

Thermodynamic Analysis of Cascade Refrigeration System using a Natural Refrigerants for Supermarket Application

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ABSTRACT: This paper deals with study of the thermodynamic analysis of eco-friendly/natural fluids used in cascade refrigeration systems. Supermarket is a large grocery store which stores the all frozen product and chilled product in different cold temperature condition. According to the Montreal protocol and Kyoto protocol underlined the need of substitution of CFC's and HCFC's due to their adverse impact on atmospheric ozone layer which protects earth from U.V rays. The CFCs have been entirely ruled out since 1995 and a long- term basis HCFCs must be replaced by 2020. HFC refrigerants which are harmless to ozone layer. Natural refrigerants are used in refrigerating system due to have low ODP and GWP in nature. A Thermodynamic analysis of a cascade refrigeration cycle with different refrigerant pair is presented in this paper. R744 is used in Low-temperature cycle whereas R134a, R290, R717 and R404a (R125 (44%)/R143a (52%)/R134a (4%)) are used in the High-temperature cycle. Effect of various operating parameters i.e. evaporator temperature, condenser temperature, temperature difference in cascade condenser and low temperature cycle condenser temperature on performance parameters viz. COP and refrigerant mass flow ratio have been studied. Thermodynamic analysis shows that out of all refrigerant pairs, the COP of R744-R717 refrigerant pair is highest as compared to other cascade refrigerant pairs and also most promising pairs for supermarket application.

KEYWORDS: Cascade refrigeration system; Co-efficient of performance; R717; R744; R290; R134a; R404a.

I. INTRODUCTION

A cascade refrigeration system consists of two independently operated single-stage refrigeration systems. A lower system that maintains a lower evaporating temperature and produces a refrigeration effect and a higher system that operates at a higher evaporating temperature. For some industrial applications that require moderately low temperatures with a considerably large temperature and pressure difference then the single stage vapor-compression refrigeration cycles become impractical. One of the solutions for such cases is to perform the refrigeration in two or more stages which operate in series. These refrigeration cycles are called cascade refrigeration cycles. Therefore, cascade systems are employed to obtain high-temperature differentials between the heat source and heat sink. Ammonia, carbon dioxide, propane and other natural refrigerants have drawn increased attention as working fluids to protect the environment. An appropriate selection of refrigerant to operate the LT and HT cycles should be made in order to obtain high coefficient of performance (COP). The temperature level in LT and HT cycle is also an important parameter to decide best working fluids along with other important characteristics such as toxicity, flammability, ODP, GWP etc.

The Montreal protocol and Kyoto underlined the need of substitution of CFC's and HCFC's regarding their bad impact on atmospheric ozone layer which protects earth from U.V rays [1]. A direct expansion in low temperature refrigeration cycle [2] involves a large pressure lift between evaporating and condensing temperatures resulting in an increase in the compression ratio and reduction of volumetric efficiency of the compressors. Now a day's GWP and ODP of the refrigerant is also considering due to environmental safety. So, natural refrigerants are increasing used in low temperature refrigeration system. Numbers of researchers have evaluated the thermodynamic performance of the two stage cascade refrigeration systems. A carbon dioxide-ammonia (R744- R717) cascade system thermodynamically to determine the optimum condensing temperature of CO₂ in LT cycle [3]. A carbon dioxide-propane (R744-R290)

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(An ISO 3297: 2007 Certified Organization)

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cascade system was analysed by [4], where an optimum cascade evaporating temperature of CO₂ in HT cycle was determined for heating application. The optimum cascade condensing temperatures of R744 for different refrigerants such as R717, R290, R1270 and R404a, which are in the HT cycle of a cascade system [5]. Thermodynamic analysis of a cascade system using a R507a-R23 are non-flammable and ozone friendly refrigeration pair [6]. The theoretical and experimental investigation of a two-stage vapor compression cascade refrigeration system using R-134a as the refrigerant [7]. The analysis of an endo-reversible two-stage cascade cycle and obtained optimum intermediate temperature for maximum exergy and refrigeration effect [8]. A comprehensive numerical model of a trans critical CO₂-C₃H₈ cascade system was developed with intent to verify the theoretical results. To carry out the experiment to analyse the performance of a CO₂-NH₃ cascade system. They compared their maximum COP, which corresponded to the optimal intermediate temperature, with the result from the correlations [9]. CO₂ in LT cycle and NH₃ in HT cycle have been successfully used in cascade system for cooling where -323K temperature is required [10]. For a low temperature application (R23-R507a) HFC based refrigerant pairs analysis carried out and result shows that the system COP reduced by about 17.10 % when temperature difference in cascade condenser has been increased[11]. The main objective of this paper is to study and find out the best working fluids pair for a cascade refrigeration cycle. In the present work, R744 was used in LT cycle and R134a, propane, ammonia, and R404a are used in HT cycle for the comparative analysis of cascade refrigeration cycle in terms of COP. The effect of various operating parameters is studied on performance parameters. Thermodynamic analysis is carried out by developing computational model in Engineering Equation solver (EES).

II. SYSTEM DESCRIPTION

A Line diagram, Pressure-enthalpy diagram and Temperature-entropy diagram of two stage cascade refrigeration system using two different refrigerants as shown in Fig.1, Fig.2 and Fig.3 respectively. Fig. 1 shows cascade refrigeration system with its components. It consists of two stages lower temperature (pressure) cycle and higher temperature (pressure) cycle which is connected by cascade condenser. The main components of cascade system are: Low temperature compressor, High temperature compressor, Condenser, Evaporator, Cascade Condenser and Throttling devices for Low temperature cycle and High temperature cycle.

During process 1-2 the low temperature cycle refrigerant is compressed isentropically. It is then passes through cascade condenser where it gives heat to refrigerant of higher temperature cycle (process 2-3). It expands in throttling device (process 3-4) and further passes to evaporator (process 4-1) to produce necessary refrigerating effect. In higher stage refrigerant is compressed in high temperature cycle compressor (process 5-6), then it passed through condenser where it rejects heat (process 6-7). It expands isentropically in throttling device (process 7-8) further passes to cascade condenser where heat transfer between two refrigerants takes place. The high temperature and low temperature cycle uses refrigerant with high critical temperature, high critical pressure and low critical temperature, low critical pressure respectively as shown in TABLE I.

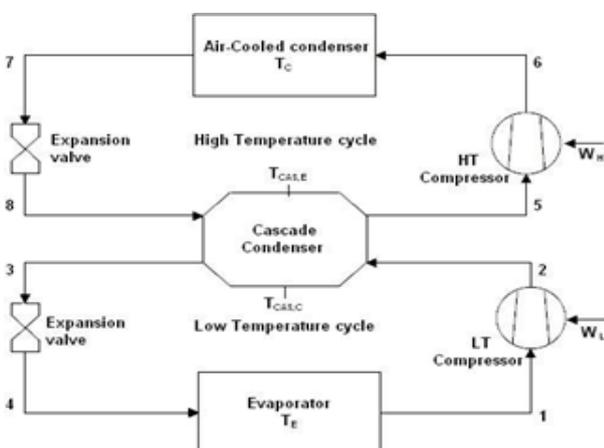


Fig.1 Line diagram of cascade system

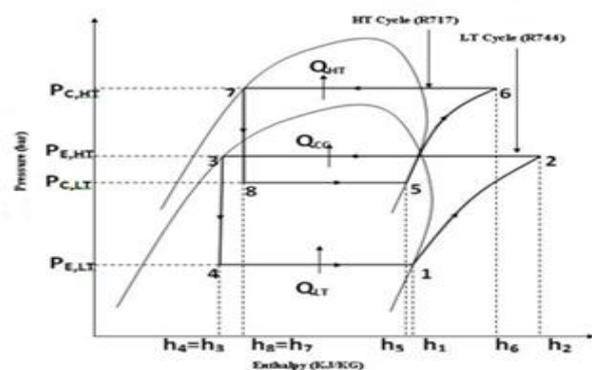


Fig.2 Pressure – Enthalpy diagram of cascade system

International Journal of Innovative Research in Science, Engineering and Technology

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TABLE 1
THERMOPHYSICAL AND ENVIRONMENTAL
PROPERTIES OF THE WORKING FLUIDS USED IN

	R744	R717	R290	R134a	R404a
Molecular Weight(kg/kmol)	44	17	44.097	102	97.6
Boiling point(°C)	-78	-33.34	-42.09	-26.3	-46.6
Critical temperature(°C)	31.40	132.4	96.7	101.06	72.14
Critical pressure(bar)	73.80	113.33	42.48	40.59	37.35
ODP	0	0	0	0	0
GWP	1	0	3.3	1430	3922

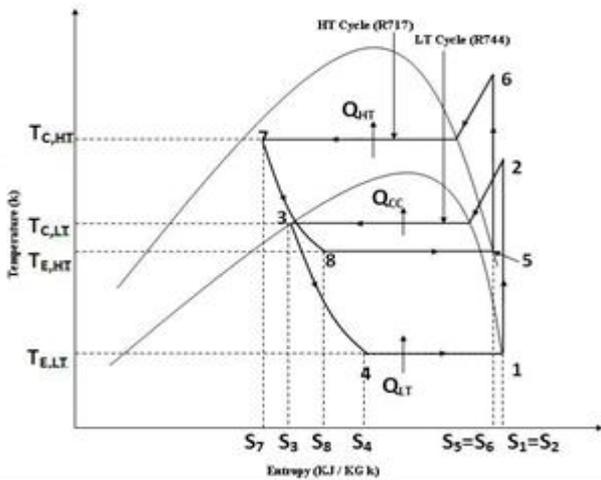


Fig. 3 Temperature-Entropy diagram of cascade system

III. THERMODYNAMIC ANALYSIS

Following assumptions are considered for the thermodynamic analysis of two stage cascade refrigeration system.

- Adiabatic and irreversible compression with an isentropic efficiency of 0.8 for both high- and low-temperature compressors.
- Negligible pressure and heat drop in the piping or system components.
- Isenthalpic expansion of refrigerants in expansion valves.
- Heat transfer process in heat exchanger is isobaric.
- Changes in kinetic and potential energy are negligible.

The thermo physical and environment properties of the refrigerants used in this paper are given in Table1. Thermodynamic analysis in this paper has been done using a software package called as Engineering Equation Solver [11]. Steady flow energy equation and mass balance equations for the analysis.

The capacity of the evaporator is defined as:

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

Compressor power consumption for low-temperature cycle is given as:

$$\dot{W}_L = \dot{m}_L (h_2 - h_1) \tag{2}$$

Whereas, for high-temperature cycles, it is given as:

$$\dot{W}_H = \dot{m}_H (h_6 - h_5) \tag{3}$$

Total work done or Actual work done:

$$\dot{W}_{total} = \dot{W}_H + \dot{W}_L \tag{4}$$

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

$$\dot{Q}_{CC} = \dot{m}_L (h_2 - h_3) = \dot{m}_H (h_5 - h_8) \tag{5}$$

From the above Eq.(5) the mass flow ratio is derived as:

$$\frac{\dot{m}_H}{\dot{m}_L} = \frac{h_2 - h_3}{h_5 - h_8} \tag{6}$$

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

$$\dot{Q}_H = \dot{m}_H (h_6 - h_7) \tag{7}$$

The overall COP of the system is determined as:

$$COP = \frac{\dot{Q}_E}{\dot{W}_{total}} \tag{8}$$

From the equations (1) to (7), COP can also defined as

$$COP = \frac{(h_5 - h_8) * (h_1 - h_4)}{((h_6 - h_5) * (h_2 - h_3) + (h_5 - h_8) * (h_2 - h_1))} \tag{9}$$

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With the help of these equations (1) to (9), the thermodynamic state points and properties like pressure, temperature, specific enthalpies of the different refrigerant pair of cascade refrigeration system should be calculated. Based on the above-mentioned thermodynamic model; COP for the cascade system has been evaluated for different pair of refrigerants. A parametric analysis has been performed by varying mass flow ratio, sub cooling and superheating, evaporating and condensing temperatures. The results and discussions are presented in next section.

IV. RESULTS AND DISCUSSION

A Computerized program has been developed to find the effect of particular parameter on the performance of a system by considering other parameters of a system as constant. The parameters assumed for the computation of results are given below.

- i. Low temperature cycle evaporating temperature, $T_E = -50^\circ\text{C}$
- ii. Low temperature cycle condensing temperature, $T_{CAS_C} = 0^\circ\text{C}$
- iii. High temperature cycle condensing temperature, $T_C = 40^\circ\text{C}$
- iv. Temperature difference in cascade condenser, $(\Delta T)_{CC} = 2.5^\circ\text{C}$
- v. Degree of superheating, $(\Delta T)_{sup} = 0^\circ\text{C}$ in both HT and LT cycle.
- vi. Degree of sub cooling, $(\Delta T)_{sub} = 0^\circ\text{C}$ in both HT and LT cycle.
- vii. Effectiveness of cascade heat exchanger $\epsilon = 1$

In cascade refrigeration system for system results mass flow rate of lower temperature cycle is assumed 0.2 kg/min. The parameters have been varied for the computation results are mentioned below:

- i. The low temperature cycle evaporator temperature is varied from $T_E = -55^\circ\text{C}$ to -30°C
- ii. The high temperature cycle condenser temperature is varied from $T_C = 30^\circ\text{C}$ to 45°C
- iii. The cascade condenser temperature difference $\Delta T_{cc} = 1^\circ\text{C}$ to 13°C
- iv. The low temperature cycle condenser temperature is varied from $T_{CAS_C} = -30^\circ\text{C}$ to 10°C
- v. Degree of sub cooling $(\Delta T)_{sub}$ and Degree of superheating $(\Delta T)_{sup}$ is varied from 0°C to 10°C

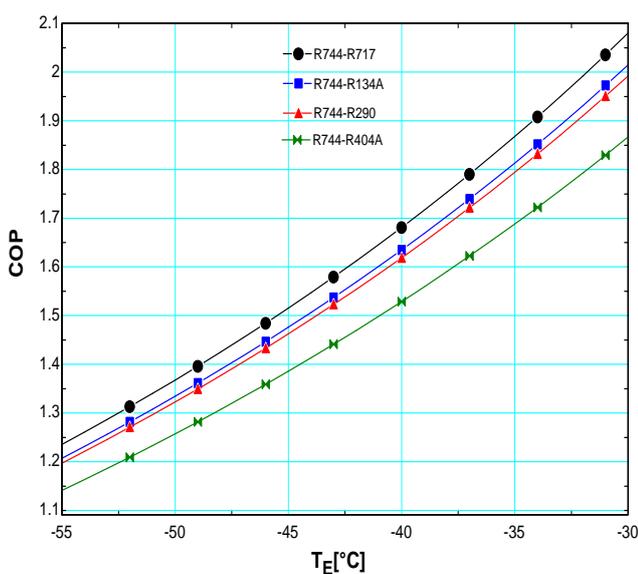


Figure 4 Effect of lower stage evaporating temperature on COP of different Refrigerating pairs

A. Effect of low stage evaporating temperature and higher stage condensing temperature

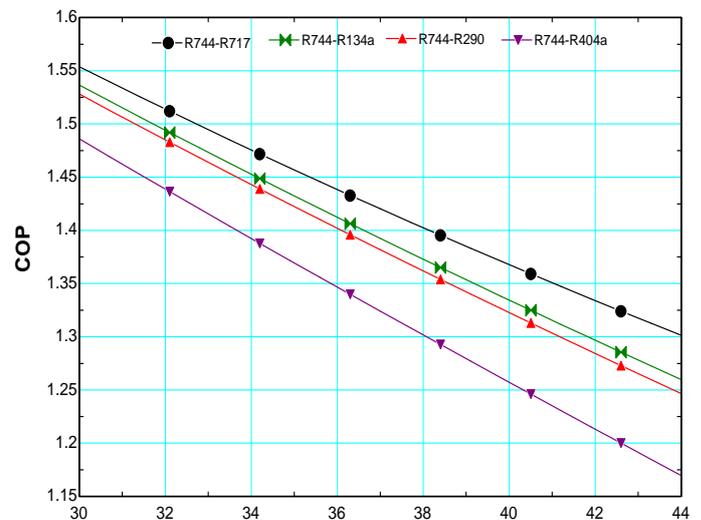


Figure 5 Effect of higher stage condensing temperature on COP of different Refrigerating

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From fig.4 shows that the cascade refrigerant pair R717-R744 is higher COP as compared to R134a-R744, R290-R744, and R744-R404a for same thermodynamic parameters with a low evaporating temperature range (T_{ELT}) of -55°C to -30°C . System performance is increased about 1.236 to 2.08, 1.207 to 2.015, 1.197 to 1.992 and 1.14 to 1.867 for R744-R717, R744-R134a, R744-R290, R744-R404a cascade pairs respectively. From fig 5 shows that when increasing the higher stage temperature of cascade system ($T_{C,HT}$) from 30°C to 55°C the cop of cascade refrigeration system with a refrigerant pair R744-R717, R744-R134a, R744-R290, and R744-R404a decreases from 1.554 to 1.301, 1.537 to 1.26, 1.528 to 1.247 and 1.488 to 1.169 for cascade pairs respectively.

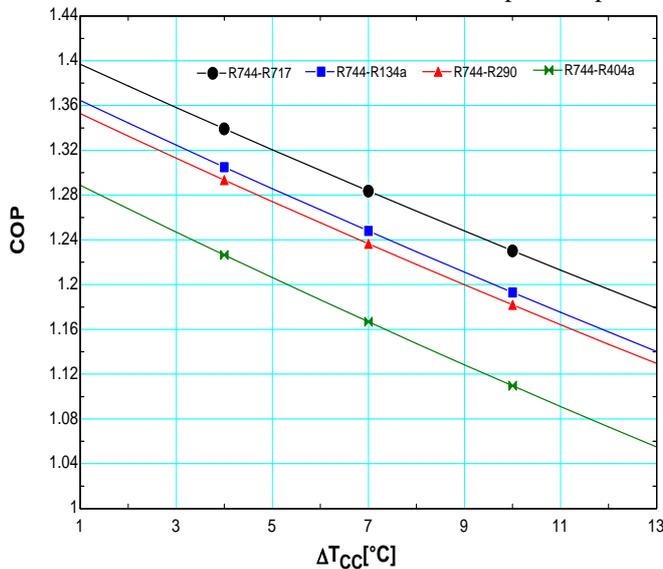


Figure 6 Effect of cascade condenser temperature difference on COP of different refrigerant pairs

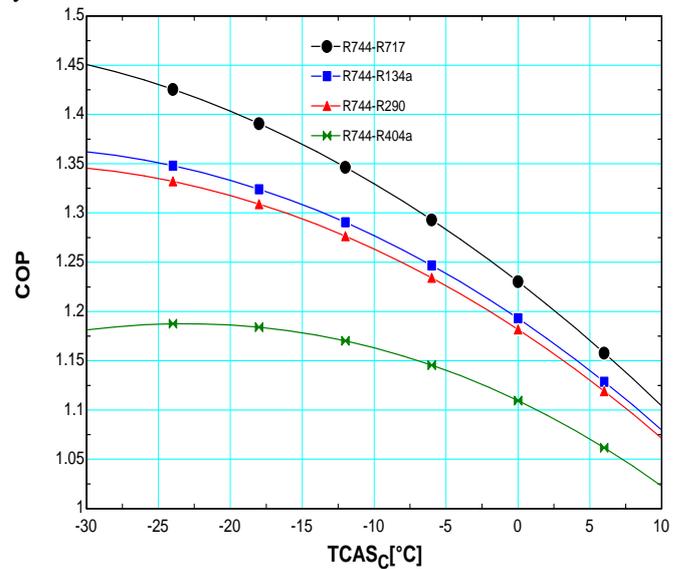


Figure 7 Effect of lower temperature cycle condensing temperature on COP of different refrigerant pairs

Fig.6 shows that when cascade condenser temperature (ΔT_{CC}) is increased from 1°C to 13°C ; COP of the system is decreased about 1.397 to 1.179, 1.364 to 1.14, 1.353 to 1.129, and 1.289 to 1.055 for R744-R717, R744-R134a, R744-R290 and R744-R404a cascade pairs respectively. Fig.7 illustrates the effect of lower temperature cycle condensing temperature on COP for different refrigerant pair. When the condensing temperature ($T_{C,LT}$) increases from -30°C to 10°C ; first COP of all cascade pair increases for a certain limits and reaches the maximum value at an optimum temperature ($T_{C,LT}$) then it start decreases. However, among the refrigerant pairs R744-R717 gives higher COP for given range of ($T_{C,LT}$) as compared to other cascade pairs and R404a-R744 is less COP. System cop is decreased from 1.451 to 1.104, 1.202 to 1.01, 1.346 to 1.071, and 1.184 to 1.005 for R744-R717, R744-R134a, R744-R290 and R744-R404a respectively.

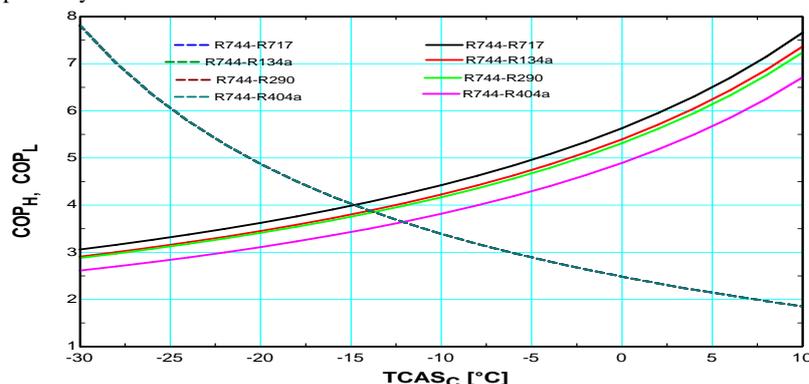


Figure 8 Effect of lower stage condensing temperature on COP, COP_{HT} and COP_{LT} for different refrigerant

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Fig.8 shows the effect of lower temperature stage condensing temperature ($T_{C,LT}$) on COP, COP_{HT} and COP_{LT} . The COP of lower temperature cycle is decreased with the increments of $T_{C,LT}$ which is increased from $-30^{\circ}C$ to $10^{\circ}C$; while COP of higher temperature is increased with $T_{C,LT}$. Overall COP of the system is initially increased to optimum temperature after further increment in $T_{C,LT}$ the COP is decreased.

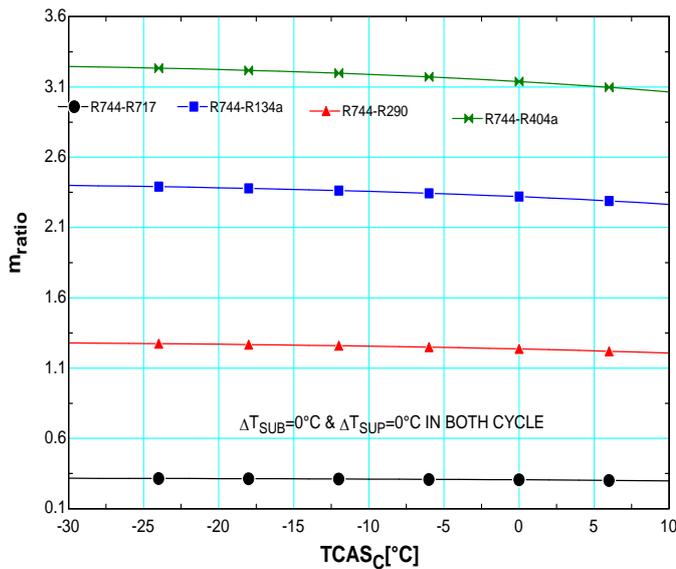


Figure 9 Effect of lower stage cycle condensing temperature on mass flow ratio of different refrigerant pairs considering at $0^{\circ}C$ superheating and $0^{\circ}C$ sub cooling in both cycle

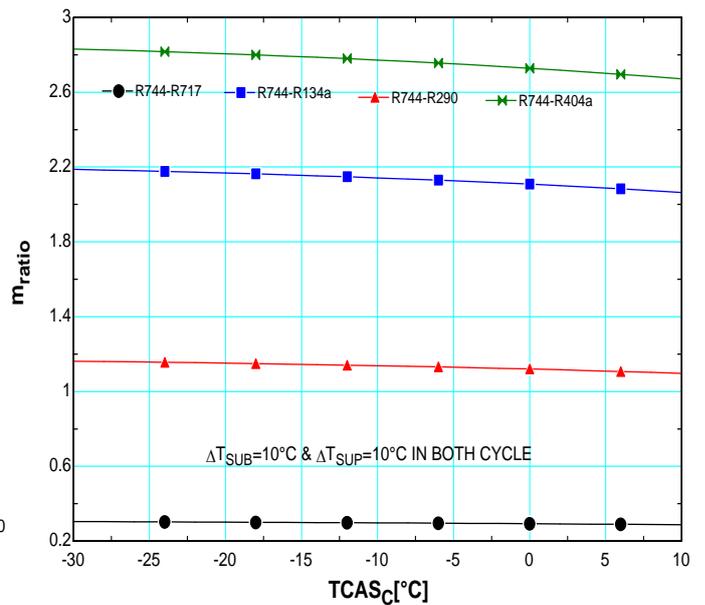


Figure 10 Effect of lower stage cycle condensing temperature on mass flow ratio of different refrigerant pairs considering at $10^{\circ}C$ superheating and $10^{\circ}C$ sub cooling in both cycle

Fig.9 and Fig.10 shows that when condensing temperature of LT is increased from $-30^{\circ}C$ to $10^{\circ}C$; the mass flow ratio is decreases. At higher $T_{C,LT}$ the difference in values of mass flow rate reduces for different degree of sub cooling and superheating. Ratio of mass flow is reduced when superheating and sub cooling is increased from $0^{\circ}C$ to $10^{\circ}C$ for both LT and HT cycle.

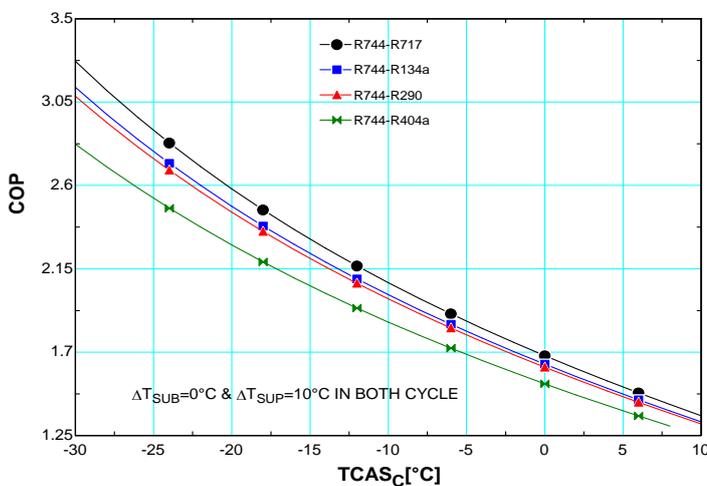


Figure 11 Effect of lower stage cycle condensing temperature on COP of different refrigerant pairs considering at $0^{\circ}C$ sub cooling and $10^{\circ}C$ superheating in both cycle

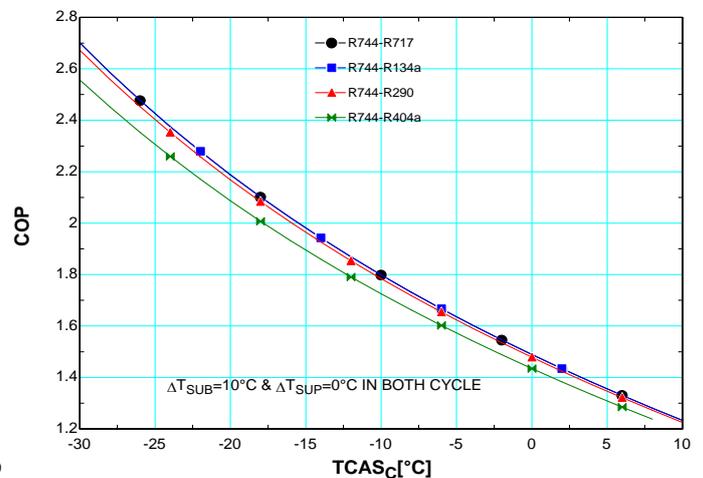


Figure 12 Effect of lower stage cycle condensing temperature on COP of different refrigerant pairs considering at $0^{\circ}C$ superheating and $10^{\circ}C$ sub cooling in both cycle

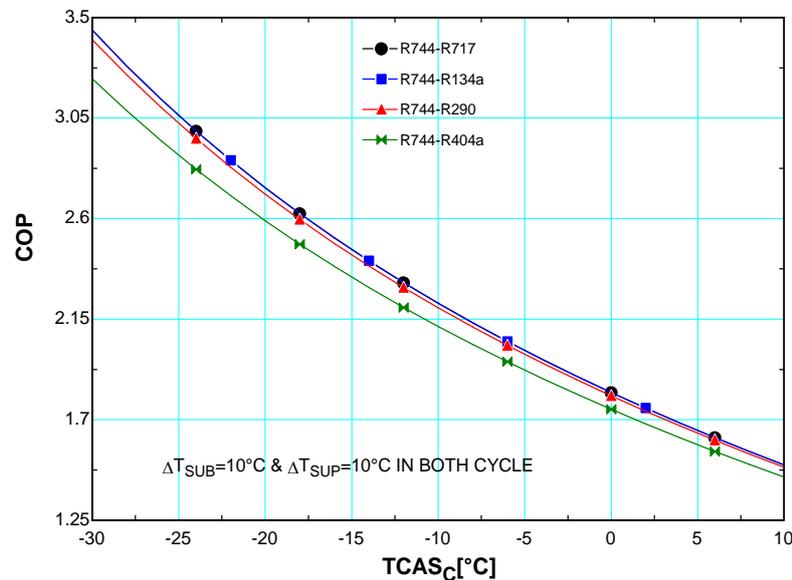


Figure 13 Effect of lower stage cycle condensing temperature on COP of different refrigerant pairs considering at 10° C superheating and 10° C sub cooling in both cycle

From fig 11 shows that when higher stage condensing temperature increased form -30°C to 10°C with considering the $\Delta T_{sub}=0^{\circ}C$ and $\Delta T_{sup}=10^{\circ}C$ in both cycle the cop is decreased from 3.27 to 1.357, 3.129 to 1.324, 3.081 to 1.313 and 2.823 to 1.248 for R744-R717, R744-R134a, R744-R290, R744-R404a cascade pairs respectively. From fig 12 shows that when higher stage condensing temperature increased form -30°C to 10°C with considering the $\Delta T_{sub}=10^{\circ}C$ and $\Delta T_{sup}=0^{\circ}C$ in both cycle the cop is decreased from 2.701 to 1.233, 2.702 to 1.233, 2.673 to 1.225 and 2.557 to 1.192 for R744-R717, R744-R134a, R744-R290 and R744-R404a cascade pairs respectively. From fig 13 shows that when higher stage condensing temperature increased form -30°C to 10°C with considering the $\Delta T_{sub}=10^{\circ}C$ and $\Delta T_{sup}=10^{\circ}C$ in both cycle the cop is decreased from 3.443 to 1.497, 3.444 to 1.497, 3.4 to 1.486 and 3.225 to 1.442 for R744-R717, R744-R134a, R744-R290 and R744-R404a cascade pairs respectively.

V. CONCLUSION

From the comparative assessment of different cascade refrigeration pairs; the R744-R717 pair has the maximum COP. Thermodynamics analysis of cascade refrigeration pair R744-R717 presents the following interpretation. The cascade refrigerant pair R744-R717 has higher COP as compared to other refrigerants pairs R744-R134a, R744-R290 and R744-R404A for the same T_C , T_E and $TCAS_C$; other parameters remain constant.

- R744-R717 pair has the higher COP as compared to other refrigerant pairs for supermarket application.
- The COP of the R744-R717 system increased from 1.236 to 2.08 at lower temperature cycle evaporator temperature (T_E) is varied from -55°C to -30°C; other parameters remain constant.
- The COP of the system R744-R717 decreased from 1.554 to 1.301 at higher temperature cycle condenser temperature (T_C) is varied from 30°C to 45°C; other parameters remain constant.
- For the same T_C , T_E , $TCAS_C$ and cascade condenser temperature difference; other parameters remain constant the cascade refrigerant pair R744-R717 has higher COP as compared to other refrigerants pairs R290-R744, R134a-R744, R23-R507a, R23-R410a and R23-R404a.pairs.
- When T_{CASC} is varied from -30°C to 10°C, the COP of higher temperature cycle increased whereas the COP of LT cycle decreased therefore the optimal value of $TCAS_C$ could found by the point where higher temperature and lower

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temperature cycles COP overlap to each other. This point's projection on horizontal axis gives optimum value of $TCAS_C$ and projection on vertical axis gives the maximum value of COP at optimum $TCAS_C$.

- Maximum COP of the system R744-R717 increased significantly with an increase in degree of superheat while increased slightly with increase in sub cooling.
- The mass flow ratio (\dot{m}_H/\dot{m}_L) decreased with rise in for different degree of superheating and sub cooling for R744-R717 system. Ratio of mass flow is reduced more in case of sub cooling as compared to superheating.

NOMENCLATURE

Abbreviation

COP	co-efficient of performance
Q	heat transfer
RE	refrigerating effect

Greek symbols

η	efficiency
ϵ	effectiveness

Subscripts

C	condenser
CC	cascade condenser
CASc	lower stage of cascade condenser
E	evaporator
HT	high temperature
LT	low temperature
sub	sub cooling
sup	superheating

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