Vapour Compression-Absorption Cascade Refrigeration System-Thermodynamic Analysis

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Abstract— **This paper describes the study carried out to analyse a refrigeration system in which a compression system at the low temperature stage and an absorption system at the high temperature stage are cascaded to generate cooling at low temperatures. These cycles were analysed thermodynamically and were compared with each other using different refrigerants in the compression and absorption sections at the same operating conditions, that are an evaporator temperature of 233K (-40˚C) and condenser temperature of 313K (40˚C). R22, R125 and R32 have been considered as refrigerants in the compression stage and the pair of NH_3–H_2O and R134a-E181 in the absorption stage**

Keywords—Cascade refrigeration system, Coefficient of performance, Vapour compression refrigeration system and Vapour absorption refrigeration system.

I. INTRODUCTION

A Vapour-compression refrigeration cycle with only one stage of compression is called a single-stage refrigeration system. A cascade refrigeration system, on the other hand, employs 2 or more individual refrigeration cycles operating at different pressure and temperature levels. The duty of the lower temperature cycle is to provide the desired refrigeration effect at a relatively low temperature. The condenser in the lower-temperature cycle is thermally coupled to the evaporator in the higher-temperature cycle. Thus, the evaporator in the higher cycle only serves to extract the heat released by the condenser in the lower cycle. Then this heat is rejected into the ambient air or a water stream in the condenser of the higher cycle.

Since high ratios of pressure across the compressor cause undesirably high discharge temperatures, low volumetric efficiencies and excessive stresses on compressor parts, the maximum allowable pressure ratio for reciprocating compressors is limited to about 9.If the pressure ratio exceeds this limit for a specific application requiring a relatively low refrigeration temperature, it can be decreased using a cascade refrigeration system instead of a single-stage one[3]**.**

In this work, cascading of vapour compression cycle and vapour absorption cycle has been done. This set up is much more energy efficient because the high temperature stage of the system uses low grade heat energy instead of high grade electrical energy which is much more expensive.

Canan Cimsit et. al.[4] analysed the compression – absorption cascade refrigeration cycles . It is seen that using the absorption and vapour compression cascade refrigeration cycle have the advantage that cooling can be done by using less electric energy consumption than classical vapour compression cycle for low temperature cooling applications. When the compression- absorption cascade refrigeration cycles and the classic vapour compression refrigeration cycles are compared for sample application at the same conditions for the same cooling capacity, depending on absorption fluid pairs such as LiBr-H_2O and NH_3-H_2O, 48-51% less electric energy is consumed in the cascade systems, but heat is supplied to the cascade cycle at 363 K generator temperature from 76.45 kW to 117.86 kW, respectively. The coefficient of performance of the cascade cycle using LiBr-H_2O and NH_3-H_2O fluid pairs in absorption section increases by increasing the generator and evaporator temperatures, but it reduces by increasing the condenser temperatures. Jose´ Fernandez-Seara et. al.[5] obtained results for complete analysis about the compression–absorption cascade refrigeration system. The intermediate temperature level is an important design parameter that causes an opposite effect on the COP of the compression and absorption systems. Therefore, the cascade system COP presents a maximum when the intermediate temperature is varied. L. Kairouani & E. Nehdi[6] concluded that an absorption/compression refrigeration system is proposed to improve the overall cycle efficiency. The COP excluding the pump work and the generator energy required is as high as 5.4– 6.2, which is higher than that of the single vapour compression cycle and absorption cycle, under the same operating conditions (evaporation temperature at 263 K and condensation temperature at 308 K). This system presents an opportunity to reduce the continuously increasing electrical energy consumption. Vaibhav Jain et. al. [10] concluded that electric power requirement in VCRS is reduced substantially by cascading it with absorption system. However, the total size of the VCRS will increase when it is cascaded, but the running cost is expected to decrease due to the utilization of waste heat available at lower cost. In addition, the COP of vapour compression section will also increase because of low electric power requirement. The larger the temperature difference in cascade heat exchanger, the lower the COP of the system; however, a lower temperature difference will lead to increased heat exchanger size and cost. Increasing the size of heat exchanger increases the overall performance of system, but it also increases the system cost. H.M. Getu et. al[11] concluded that an increase in condensing temperature resulted in a decrease in COP and an increase in refrigerant mass flow ratios. An increase in evaporating temperature increased COP of the system and decreased mass flow ratios. An increase in temperature difference in cascade condenser reduced both COP and mass flow ratios. An increase in isentropic efficiency of compressors increased COP linearly. Hansaem Park et al.[12] concluded that the optimal intermediate temperature, which lets the system COP maximized, always exists when other conditions are fixed. When the R134a condensing temperature increases, the optimal intermediate temperature is also elevated and the corresponding maximum COP decreases. If the R410A evaporating temperature rises, both the optimal intermediate temperature and the corresponding maximum COP escalate. The temperature difference in a cascade heat exchanger has a negative influence on the system COP when it gets bigger. The optimal temperature also diminishes if the temperature difference increases.

The objectives of this work are to study the performance of the refrigerants in the compression stage and refrigerants in the absorption stage and its overall effect on the cascade system,to study and analyse the cascade system by varying various performance parameters.

II. THE ANALYSIS OF THE CYCLES

The thermodynamic analyses of total six cycles were done using the same methodology as explained in previous section. For vapour compression stage of the cascade cycle three refrigerants were chosen; R22, R125 and R32.For vapour absorption stage of the cascade refrigeration system two solutions which were chosen are NH_3-H_2O and R134a-DMETEG.For the description of this cycle we assume that NH_3-H_2O was used in the absorption section and R-32 was used in the vapour compression section of this sample cycle. In this cycle, the strong solution of NH_3-H_2O is drawn from the heat exchanger by a pump. The high pressure cool mixture enters the generator, where the heat is added to drive the ammonia from the solution. The weak solution of NH_3-H_2O leaves the generator, its pressure is reduced to absorption pressure by a solution expansion valve (TV), and then it enters the absorber. To increase the temperature of stream (1') and decrease the temperature of the stream (3), a counter flow heat exchanger is located between strong and weak solutions. The refrigerant (ammonia) is condensed in the condenser, leaves it as a saturated liquid; then it passes through the refrigerant expansion valve (TV). Afterwards, the refrigerant passes through the cascading condenser, where it evaporates by absorbing the heat released by the condenser of the vapour compression cycle. This cold vapour then enters the absorber, where it mixes with a hot solution, and it is absorbed. In the vapour compression section, the refrigerant is compressed to a high pressure in the compressor, and then enters the condenser 1. Then, the condensed refrigerant passes through the refrigerant expansion valve (TV), and enters the evaporator.

A simple compression-absorption cascade system as shown in fig.3.4 can be used for R134a-DMETEG solution but NH_3-H_2O solution requires rectification for ammonia vapour in the generator. Therefore, schematic diagram for ammonia-water pair will be slightly different as shown in figure 1.It will have an additional rectifying column in the absorption stage of the cascade refrigeration system. This rectification column (Fig.1) will ensure that only ammonia vapours are entering the condenser instead of ammonia- water mixture.

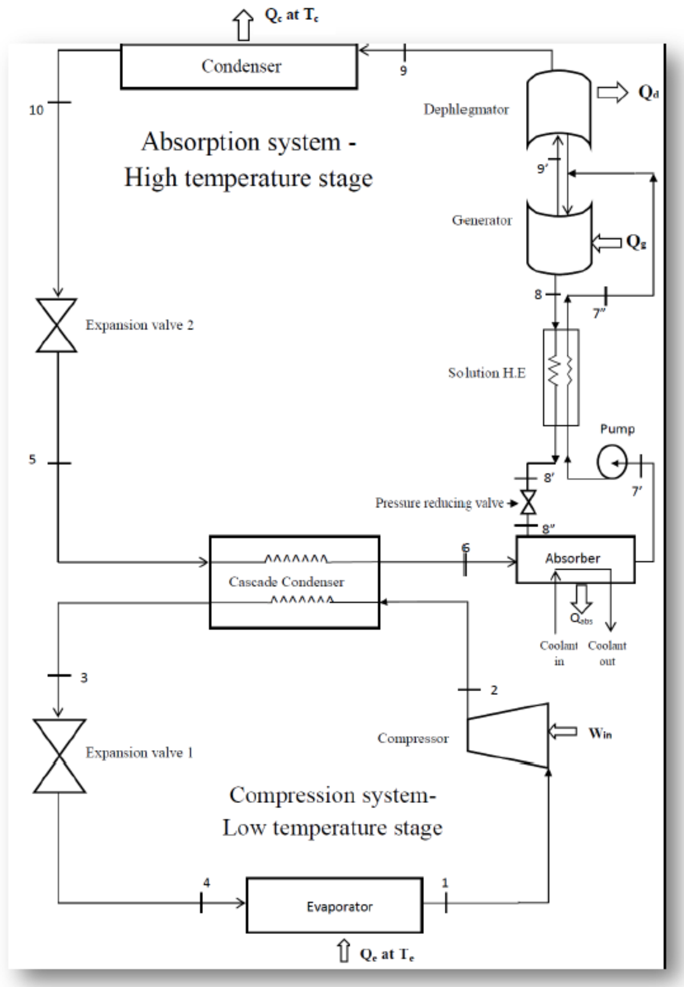

Fig. 1 Compression-absorption system with a rectifying column

A.Thermodynamic Analysis

The assumptions which were used during the following analysis were:-

1. The system is in steady state.
2. Ammonia and water solutions in the absorber and generator are assumed to be in equilibrium at their respective temperatures and pressures.
3. Refrigerants (ammonia and R134a) are in saturated states in condenser and evaporator pressure.
4. Strong solution and weak solution leaving the absorber and generator are saturated.

The required calculation can be done by using mass and energy balance relations for every component in the cycle.

For the compression stage of the cascade system:-

Cascade Condenser:-

$$Q_{casc} = m(h_2 - h_3) \quad (1)$$

Compressor:-

$$W_{in} = m(h_2 - h_1) \quad (2)$$

Expansion Valve:-

$$h_3 = h_4 \quad (3)$$

Evaporator:-

$$Q_{casc} = m(h_1 - h_4) \quad (4)$$

m is mass flow rate of refrigerant, kg/s; *h* is enthalpy, kJ/kg; and W_{in} is compressor power input, kW

For the absorption stage of the cascade system:-

Absorber:-

$$m_6 h_6 + m_{8''} h_{8''} = m_{7'} h_{7'} + Q_{abs} \quad (5)$$

$$m_{ws} X_{ws} + m_r = m_{ss} X_{ss} \quad (6)$$

where Q_{abs} is the absorber head load in kW; X is the concentration; m_{ws} = m_6 is the mass flow rate of the weak solution in kg/s; m_{ss} = $m_{7'}$ is the mass flow rate of the strong solution in kg/s; and $\dot{m}_r$ is the mass flow rate of the refrigerant in kg/s. Here, state 1 is a saturated liquid at the lowest temperature in the absorber and is determined by the temperature of the available cooling water flow or air flow

Solution Heat exchanger:-

$$m_8 h_8 + m_{7'} h_{7'} = m_{8'} h_{8'} + m_{7''} h_{7''} \quad (7)$$

Where *m* is the mass flow rate through the heat exchanger in kg/s and h is enthalpy in kJ/Kg.Subscripts have their usual meaning which can be referred from the diagram.

Generator:-

$$m_{ws} X_{ws} + m_r = m_{ss} X_{ss} \quad (8)$$

$$m_{9'} h_{9'} + Q_g = m_9 h_9 + m_{7'} h_{7'} \quad (9)$$

Where *m* is the mass flow rate through the heat exchanger in kg/s and h is enthalpy in kJ/Kg. Subscripts have their usual meaning which can be referred from the diagram. X is the concentration.Q_g is the heat input to the generator.

Condenser:-

$$m_9 h_9 = m_{10} h_{10} + Q_c \quad (10)$$

Where *m* is the mass flow rate through the heat exchanger in kg/s and h is enthalpy in kJ/Kg. Subscripts have their usual meaning which can be referred from the diagram. Q_{cond} is the heat rejected from the condenser.

Expansion (throttling) valves:-

$$h_{8'} = h_{8''} \quad (11)$$

$$h_{10} = h_5 \quad (12)$$

h is enthalpy in kJ/Kg. Subscripts have their usual meaning which can be referred from the diagram

Cascade condenser:-

$$m_5 h_5 + Q_{casc} = m_6 h_6 \quad (13)$$

Where *m* is the mass flow rate through the heat exchanger in kg/s and h is enthalpy in kJ/Kg. Subscripts have their usual meaning which can be referred from the diagram. Q_{casc} is the heat absorbed in the cascade condenser.

B.Model data table for thermodynamic analysis

A model input data table has been tabulated for parameters which will stay constant throughout the analysis in this thesis. The cascade refrigeration system has been designed to attain a temperature of -40˚C. All the required parameters have been tabulated in the table 1.Pump efficiency and compressor efficiency were assumed to 0.6 and 0.75 respectively .This analysis was conducted for a refrigeration capacity of 15 ton of refrigeration which is equivalent to 52.5 kW.

Table 1 Model input data for refrigeration cycle

Evaporation temperature at the compression system, T_{evap} [C]	-40℃
Refrigeration capacity ,Q	15 TR
Condensation temperature at the compression system, T_{cond} [C]	40℃
Generation temperature at the absorption system, T_{gen} [C]	170℃
Absorber temperature at the absorption system, T_{abs}	45℃
Temperature difference at the evaporator–condenser, $\Delta T_{evap-con}$ [C]	6℃
Pressure drop between evaporator and absorber, $\Delta P_{evap-abs}$ [bar]	0.1 bar
Exit temperature of the weak solution from pre-heating heat exchanger, $T_{x,exit}$	50℃
Solution heat exchanger Effectiveness, ε	0.8
Pump efficiency,η	0.6
Compressor efficiency,η_{comp}	0.75

III. RESULTS AND DISCUSSION.

The analysis has been done for the compression-absorption cascade refrigeration cycle for different cascade cycle parameters such as intermediate temperature, intermediate temperature differences, generator temperature and condenser temperatures. The performance of the cascade cycles are discussed as follows:-

A. R32, R125, R22 in compression stage paired with NH_3-H_2O and R-134a-DMETEG in the absorption stage.

This analysis was carried out to determine the optimum intermediate temperature at which the system exhibits maximum COP for the cascade cycle. NH3-H2O is considered for the absorption stage of the system. It was observed that system has optimum intermediate temperature at 276K when R22 is used for compression stage and has optimum intermediate temperature at 278K when R125 and R32 are used for compression stage.

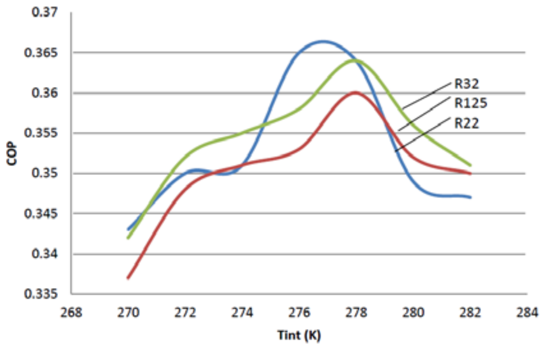

Fig.2 COP of the cascade system V/S intermediate temperature (Tint) :- R717-Water

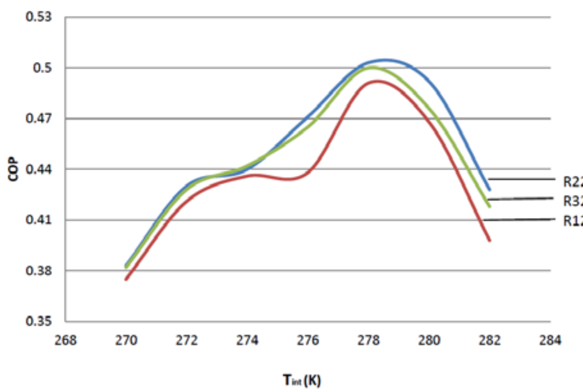

Fig.3 COP of the cascade system V/S intermediate temperature (Tint) (R134a-DMETEG)

This analysis is same as the previous one but only difference is that in R134a-DMETEG is considered for the absorption stage of the system in this case. It was observed that system has optimum intermediate temperature at 278K for all the three refrigerants viz. R22, R125 & R32 in the compression stage of the cascade system.Again, R22 turns out to be a better performing refrigerant in terms of COP whereas R125 is the least performing one. It can also be observed that system is exhibiting higher COPs at the optimum intermediate temperature when R134a-DMETEG solution is used in absorption stage.

B. Performance evaluation by varying the temperature difference in the cascade condenser.

This analysis is carried out to determine the optimum temperature difference between the compression stage and the absorption stage of the cascade system. In the first case, ammonia –water solution in the absorption stage is coupled with R22, R125 & R32 in the absorption stage. It is observed that the system performance increases with decrease in temperature difference in the cascade condenser whereas a higher temperature difference tends to degrade the system performance in terms of COP. The results suggest that the temperature difference in the cascade condenser should be as low as possible for higher COPs of the cascade system but it is not advisable to have ΔT less than 6 degrees. Cascade condensers with very low ΔT values have sizing problems i.e to transfer the same amount of heat; a cascade condenser of a higher capacity will be required because of smaller temperature difference.

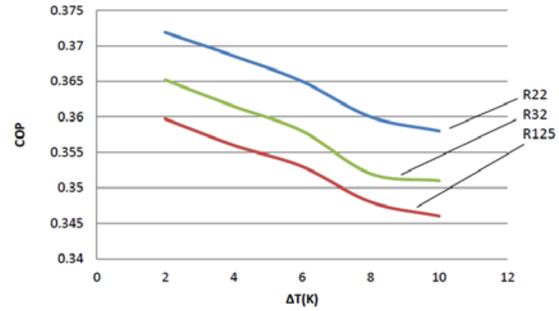

Fig 4 COP of the cascade system v/s intermediate temperature difference :-R717-Water

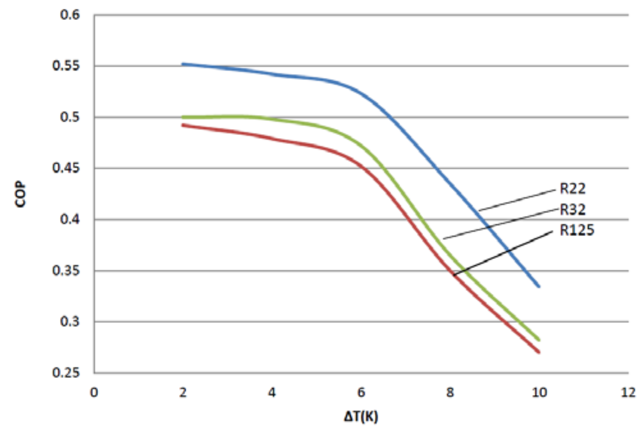

Fig. 5 COP of the cascade system v/s intermediate temperature difference:-R134a-DMETEG

In both the cases, R22 in the compression stage proves to be the better performing refrigerant. It is evident that the system COP is higher when R134a-DMETEG pair is used in absorption stage as compared to R717-Water pair. The COP value for R22 is varying from 0.55 at ΔT = 2K to 0.35 at ΔT=10K.Therfore, there is a considerable degradation in the system performance when intermediate temperature difference in the cascade condenser is high.

C. Variation in heat input to generator by varying the temperature difference in the cascade condenser.

The results show that as the temperature difference in the cascade condenser increases; the heat input required in the generator also increases. Therefore, we must opt for an optimum temperature difference in the cascade condenser.It is quite evident from the results that when refrigerant R125 is used in the compression stage then the system requires higher heat input in the generator whereas using R22 in the compression stage requires the least amount of generator heat input. R125 in the compression stage leads to higher generator heat input because the compressor work required for R125 is higher than R22 and R32 for the same pressure lift. Therefore, cascade condenser load is always higher when R125 is used. This leads to the requirement of higher mass flow rate of refrigerant in the absorption stage which results in higher heat input requirement in the generator to maintain the desired generator temperature.

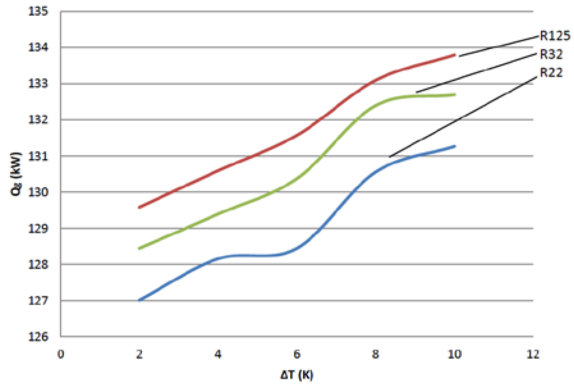

Fig. 6 Generator heat input(Q_g) v/s intermediate temperature difference (ΔT) :-R717-Water

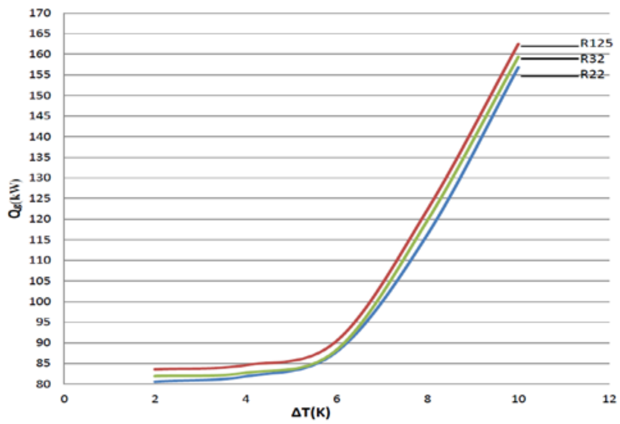

Fig.7 Generator heat input(Q_g) v/s intermediate temperature difference (ΔT) :-R134a-DMETEG

D.Variation in heat input required in generator while varying the intermediate temperature.

It is evident that R125 requires maximum amount of heat input in the generator. Heat required in the generator for a R125 system is varying from 141 kW at 270 K to 128kW at 278 K(optimum intermediate temperature) and then it starts rising again to 132kW at 282K.Therefore system should work under optimum intermediate temperature for higher COP and lesser heat input to the generator.

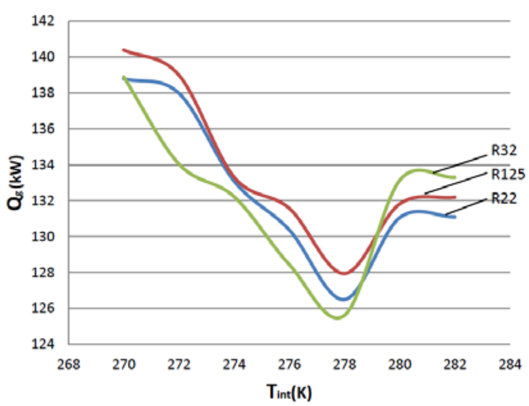

Fig. 8 Generator heat input(Q_g) v/s intermediate temperature (T_{int}):-R717-water)

This same analysis was repeated by replacing R717-water pair in absorption stage by R134a-DMETEG .The results showed similar trend i.e Q_g decreases as the intermediate temperature increases but then increases again as the temperature exceeds the optimum intermediate temperature.

In this case as well; R125 requires maximum amount of heat input in the generator. Heat required in the generator for a R125 system is varying from 103 kW at 270 K to 84kW at 278 K(optimum intermediate temperature) and then it it starts rising again to 105kW at 282K.Therefore system should work under optimum intermediate temperature for higher COP and lesser heat input to the generator.

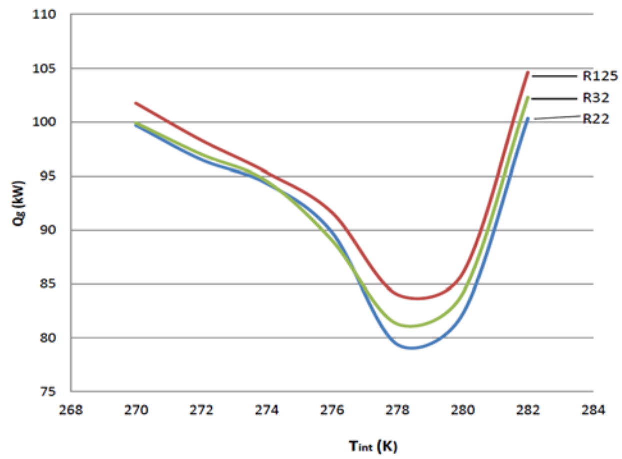

Fig.9 Generator heat input(Q_g) v/s intermediate temperature (T_{int}) :-R134a-DMETEG).

E. Performance evaluation by varying the condenser temperature.

Up till now, all the analysis was carried out at the condenser temperature of 40 ˚C but the should also be analysed for temperatures higher and lower than 40 ˚C .Therefore, an analysis was carried out by varying the condenser temperature from 35˚C to 55˚C. It was found out that as the condenser temperature increases; the heat input requirement in the generator also increases.

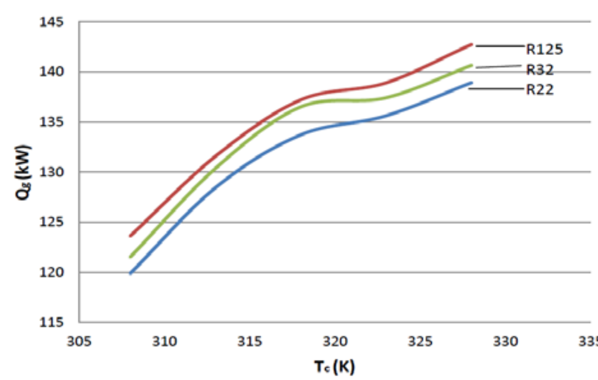

Fig. 10 Generator heat input(Q_g) v/s Condenser temperature T_c :-(R717-Water)

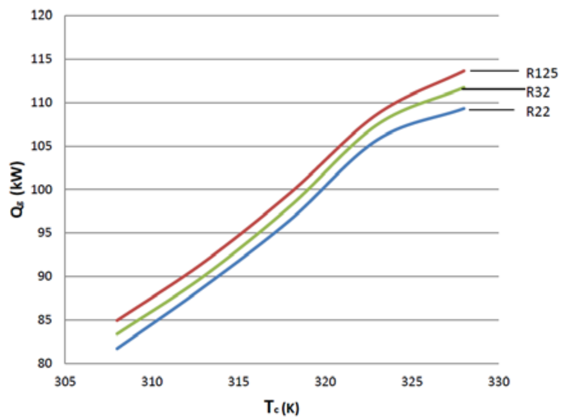

Fig. 11 Generator heat input(Q_g) v/s Condenser temperature (T_c) :- R134a-DMETEG

The same trend was seen when R134a-DMETEG solution was used in the absorption stage of the system.

F.Variation in heat input required in the generator (Q_g) while varying the generator temperature.

This analysis was performed to see the variation in Q_g while varying the generator temperature. Results show that the heat input required in the generator (Q_g) decreases with the increase in generator temperature. Therefore, high generator temperature is preferred. Results show that when R125 is used in the compression stage then it requires maximum heat input in the generator in the absorption stage. One main reason of preferring high generator temperature is that it leads to high concentration of ammonia vapours at the condenser inlet. Usually, Q_g increases with increase in generator temperature but for ammonia-water refrigerant pair it is exactly opposite. There is a dip of 30 kW as we move from 120K to 170K.

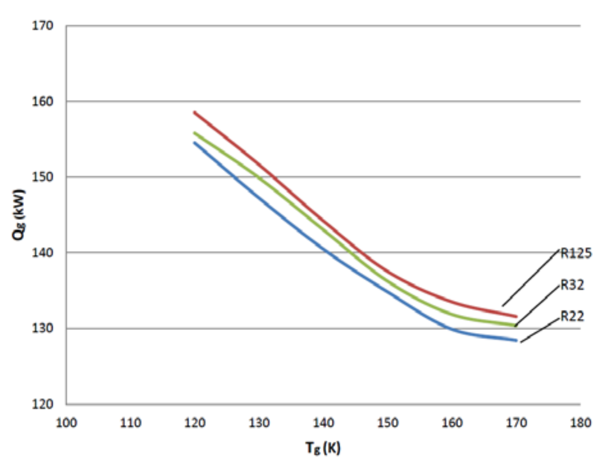

Fig. 12 Generator heat input(Qg) v/s Generator temperature Tg :- R717-Water

In case of R134a-DMETEG refrigerant pair, Q_g increases with increase in generator temperature which is expected because we need higher heat input to maintain the generator at the higher temperature. But this variation in Q_g is very small .The increase in Q_g is only 5kW as we move from 120 K to 170 K.

G. Variation in discharge temperature with respect to the change in intermediate temperature.

This analysis was done to observe the change in refrigerant discharge temperature from the compressor while varying the intermediate temperature in the cascade condenser. The results show that refrigerant R32 has the highest discharge temperature as compared to R125 and R22.

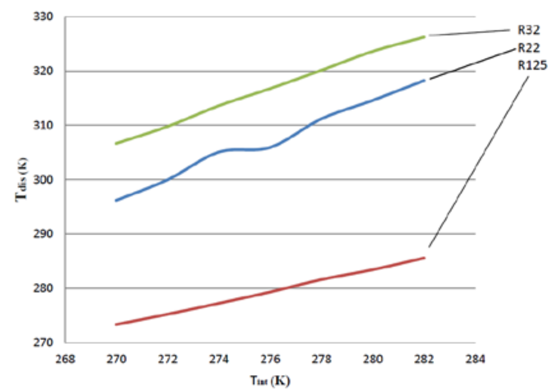

Fig. 13 Intermediate temperature (T_{int}) v/s Compressor discharge temperature (T_{dis})

Conclusion

Even though vapour compression-absorption systems have lesser COPs but they are more economical since energy input in the absorption stage is low grade energy i.e. heat. The present work also concludes that the temperature difference between the refrigerants of the compression stage and the absorption stage is an important parameter. If the temperature difference is too small then the size of the cascade condenser required will be very large leading to design problems and if the temperature difference is very large then there will be a considerable loss in system COP leading to poor system performance. It was found out that R125 is the most compressor friendly refrigerant among all the refrigerants used in this analysis.R125 has low discharge temperatures ;therefore it is safe to use R125 with hermetic compressors

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