WORK STUDY ON LOW TEMPERATURE (CASCADE) REFRIGERATION SYSTEM

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Abstract - *Ammonia/CO_2 mixture is presently considered to be the best for cascade refrigeration up to 216.58 K. However to approach temperatures lower than 216.58 K the mixture of CO_2 with other refrigerants are required. The aim of present work was to study the possibility of using carbon dioxide mixtures in those applications where temperatures below triple point (216.58 K) are needed. The blends considered for analysis purpose are R744/R125, R744/R41, R744/R32, and R744/R23. The analysis of cascade refrigeration cycle has been carried out with the help of a computer program developed using Engineering Equation Solver (EES). The composition of CO_2 is varied from a mole fraction of 0.1 to 0.8 along with R23, R32, R41 and R125. The mixtures properties of the investigated blends (R744/R125, R744/R41, R744/R32, and R744/R23) were calculated by Ref Prop 7.0 and used in EES program. Ammonia is used as refrigerant in high-temperature-circuit. Blend of CO_2 with R125 having a mole fraction ratio of 0.3/0.7 is the best among all the blends considered as it offers highest COP values. Up to mole fraction ratio 0.5/0.5, CO_2/R125 blend is better than other blends for same mole fraction ratio. At mole fraction ratio of 0.8/0.2, the blends R744/R125, R744/R41and R744/R32 have almost same values of COP and that of R744/ R32 blend is lowest. Hence R744 blends are an attractive option for the low-temperature circuit of cascade systems operating at temperatures approaching 200 K.*

***Key Words*: Cascade System, Mole Fraction, R744 Blends, COP Values, Triple Point,**

1. INTRODUCTION

Refrigeration is defined as -the transfer of heat from a lower temperature region to a higher temperature one. Refrigeration devices that produce refrigeration are heat pumps, refrigerators, automotive air-conditioners, and residential/commercial air-conditioners. All of these devices have one thing is common, to reduce the temperature of an enclosed environment. Vapor compression cycle can be used in temperature range -10 to -30°C easily. And low-temperature refrigeration systems are typically required in the temperature range from −30°C to −100°C for applications in food, pharmaceutical, chemical, and other industries, e.g., blast freezing, cold storages, liquefaction of gases such as natural gas, etc. At such low temperatures, single-stage compression systems with reciprocating compressors are generally not feasible due to high pressure ratios. A high pressure ratio implies high discharge and oil temperatures and low volumetric efficiencies and, hence, low COP values. Screw and scroll compressors have relatively flat volumetric efficiency curves and have been reported to achieve temperatures as low as –40°C to –50°C in single-stage systems (Stegmann, 2000). Further, the use of a single refrigerant over such a wide range of temperature results in either extremely low pressures in the evaporator and large suction volumes or extremely high pressures in the condenser. To increase volumetric efficiency and refrigerating effect and to reduce power consumption, multistage with intercooling is often employed (Bansal and Jain, 2006).

J. Sarkar and S. Bhattacharya (2008) did their research work on assessment of blends of CO_2 with Butane and Isobutene as working fluid in cascade system. Their research work comes to the result that due to gliding temperature during evaporation and condensation the zoetrope blends instead of pure container can be employed very effectively in heat pumps for variable temperature or simultaneously cooling and heating applications. The blend R744/R600a can be the best alternative refrigerant to R114 for high temperature heating due to superior COP (more than twice) over R600 and R600a. **J.Alberto Dopazo, Jose F. Seara (2010);** evaluated experimentally the design, construction of an experimentally prototype of a CO_2-NH_3 cascade refrigeration system to apply horizontal plate freezer of 9KW of normal refrigeration capacity at -50*C evaporating temperature, the analysis shows that the overall efficiency of the system will be increased up to 19.5% higher than the common double stage refrigeration process at -40*C evaporating temperature.

1.1 Thermal Performance Analysis of Cascade System

Fig.1 schematically represents a cascade refrigeration system. Fig.2 represents the corresponding 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 blend with HFCs (R-23, R-32, R-41 and R-125) are the refrigerants 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.

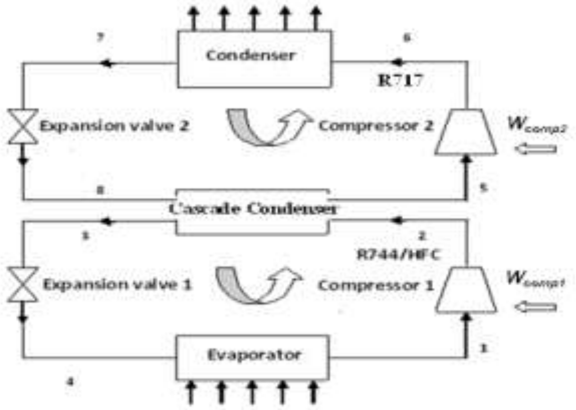

Fig.1 Schematic representation of two stage cascade refrigeration cycle

The condenser in this cascade refrigeration system rejects a heat of QC from the condenser at condensing temperature of TC, to its warm coolant or environment at temperature of T0. The evaporator of this cascade system absorbs a refrigerated load QE from the cold refrigerated space at evaporating temperature T_E. The heat rejected by condenser of LTC 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.

Approach represents the difference between the condensing temperature of LTC and the evaporating temperature of HTC. The evaporating temperature TE, the condensing temperature TC, and the temperature difference in the cascade-condenser are three important design parameters of a cascade refrigeration system.

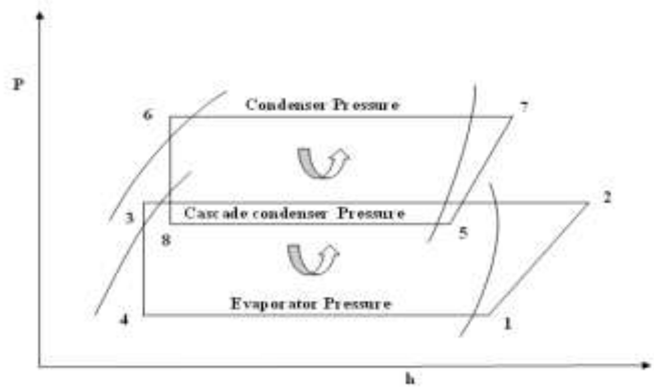

Fig.2 P-h diagram of cascade refrigeration system

1.2 Thermodynamic Analysis

In the present work, a parametric study with fixed mass flow rate in LTC, and various condensing temperature, evaporating temperature and approach in the cascade-condenser have been conducted to determine the optimum condensing temperature of the cascade- condenser.

The following assumptions are made to simplify the thermodynamic analysis, including energy analysis.

1. All components are assumed to be a steady-state and steady-flow process. The changes in the potential and the kinetic energy of the components are negligible.
2. The high and low-temperature circuit compressors are non-isentropic.
3. The heat loss and pressure drops in the piping connecting the components are negligible.
4. All throttling devices are isenthalpic.

Based on the assumptions mentioned above, the equations for mass and energy balance are written for each component. Each component in the cascade refrigeration system, shown in Fig.7 considered as a control volume.

Mass balance: - $\sum_{in} m_r = \sum_{out} m_r$ (1)

Energy balance:-

$$Q - W + \sum_{in} m_r h - \sum_{out} m_r h = 0 \quad (2)$$

Energy changes in each component of cascade refrigeration cycle are as follows:-

Evaporator: evaporator is a heat exchanger which abstract heat from the cold room and this heat is called refrigeration effect, which is given by:-

$$QE = m_{r1} (h_1-h_4) \quad (3)$$

Compressor: compressor is a work absorbing device in which, the isentropic work input to the compressor is expressed as

In compressor 1:- $W_{comp1} = m_{r1} (h_2-h_1)$ (4)

In compressor 2:- $W_{comp2} = m_{r2} (h_6-h_5)$ (5)

Condenser: condenser is a heat exchanger in which heat rejected by the condenser to the environment or warm coolant is given as:-

$$Q_c: - m_{r2} (h_6-h_7) \quad (6)$$

Expansion Valve: expansion valve is a device in which expansion of refrigerant occurred and enthalpy remains constant.

In expansion valve 1: $h_3 = h_4$ (7)

In expansion valve 2: $h_7 = h_8$ (8)

Cascade condenser: it acts as an evaporator in the high-temperature stage and as a condenser in the low-temperature stage. Cascade condenser is a heat exchanger in which all the heat released by the low-temperature-circuit condenser is rejected to the high- temperature-circuit evaporator. Energy balance in cascade condenser is:-

$$m_{r1}(h_2-h_3) = m_{r2}(h_5-h_8) \quad (9)$$

Where mr_1 = mass flow rate in low temperature circuit

mr_2 = mass flow rate in low temperature circuit

Coefficient of performance:-

The performance parameter, COP of a cascade system is defined as the ratio of the refrigerating effect produced in the evaporator to the total work input to all compressors in the system. It can be express by following expression.

$$COP = Q_E \div (Wcomp_1 + W_{comp2}) \quad (10)$$

Volumetric cooling capacity:-

The volumetric cooling capacity is the cooling capacity per unit volume flow rate at the inlet to the compressor. It can be express by following expression:-

$$VCC = Q_E \div (m_{r1} \times V_S) \quad (11)$$

Where Vs = specific volume at inlet of the compressor

2. Cascade refrigeration system with liquid vapor heat exchanger

Analysis of the cascade Refrigeration System shown in fig.1 and fig.3 has been carried out in this work.

The computer program has been developed in EES (Klein and Alvarado, 2005). The mixtures properties of the investigated blends (R744/R125, R744/R41, R744/R32, and R744/R23) were calculated by Ref Prop 7.0 (Huber et al. 2002 and used in EES program to calculate the performance parameters for a cascade refrigeration cycle. For validation of program, results of the present work are compared with results of Nicola et al. (2005). Comparison of results for carbon dioxide (0.1 mole fraction) blend with R-125 and R-41 are shown in figure 6 and 7. The results from present work are 5% higher than that of Nicola et al. (2005).

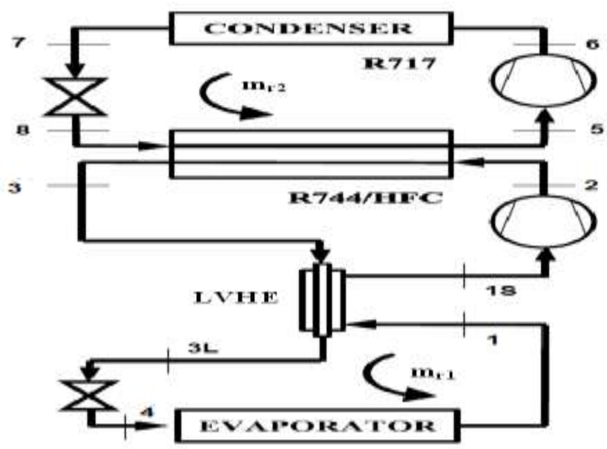

Fig -3 Schematic representation of cascade refrigeration cycle with liquid vapor heat exchanger

The Input Parameters taken for computation of results are given below

Condenser Temperature T_C = 40 to 50°C in steps of 5ºC

Evaporator Temperature T_E = -70°C

Cascade condenser temperature T_{MC} = 10 to -60°C in steps of 5ºC

Approach A = 0, 5°C

Mass flow rate in low temperature circuit m_{r1} = 1 kg/sec

Compressor efficiency in low temperature circuit η_{c1}= 0.7

Compressor efficiency in high temperature circuit η_{c2}= 0.7

Refrigerant in high temperature circuit = R-717

Refrigerant in low temperature circuit= CO_2 blend with HFCs

HFCs used with carbon dioxide: R-23, R-32, R-41, R-125

Sub cooling (in HTC and LTC) = 0, 5°C

Superheating (in HTC and LTC) = 0, 10°C

By carrying out the thermodynamic analysis of the system for the conditions stated above the values at various state points of the cascade refrigeration cycle have been obtained. The computer program for the thermodynamic analysis of the system developed in EES has been given in Appendix A along with its flow diagram for computation procedure.

In the present work, following parameters have been computed.

- Total compressor work.

- Coefficient of performance of the cascade refrigeration cycle.
- Volumetric cooling capacity of the compressor.
- Ratio of mass flow rate in HTC and LTC.

Based on above we have computed optimal cascade condenser temperature.

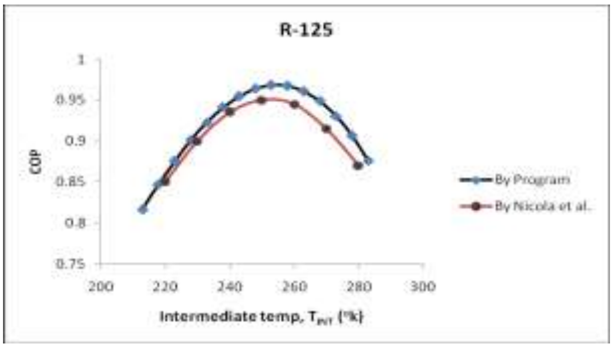

Fig -4 comparison of results with Nicola et al. results for R-125

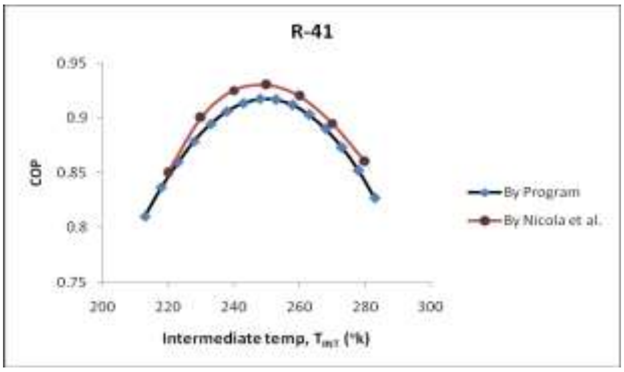

Fig -5 comparison of results with Nicola et al. results for R-41

3. Result and discussion

Figs 6 to 8 shows the variation of COP for cascade refrigeration cycle with varying intermediate temperature for carbon dioxide blend with HFCs (R-23, R-32, R-41 and R-125) for different mole fraction. These figures show that the COP increases with increase in intermediate temperature up to a temperature (optimum temperature) after this COP decreases with increase in intermediate temperature at a particular evaporator and condenser temperature. COP for R-125 mixture is highest among all mixtures followed by R-32, R-41 and R-23 respectively. Optimum temperature range for cascade refrigeration system is 240°K to 260°K.

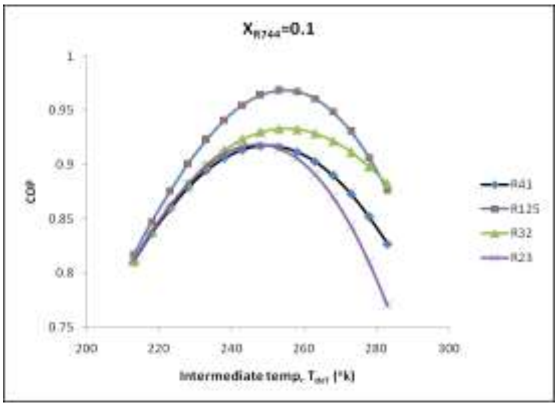

Fig -6 Variation in COP with intermediate temperature for cascade system operating with R744 blend at 0.1 mole fraction as low temperature working fluid with approach 0ºC

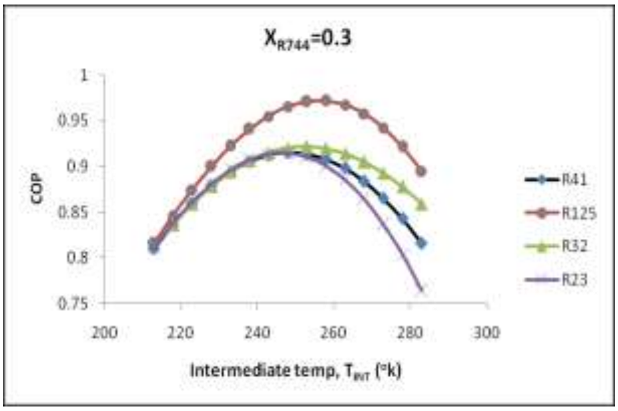

Fig -7 Variation in COP with intermediate temperature for cascade system operating with R744 blend at 0.3 mole fraction as low temperature working fluid with approach 0ºC

3.1 Effect of approach

Approach is temperature difference between high temperature circuit evaporator and low temperature condenser. Figs 6 to 8 (with approach 0) and Figs 9 to 11 (with 5°C approach) show that by taking approach 5°C, COP decreases as around 8%, because of increase in compressor work. Optimum temperature is shifted 2% to higher side by taking approach.

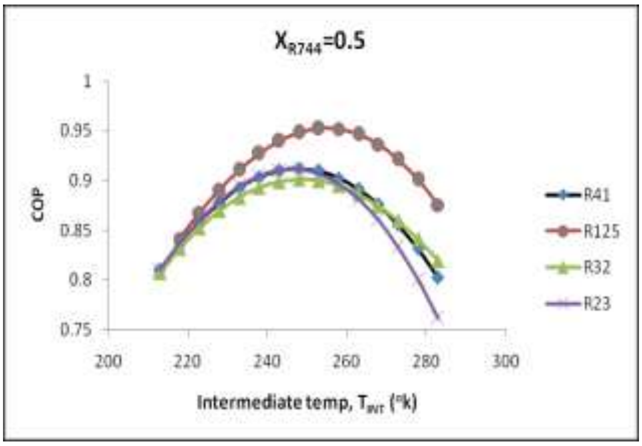

Fig -8 Variation in COP with intermediate temperature for cascade system operating with R744 blend at 0.5 mole fraction as low temperature working fluid with approach 0ºC.

3.2 Effect of superheating (10°C) and subcooling (5°C)

Figs 6 to 8 show the effect of superheating of refrigerant inlet to compressor and subcooling of refrigerant outlet to condenser. By the comparison of figures with superheating and without superheating (Figs 9 to 11), COP increases by superheating and subcooling, this happens because by superheating work done increases and by subcooling refrigerating effect also increases, and combined effect of these is to increase COP. For R-125 mixture (with carbon dioxide) effect of superheating and subcooling on increase in COP is very high. At optimum temperature COP increased around 1% by superheating and subcooling.

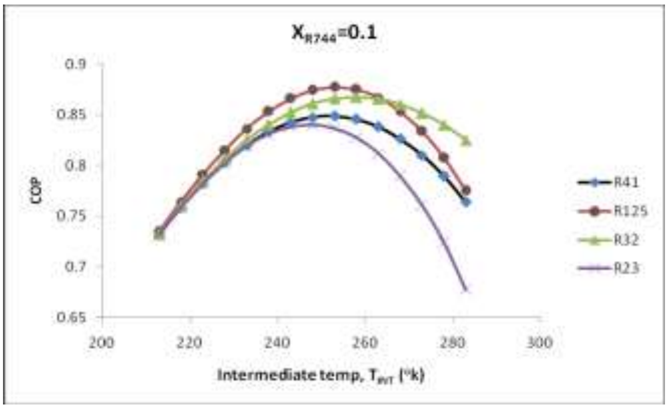

Fig -9: Variation in COP with intermediate temperature for cascade system operating with R744 blend at 0.1 mole fraction as low temperature working fluid with approach 5ºC.

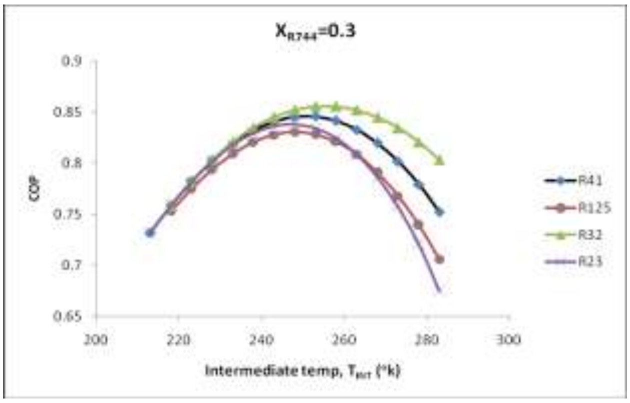

Fig -10: Variation in COP with intermediate temperature for cascade system operating with R744 blend at 0.3 mole fraction as low temperature working fluid with approach 5ºC.

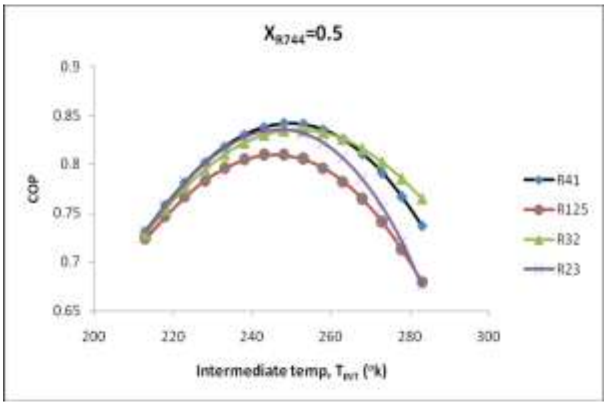

Fig -11: Variation in COP with intermediate temperature for cascade system operating with R744 blend at 0.5 mole fraction as low temperature working fluid with approach 5ºC.

3.3 Effect of mole fraction of carbon dioxide

Fig.12 show the variation of mole fraction of carbon dioxide in mixture with HFCs. Figure show that by increasing mole fraction of carbon dioxide COP affect slightly. For R-23 and R-41, COP decreases with increase in mole fraction of carbon dioxide. For R-125, COP increases up to mole fraction 0.3 and after this decrease up to mole fraction 0.9. For R-32, COP decreases up to mole fraction 0.8 and after this COP increases.

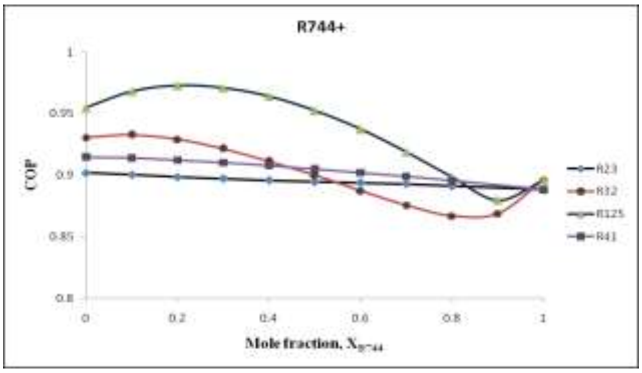

Fig -12: Variation in COP with CO_2 mole fraction for cascade system operating with R744 blends as low temperature working fluid.

4. CONCLUSIONS

In the present study, thermodynamic analysis have been carried out for cascade refrigeration cycle using CO_2/HFC blends as the low-temperature fluid and ammonia as the high- temperature fluid with a view to extending the applicability of carbon dioxide in such systems below its triple point (216.58 K).

The results obtained permit the following remarks:

1. COP increases with increase in intermediate temperature up to a temperature

   (optimum temperature) after this COP decreases with increase in intermediate temperature.

2. COP for R-125 mixture is highest among all mixtures followed by R-32, R-41 and R-23 respectively.

3. Optimum temperature range for cascade refrigeration system is 245°K to 260°K without superheating and subcooling and 250°K to 265°K, with superheating and subcooling.

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