COMPARATIVE ANALYSIS OF THERMODYNAMIC PERFORMANCE OF A CASCADE REFRIGERANTION SYSTEM FOR REFRIGERANT COUPLE (R717-R744) &(R290-R744)

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Abstract: Present time Global warming is many effect to environment and uses of High refrigerant use in Refrigerant system more harmful effect on environment. In this project, compare Natural refrigerant pair(R717-R744) & (R290-R744) refrigeration pairs for high output. And best pair to optimize design and operating parameter of the system study operating parameter evaporator temperature effect, condensing temperature effect, temp difference in cascade lower temperature cycle and higher temperature cycle and also exergetic efficiency for both refrigerant pair.It is calculated by computer simulation with the calculation software EES (Engineering Equation Solver).

Keywords: Eco-friendly refrigerant, ozone depletion, global warming, cascade system, Co-efficient of performance.

I INTRODUCTION

Fossil Refrigeration is a technology which regulates and maintains temperatures lower than the surrounding. Among its applications, food storage and air condition systems are the two typical examples. Vapor compression refrigeration dates back to 1834 when Jacob Perkins received a patent for closed cycle ice machine. Since the development of the vapor compression cycle, refrigeration obtained a very fast growth [13]. From chilling and chilled storage to freezing and frozen storage, from refrigerated transport to retail distribution, accompanied by the development of air conditioning technology, refrigeration plays a vital role in the commercial sectors, especially in supermarkets. However, supermarkets are the most energy-intensive types of commercial buildings, which contain heating, cooling, and ventilation (HVAC) as well as refrigeration systems as single integrated facility [7]. Ozone-layer depletion is one of the most critical environmental problems. This problem is engendered by the interaction of destructive chemical substances, such as bromine and chlorine gases, with stratospheric ozone. The presence and stability of harmful substances such as volatile organic compounds, chlorofluorocarbons(CFCs),and hydrochlorofluoro carbons (HCFCs) in the lower atmosphere has increasingly threatened the environment, as well as people's health and safety, over the last two centuries. **Tzong-shing lee et.al.(2006)[6]** Has provided a thermodynamic analysis of carbon dioxide–ammonia (R744–R717) cascade refrigeration system is presented to optimize the design and operating parameters of the system. The design and operating parameters considered in this study include (1) condensing, sub cooling, evaporating and superheating temperatures in the ammonia

(R717) high-temperature circuit, (2) temperature difference in the cascade heat exchanger, and (3) evaporating, superheating, condensing and sub cooling in the carbon dioxide (R744) low-temperature circuit. **H.M. Getu et.al.(2007)** [3] This paper talks about the multi-goals streamlining of a course refrigeration framework utilizing refrigerant C3H8 in high temperature circuits (HTC) and a blend of C2H6/CO2 in low temperature circuits (LTC).here are two target works that ought to be all the while upgraded including the all out yearly cost which comprises of the capital and operational expense and the absolute Exergy decimation of the framework. **Nasruddin a et.al.(2016)**[13] Thermodynamically analyzed R507A-R23 cascade refrigeration system to optimize the design and operating parameters of the system. The design and operating parameters include: Condensing, evaporating, sub cooling and superheating temperatures in the high temperature circuit, temperature difference in the cascade heat exchanger, and condensing, evaporating, sub cooling and superheating temperatures in the low temperature circuit. **A.D.Parekh et.al. (2011)**[9] .A carbon dioxide-propane(R744-R290) cascade system was studied by an optimum cascade evaporating temperature of R744 in the high-temp circuit was determined heating applications **Bhattacharyya(2005)**[2] **.** the experiment on a cascade refrigeration cycle using R744/R290 blends (71/29, mole fraction) as the low temperature fluid was first carried out in this paper. The performance of R744/R290 blends (71/29, mole fraction) and R13 were compared under the same ($T_{in\_evap\_ref}$) while the condensing and evaporating pressure of HTC was kept invariable. R744/R290 blends (71/29, mole fraction) is a natural working fluid. Its ODP is zero and GWP is smaller than 20.R744/R290 blends (71/29, mole fraction) had more refrigeration capacity than R13, and COP was also higher than that of R13.R744/R290 blends (71/29, mole fraction) had good cycle performance compared with R13. It can be considered as a good substitute for R13 when the temperature of evaporator is higher than 201 K. **Baolian Niu et.al (2005)**[11].Analysis of 404a/508b cascade refrigeration cycle for low temperature. The analysis includes three basic parameters as: Evaporator temperature(Te),Condenser temperature (T_c), and temperature difference in cascade condenser (D_t) .These parameters are varied one by one up to a limited range keeping other parameters constant and the effect of these parameters on system COP , exegetic efficiency, mass flow ratio etc is analyzed. Analysis results also give the optimum values of the evaporator, condenser and cascade condenser temperature. **Devanshu Pyasi et.al(2011)** [8] **.** Experimental analysis of R-450A and R-513A as replacements of R-134a and R-507A in a medium temperature commercial refrigeration system .they concluded from this experimentation that the use of the reduced GWP refrigerants R-513A (GWP=573) and R-450A (GWP=547) as drop-in replacements of R-134a (GWP=1301) is possible. They offer a slight increase on energy consumption, but they will offer important reductions of the direct emissions **Rodrigo Llopis (2017).**[12]

II. CASCADE SYSTEM[6]

A CO_2/NH_3 cascade refrigeration system uses ammonia and carbon dioxide as refrigerants in high- and low temperature circuits, respectively. Carbon dioxide (R744) was a commonly used natural refrigerant in vapour compression refrigeration systems for over 130 years, but it has only been fully exploited during the last decade. Some of the characteristics of CO_2 make it a good alternative to ammonia for use in large-scale refrigeration plants operated at low temperatures. The most obvious advantages of carbon dioxide are that it is nontoxic, incombustible and has no odor. Moreover, as compared with ammonia two-stage refrigeration system, the CO_2/NH_3 cascade refrigeration system has a significantly lower charge amount of ammonia, and the COP of the cascade system exceeds that of a two-stage system at low temperatures. Therefore, many investigations of the CO_2/NH_3 cascade refrigeration system are attracting attention.[6]

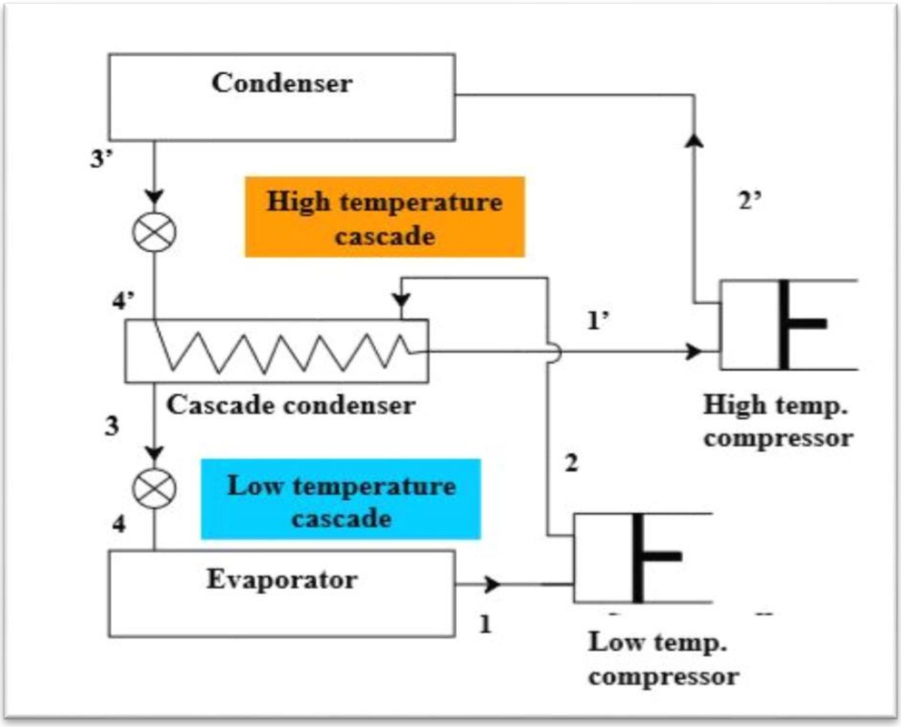

Fig.1: CO_2/NH_3 cascade refrigeration system[18]

Figure 1 schematically depicts a CO_2/NH_3 cascade refrigeration system. **Figure 2** presents the corresponding temperature entropy and pressure enthalpy diagrams. This refrigeration system comprises two separate refrigeration circuits- the high-temperature circuit (HTC) and the low-temperature circuit (LTC). Ammonia is the refrigerant in HTC, whereas carbon dioxide is the refrigerant in LTC. The circuits are thermally connected to each other through a cascade-condenser, which acts as an evaporator for the HTC and a condenser for the LTC.**Figure 2** shows that the condensing and evaporating pressures in the NH_3 circuit are both lower than those in the CO_2 circuit. Therefore, the NH_3 circuit is called the high-temperature circuit (HTC) rather than the high-pressure circuit, and the CO_2 circuit is called the low-temperature circuit (LTC) rather than the low-pressure circuit. Fig.4.1 indicates that the condenser in this cascade refrigeration system rejects a heat of Q_H from the condenser at condensing temperature of T_C, to its warm coolant or environment at temperature of T_0.

The evaporator of this cascade system absorbs a refrigerated load Q_L from the cold refrigerated space at T_{CL} to the evaporating temperature T_E. The heat absorbed by the evaporator of the LTC plus the work input to the LTC compressor equals the heat absorbed by the evaporator of the HTC.

T_{MC} and T_{ME} represent the condensing and evaporating temperatures of the cascade condenser, respectively. $\Delta T = (T_{MC} - T_{ME})$ represents the difference between the condensing temperature of LTC and the evaporating temperature of HTC. The evaporating temperature T_E, the condensing temperature T_C, and the temperature difference in the cascade-condenser are three important design parameters of a CO_2/NH_3 cascade refrigeration system.

The main components of the test CO_2 /NH_3 cascade refrigeration system reported are NH_3 compressor, evaporative condenser, NH_3 expansion device, cascade-condenser, CO_2 expansion device and CO_2 compressor.

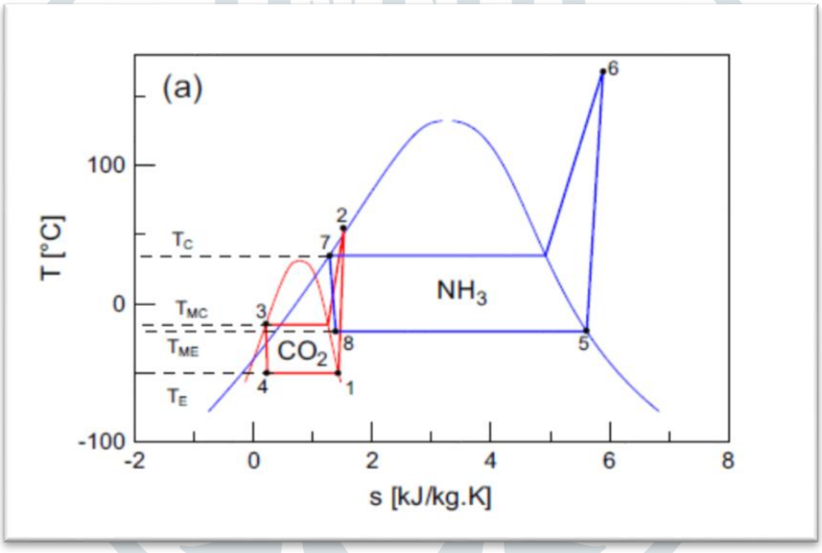

Figure 2 T-s diagram of CO_2/NH_3 cascade refrigeration system [6]

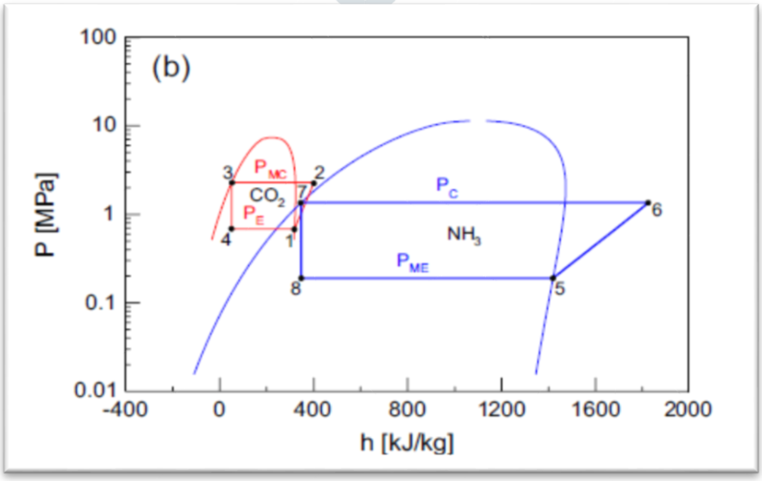

Figure 3 P-h and of CO_2/NH_3 cascade refrigeration system[6]

III. THERMODYNAMIC ANALYSIS OF A CASCADE

i. The thermodynamic analysis of the two-stage cascade refrigeration system is performed based on the following general assumptions:
ii. Adiabatic compression with an isentropic efficiency of **0.7** for both high- and low-temperature compressors,
iii. Negligible pressure and heat losses/gains in the pipe networks or system components,
iv. Isenthalpic expansion of refrigerants in expansion valves, and
v. Negligible changes in kinetic and potential energy.

The thermo physical and environmental properties of refrigerants used in this analysis are given in **table-2.** Thermodynamic analysis has been done using a software package called as Energy Equation Solver.

COMPONENT	MASS	ENERGY	Exergy
HTC compressor	$\dot{m}_H = \dot{m}_5 = \dot{m}_6$	$\dot{W}_H = \dot{m}_H\ (h_6 - h_5)\ /\eta_{is}$	$X_{des} = \dot{m}_H\ (\psi_5 - \psi_6) - \dot{W}_H$
Condenser	$\dot{m}_H = \dot{m}_6 = \dot{m}_7$	$\dot{Q}_H = \dot{m}_H\ (h_6 - h_7)$	$X_{des} = (1-\frac{\dot{T}_o}{\dot{T}_c})\ \dot{Q}_H + \dot{m}_H\ (\psi_6 - \psi_7)$
HTC exp. device	$\dot{m}_H = \dot{m}_7 = \dot{m}_8$	$h_7 = h_8$	$X_{des} = \dot{m}_H\ (\psi_7 - \psi_8)$
Cascade condenser	$\dot{m}_H = \dot{m}_5 = \dot{m}_8$ $\dot{m}_L = \dot{m}_2 = \dot{m}_3$	$\dot{Q}_{CAS} = \dot{m}_H\ (h_5 - h_8) = \dot{m}_L\ (h_2 - h_3)$	$X_{des} = \dot{m}_L(\psi_2 - \psi_3) + \dot{m}_H\ (\psi_8 - \psi_5)$
LTC compressor	$\dot{m}_L = \dot{m}_1 = \dot{m}_2$	$\dot{W}_L = \dot{m}_L\ (h_2 - h_1)\ /\eta_{is}$	$X_{des} = \dot{m}_L(\psi_1 - \psi_2) - \dot{W}_L$
LTC exp. device	$\dot{m}_L = \dot{m}_3 = \dot{m}_4$	$h_3 = h_4$	$X_{des} = \dot{m}_L(\psi_3 - \psi_4)$
Evaporator	$\dot{m}_L = \dot{m}_1 = \dot{m}_4$	$\dot{Q}_E = \dot{m}_L\ (h_1 - h_4)$	$X_{des} = (1-\frac{\dot{T}_o}{\dot{T}_E})\ \dot{Q}_L + \dot{m}_L(\psi_4 - \psi_1)$

Table-2 The balance equation for CO_2/NH_3 cascade refrigeration system[8]

The capacity of the evaporator is defined by

$$\dot{Q}_E = \dot{m}_L\ (h_1 - h_4) \quad (1)$$

Compressor power consumption for high-temperature circuit is given by:

$$\dot{W}_H = \dot{m}_H \,(h_6 - h_5)\,/\eta_{is} \qquad (2)$$

Whereas for low-temperature circuit, it is given by:

$$\dot{W}_L = \dot{m}_L \,(h_2 - h_1)\,/\eta_{is} \qquad (3)$$

The rate of heat transfer in the cascade heat exchanger is determined from:

$$\dot{Q}_{CAS} = \dot{m}_H \,(h_5 - h_8) = \dot{m}_L \,(h_2 - h_3) \qquad (4)$$

The mass flow ratio can be derived from Eq. (4):

$$\dot{m}_H \,/\, \dot{m}_L = \frac{(h_2 - h_3)}{(h_5 - h_8)} \qquad (5)$$

The rate of heat rejection by the air-cooled condenser is given by:

$$\dot{Q}_H = \dot{m}_H \,(h_6 - h_7) \qquad (6)$$

The COP of low-temperature circuit, it is given by:

$$COP_{LTC} = \frac{\dot{Q}_E}{\dot{W}_L} \qquad (7)$$

The COP of high-temperature circuit, it is given by:

$$COP_{HTC} = \frac{\dot{Q}_H}{\dot{W}_H} \qquad (8)$$

The overall COP of the system is determined by:

$$\mathrm{COP} = \frac{\dot{Q}_E}{\dot{W}_H + \dot{W}_L} \qquad (9)$$

The COP can also be defined solely as a function of specific enthalpies by substituting Eqs. (1)– (5) In Eq. (9).

$$\mathrm{COP} = \frac{(h_1 - h_4)(h_5 - h_8)}{(h_5 - h_8)(h_2 - h_1) + (h_2 - h_3)\,(h_6 - h_5)} \qquad (10)$$

OR

The overall COP of the system is determined by:

$$\mathrm{COP} = \frac{(COP_{LTC})(COP_{HTC})}{1 + COP_{LTC} + COP_{HTC}} \qquad (11)$$

The Second law efficiency of the whole system is defined as the ratio of the actual cop to the ideal cop Carnot, which is

$$\eta_{II} = \frac{COP}{C\dot{O}P_{Carnot}} \qquad (12)$$

Where, $C\dot{O}P_{Carnot} = \frac{T_E}{\dot{T}_C + \dot{T}_E}$

IV. RESULTS AND DISCUSSION OF THE SYSTEM

The equations of the mathematical model reveal that the system's COP and can be expressed as a function of six design/operating parameters, as shown in the equation (COP, R_E, WD. m_H/m_L) = f ($T_{Eva\ CO2}$, $T_{Cond\ NH3}$, ΔT, $T_{Cond\ CO2}$, $\eta_{Comp\ CO2}$, $\eta_{Comp\ NH3}$)

To determine the influence of each design and operating parameter on the system's COP a parametric study was done. The ranges of values for each parameter comprise the specific intervals of interest. Evaporator temperature in LT circuit T_E, Condenser temperature in HT circuit T_C and Condenser temperature in LT circuit T_{MC} were varied from -35 to 10°C, from 60 to 15°C and from -30 to 15°C, respectively, at intervals of 5°C. The temperature difference in the cascade heat exchanger was varied from 1 to 18°C.

To solve each case derived from the parametric study specific software Engineering Equation Solver (EES) has been employed. The thermodynamic properties of CO_2,NH_3 ,C_3H_8 were calculated from EES.

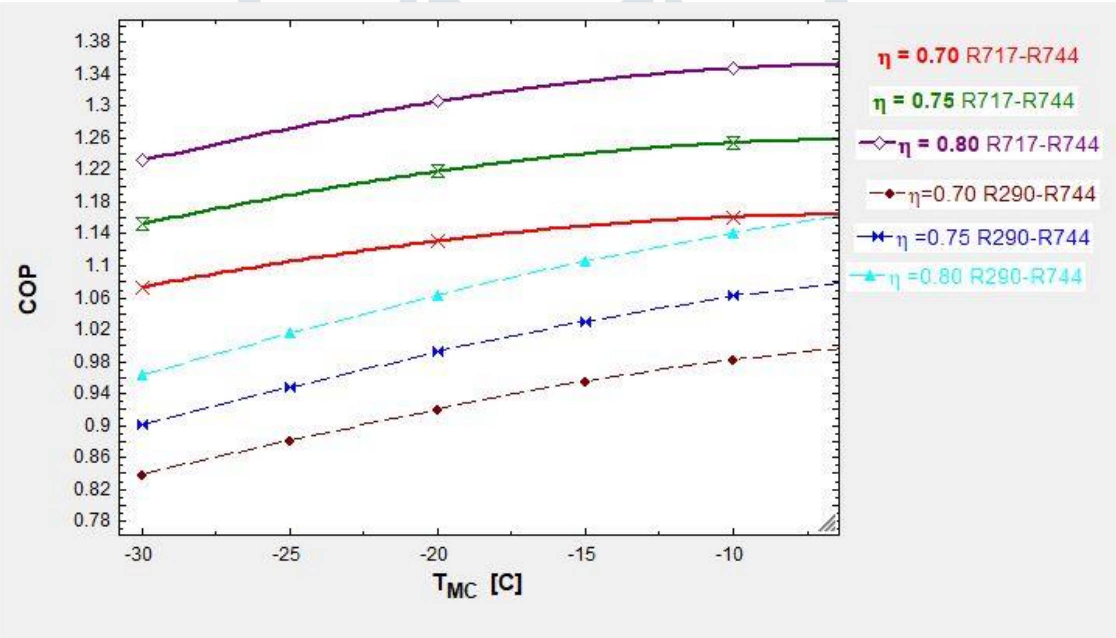

Figure 4 Effect of T_{MC} on COP at different isentropic efficiency of compressors for(R7171-R7444) and (R290-R744)

Figure 5 depicts variation in COP with T_{MC} for different isentropic efficiency of both HT and LT compressors. Maximum COP shifts upward proportionally with an increase in isentropic efficiency. When isentropic efficiency of both LTC and HTC compressor is increased from 0.70 to 0.80 maximum COP is increased 14% for Refrigerant pair (R7171-R744) and Refrigerant pair (R290-R744) also increased 12% but compare to (R717-R744) is Lower.

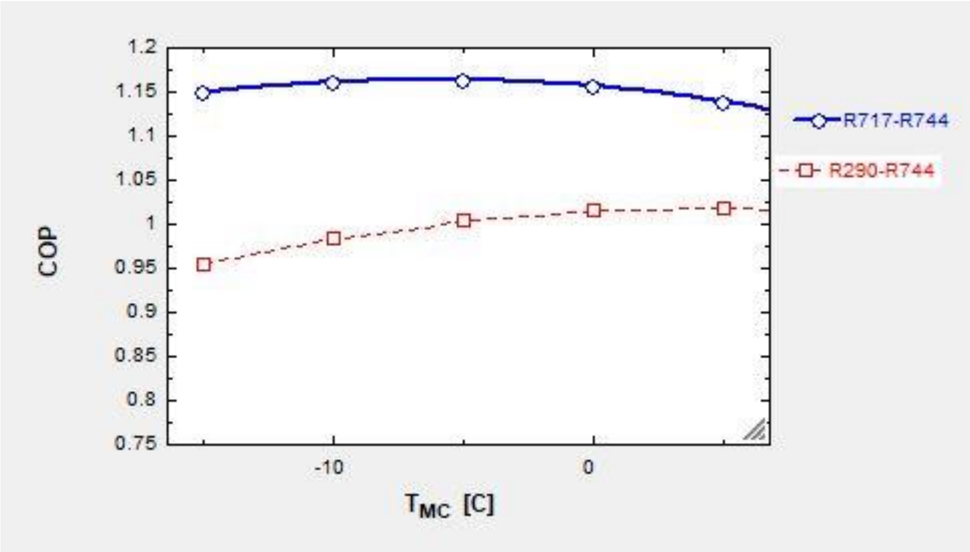

Figure 5 Effect of condensing temperature on COP both cycle (R717-R744) and (R290-R744)

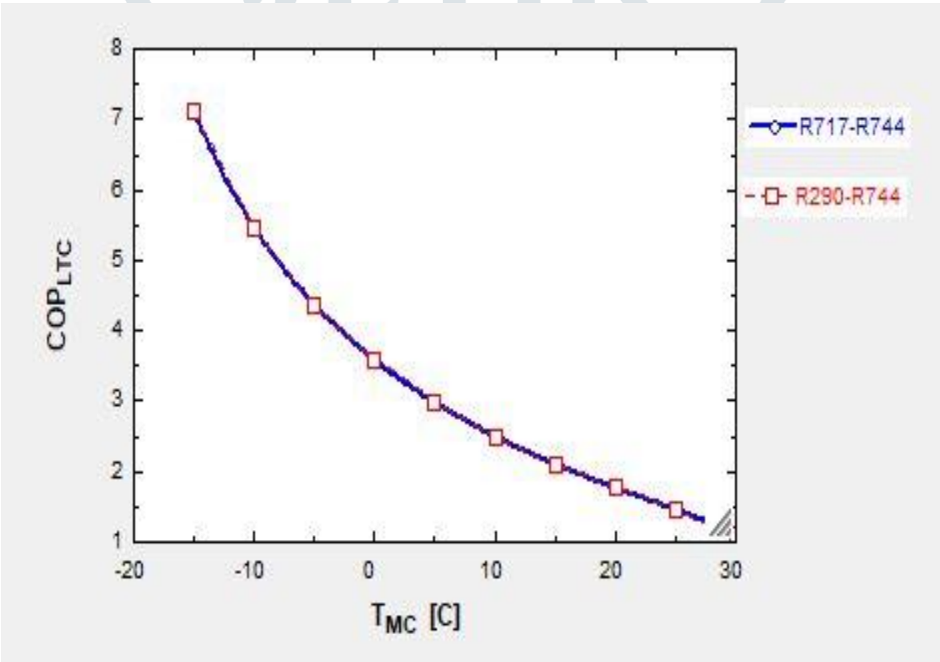

Figure 6 Effect of condensing temperature on COP of LTC for both cycle (R717-R744) and (R290-R744)

Figure 6 displays the effect of LT circuit condensing temperature T_{MC} on COP of LT circuit. The COP of LT circuit decreased with T_{MC}. When T_{MC} is change from -15 °C to 30°C, COP of LT circuit is decreased by 59% for both refrigerant pair.

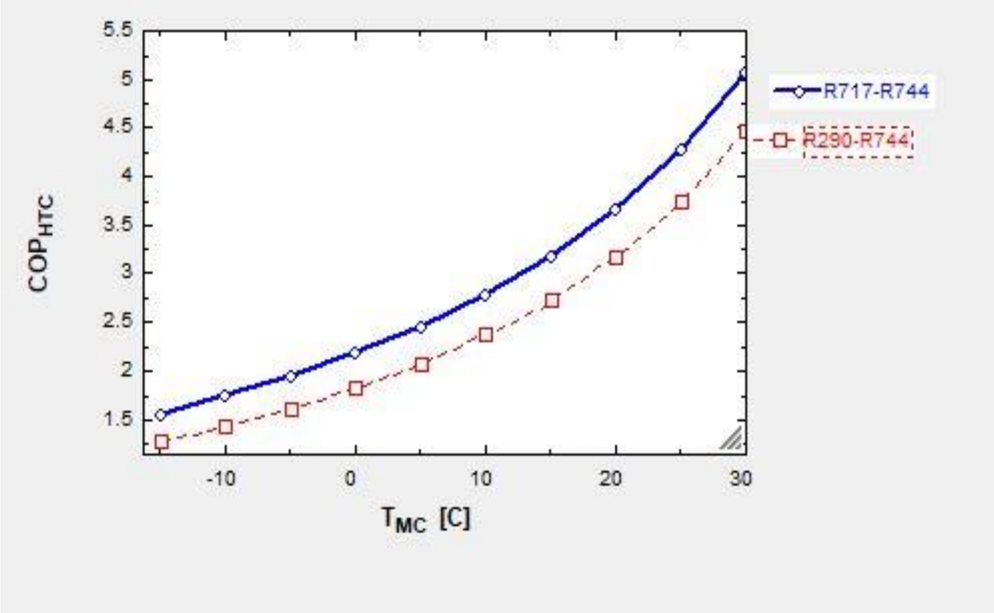

Figure 7 Effect of condensing temperature on COP of HTC for both cycle (R717-R744) and (R290-R744)

Figure 7 displays the effect of condensing temperature T_{MC} on COP of HT circuit. The COP of HT circuit increased with T_{MC}. When T_{MC} is change from -15 °C to 30°C, COP of HT circuit is increase by 39% for refrigerant pair (R717-R744) and also increase COP(31%) in refrigerant pair (R290-R744) but compare to refrigerant pair (R717-R744) is lower.

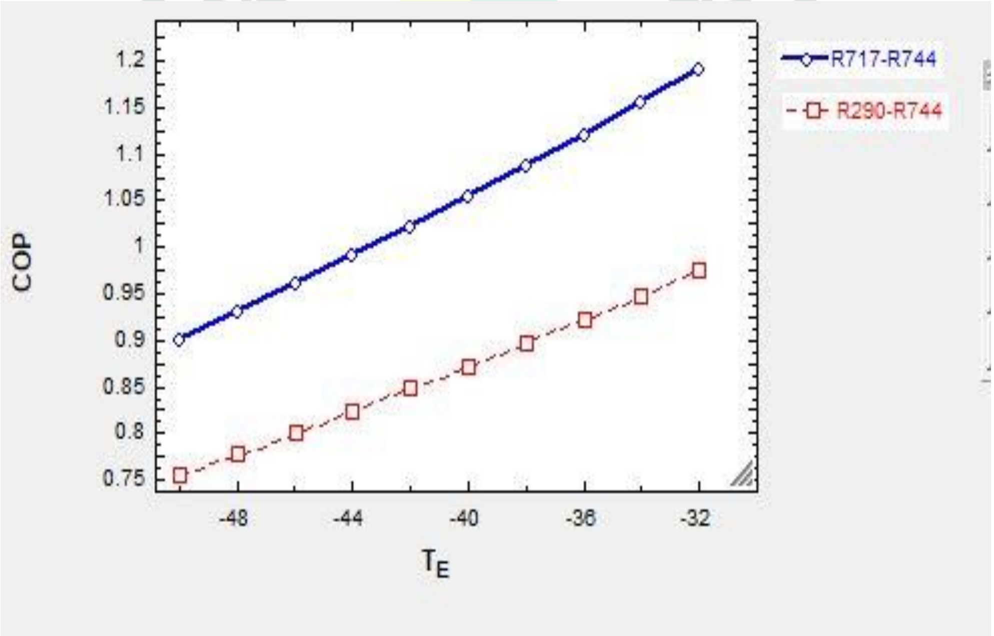

Figure 8 Effect of T_E on for both cycle (R717-R744) and (R290-R744)

Figure 8 present the effect of the evaporating temperature T_E on the COP when evaporating temperature T_E is increase so cop is also increase in both refrigerant pair. But refrigerant pair (R717-R744) cop increases

2.9% and refrigerant pair (R290-R744) cop increase 2%. Refrigerant pair(R717-R744) cop more increase compare to (R290-R744).

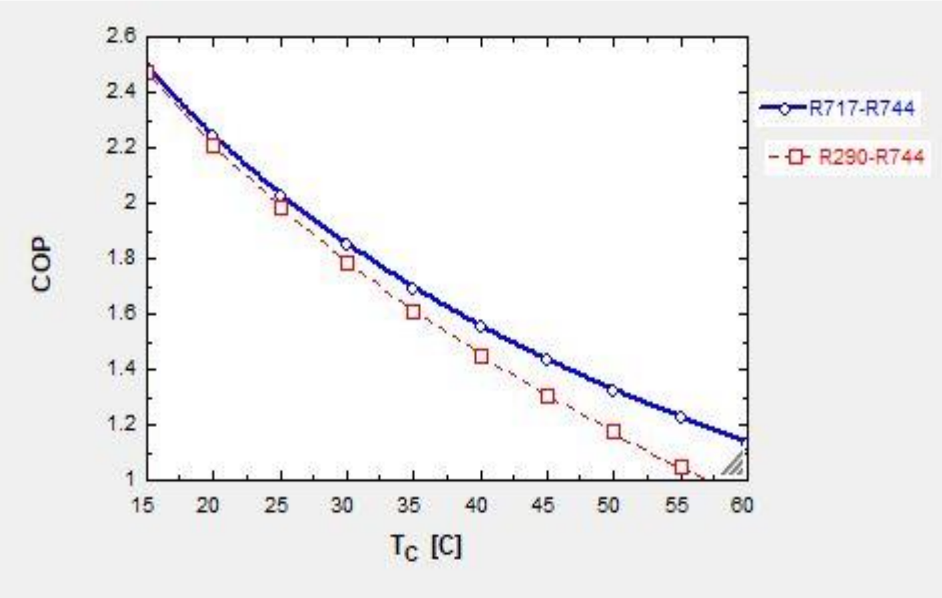

Figure 9 Effect of TC on for both cycle (R717-R744) and (R290-R744)

Figure 9 present the effect of the condenser temperature T_C on the COP when condenser temperature T_C is increase so cop is decrease in both refrigerant pair. But refrigerant pair (R717-R744) cop decreases 13% and refrigerant pair (R290-R744) cop decrease 15%. Refrigerant pair (R290-R744) cop more decrease compare to (R717-R744).

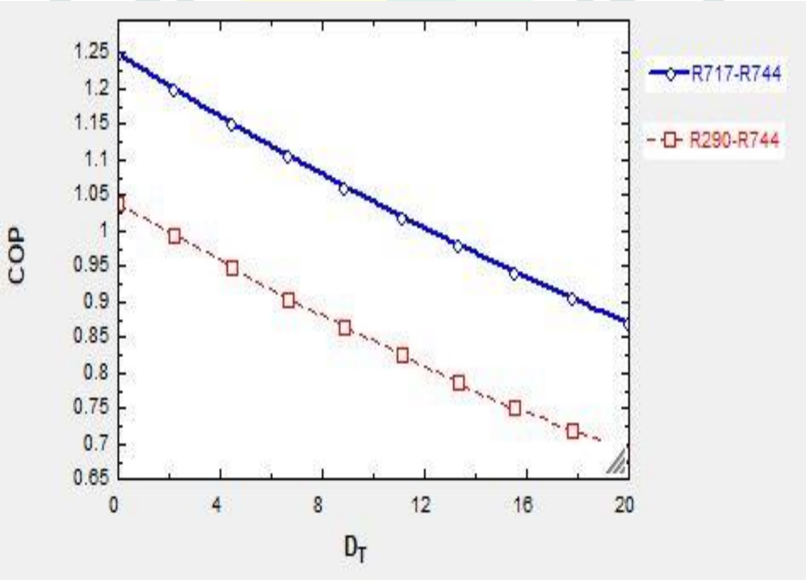

Figure 10 Effect of ΔT on for both cycle (R717-R744) and (R290-R744)

Figure 10 present the effect of the temperature difference ΔT on the COP when temperature difference ΔT is increase so cop is decrease in both refrigerant pair. But refrigerant pair (R717-R744) cop decreases

3.5% and refrigerant pair (R290-R744) cop decrease 3.5%. Refrigerant pair (R290-R744) and refrigerant pair (R717-R744) both pair cop is equal decreases.

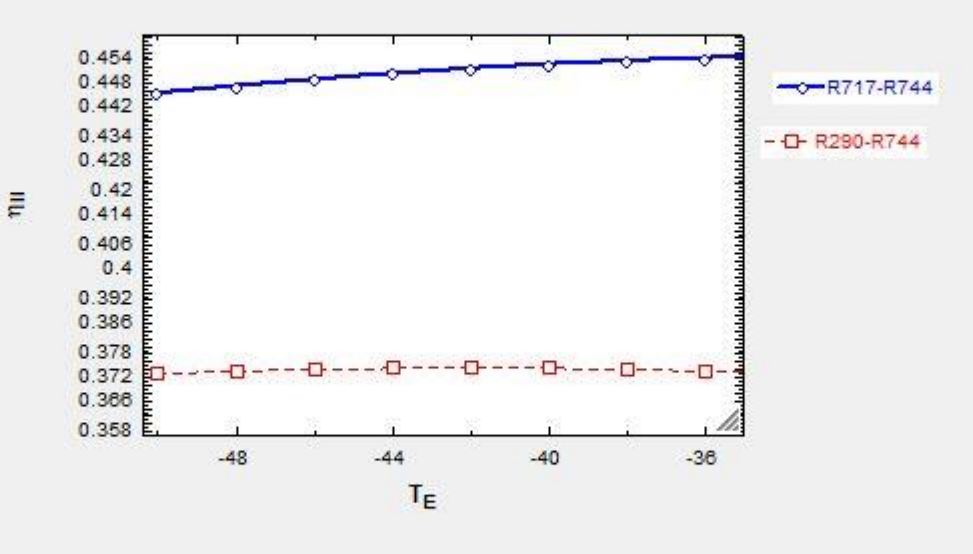

Figure 11 Effect of LTC evaporator T_E on exergetic efficiency for both pair

Figure 11 depicts, while varying the low circuit evaporator temperature (T_E) and keeping other parameters constant Exergy efficiency increases with very small amount from (0.445 to 0.455)(1%) with increase in evaporator temperature for refrigerant pair(R-717-R744). But refrigerant pair (R290-R744) very small increases(0.3730-0.3736)(0.06%) compare to (R717-R744) is less increases.

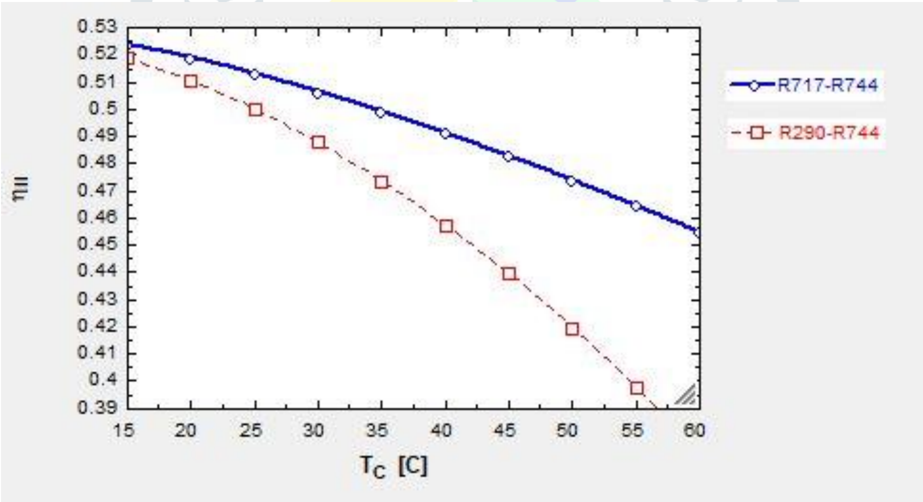

Figure 12 Effect of HTC condenser T_C on exergetic efficiency for both pair

Figure 12 shows the Exergy efficiency η_{II} decreases when high temperature circuit condenser temperature T_C is varied keeping other parameters constant. The exegetic efficiency decrease from (0.524 to 0.455)(6.9%) for refrigerant pair(R717-R744)and other refrigerant pair (R290-R744) is more decreases (0.524 to 0.391)(13%) compare to Refrigerant pair (R717-R744).

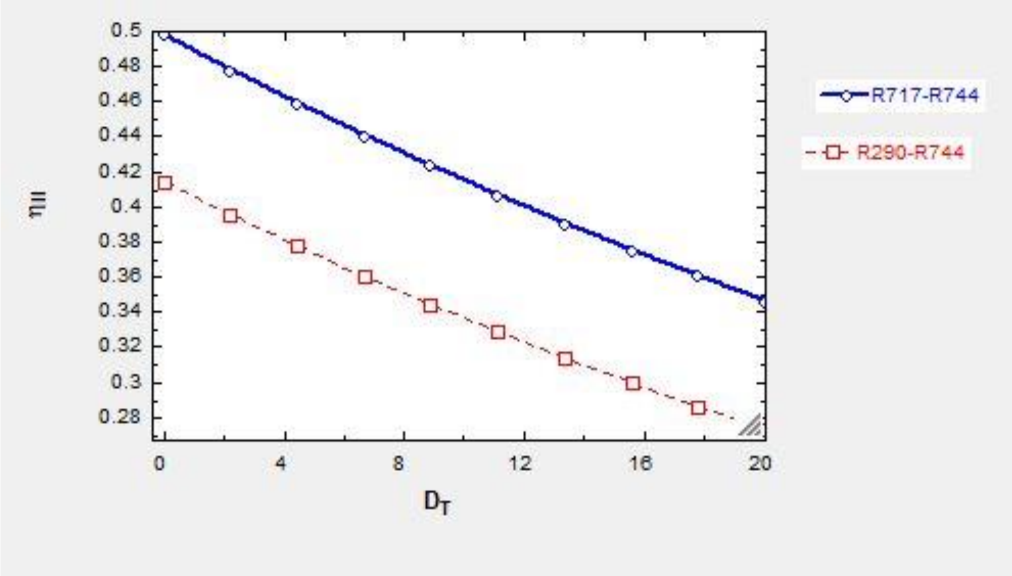

Figure 13 Effect of temperature difference ΔT on exergetic efficiency[30]

Figure 13 depicts, while varying the temperature difference ΔT and keeping other parameters constant Exergy efficiency decreases with amount from (0.498 to 0.360) (13.8%) with increase in temperature difference ΔT for Refrigerant pair(R717-R744) and other refrigerant pair exergetic Efficiency decreases(0.421-0.280)(14.1 %) more compare to refrigerant Pair (R717-R744).

V. CONCLUSION

From the thermodynamic analysis of the (R717-R744) pair and (R290-R744)

Cascade refrigeration systems following results are obtain:

1. When condensing Temperature (T_{MC}) increases then isentropic efficiency increases in both pair but refrigerant pair (R290-R744) is less compare to refrigerant pair(R717-R744).
2. Increases T_{MC} then COP is increases but refrigerant pair (R717-R744) is COP more increase Compare to refrigerant pair (R290-R744).
3. When T_{MC} is increases both pair COP_{LTC} is same decreases.
4. When T_{MC} is increases both pair COP_{HTC} is increases but (R290-R744) pair is low increase compare to (R717-R744) pair.
5. When Evaporator temperature T_E increase both pair COP increase but refrigerant pair(R717-R744) is more increase compare to refrigerant pair (R290-R744).
6. When Condenser temperature T_C is increase both pair COP is decrease but refrigerant pair (R290-R744) is more decrease.
7. When temperature difference ΔT increase both pair COP is same decrease.
8. When Evaporator temperature T_E increase both pair very small amount of exergetic efficiency increase but refrigerant pair (R717-R744) more amount increase to other pair.

9. When Condenser temperature T_C is increase both pair exergetic efficiency is decrease but refrigerant pair (R290-R744) is more decreases compare to other pair.
10. Temperature difference ΔT increase both pair exergetic efficiency decrease but refrigerant pair (R290-R744) is more decrease other pair.

VIII. ACKNOWLEDGEMENT

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