance (or energy cost) and the total heat transfer area (or manufacturing cost) are inversely related [17].For theoretical considerations, the optimal heat recovery ratio value was determined to be 0.8.A comparison of total heat transfer area required for single effect combined ejector absorption cooling system is made in Fig.6. It is shown that for system cooling capacity of 5 kW, the total heat transfer area required is 7.764 m^2 at generator temperature is 100 °C to reach the COP nearly 0.6148 .

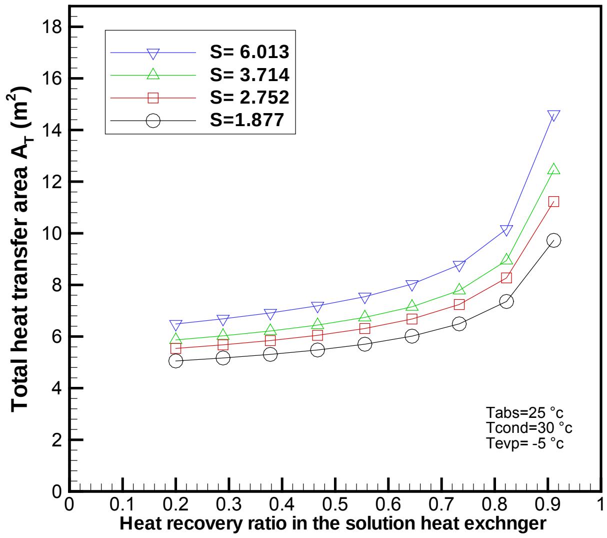

Fig.5. Effect of heat recovery ratio and the solution circulation ratio on the total heat transfer area.

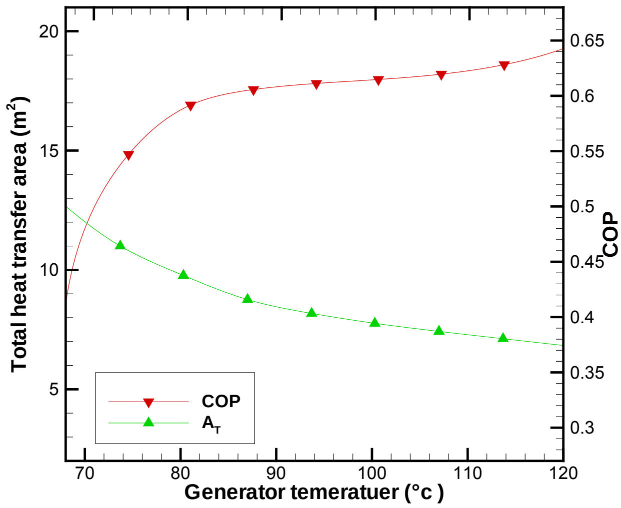

Fig.6. Total heat transfer area of the cycle (left axe) and the COP (right axe) as a function of Tg.

6. Case study

With the aim to analyses the system performance of the combined ejector absorption cooling system, comparisons (shown in Figs.7,a and b.) have been made between the tube length and tube number . Fig.7a. shows the evolution of COP with variation of generator temperature at different tubes number of the evaporator. It should also be pointed out that the COP increases with increasing amount of tubes.

When the number of tube increases, the heat transfer area increases. As seen in Fig.7b, the length of the tube of the evaporator greatly influences the performance of the cycle, and it can be observed that the COP undergoes a sharp increases up till it reaches its maximum value, upon which it will remain constant visavis the generator's temperature. There exists a low generator temperature limit, which is cannot be operated at lower than its limit [18]. An important point is that the maximum COP value for the evaporator load of 4.936 kW is obtained in case of (L_T=1 m and NT=25).

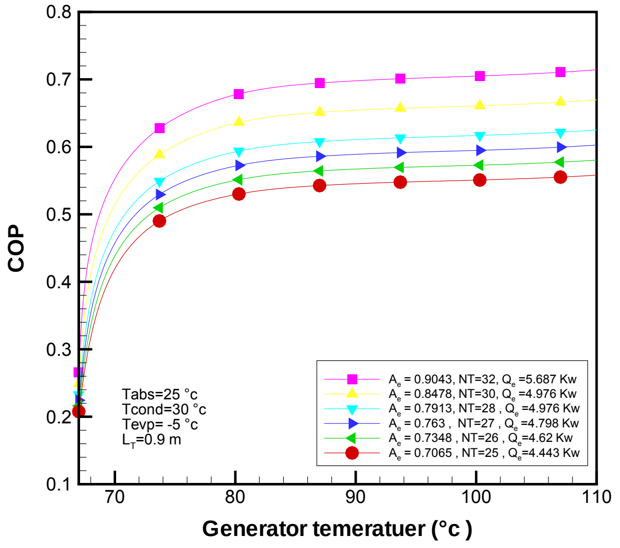

Fig.7, a.Variation in COP with respect to generator temperature at different NT.

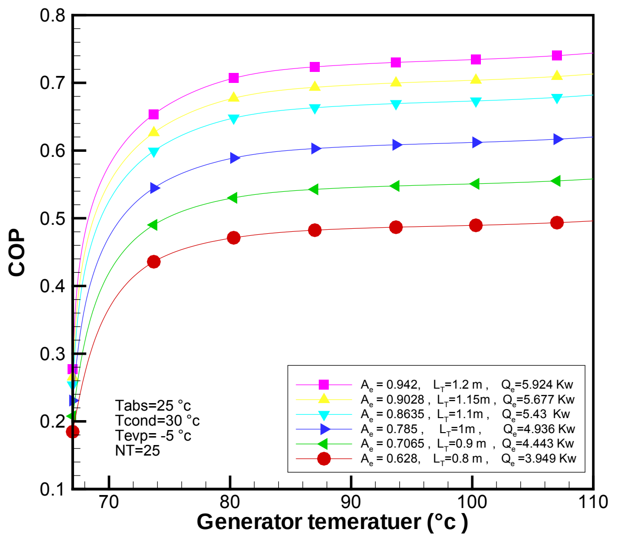

Fig.7, b.Variation in COP with respect to generator temperature at different tube lengths.

7. Conclusions

This paper paid attention to the numerical analysis of the influence of parametric design on COP and the total heat transfer area. The findings from the numerical analysis under differing design conditions are summarized below.

- It was discovered that COP is directly proportional to the heat recovery ratio of the solution heat exchanger, and inversely proportional to solution circulation ratio.
- The generator temperature has a great effect on the system performance and the solution circulation ratio.
- The total heat transfer area is directly proportional to the heat recovery ratio and solution circulation ratio, but is inversely proportional the temperature of the generators.
- The COP of the system is 0.6148 working between condenser pressure of 11.67 bar and evaporator pressure of 3.544 bar and required total heat transfer area approximately of 7.764 m^2.
- It is observed that the dimension of the tube of the evaporator strongly effects the performance of the cycle.

  Therefore , this study can be useful to show system parameters affecting on the system performance.

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