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Performance Study of Cascade Refrigeration System Using Alternative Refrigerants

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Abstract**—**Cascade refrigeration systems employ series of single stage vapor compression units which are thermally coupled with evaporator/condenser cascades. Different refrigerants are used in each of the circuit depending on the optimum characteristics shown by the refrigerant for a particular application. In the present research study, a steady state thermodynamic model is developed which simulates the working of an actual cascade system. The model provides COP and all other system parameters e.g. total compressor work, temperature, pressure, enthalpy and entropy at different state points. The working fluid in low temperature circuit (LTC) is CO_2 (R744) while Ammonia (R717), Propane (R290), Propylene (R1270), R404A and R12 are the refrigerants in high temperature circuit (HTC). The performance curves of Ammonia, Propane, Propylene, and R404A are compared with R12 to find its nearest substitute. Results show that Ammonia is the best substitute of R12.

Keywords**—**Cascade system, Refrigerants, Thermodynamic model.

I. Introduction

AFTER getting insight into the harmful effects of CFC based refrigerants on ozone layer depletion and global warming, search to find alternatives to these working fluids gained large interest during last few years. Moreover international conventions are also requesting to reduce their usage. Therefore, the researchers are working hard to find the alternatives, which can replace CFCs.

Low temperature applications require refrigeration in the range of -30°C to -100°C. Single stage vapor compression refrigeration system is unable to achieve such low temperatures with the use of reciprocating compressor, due to very high pressure ratios across the compressor. Higher-pressure ratios lead to higher condenser temperatures. Also the volumetric efficiency and hence the capacity of the reciprocating compressor significantly reduces. Although multistage or screw compressors can help, but the use of single refrigerant at low temperature is restricted by solidification temperature of the refrigerant, extremely low pressures in the evaporator, large suction volumes in the evaporator for a high boiling point refrigerant and high condenser pressure for a low boiling refrigerant. This necessitates integrating for other viable options to partially or fully overcome the above shortcomings.

The characteristics of any refrigerant to exhibit best performance, when operating in a certain range of temperature and pressure, provide cascade refrigeration systems an edge over single stage and multistage refrigeration systems for low temperature applications. Cascade refrigeration systems employ series of single stage units which are thermally coupled with evaporator/condenser cascades. Different refrigerants are used in each of the circuit depending on the optimum characteristics shown by the refrigerant for a particular application.

Two stage cascade refrigeration systems employ two circuits namely a high temperature circuit (HTC) and a low temperature circuit (LTC). The high temperature circuit serves to extract heat from the low temperature circuit and the desired cooling is achieved at the evaporator of the low temperature circuit. The two circuits are coupled together by a heat exchanger called the cascade condenser, where the refrigerant vapors of LTC are condensed by rejecting heat to the refrigerant in the HTC. The intermediate temperature between the two circuits is an important design parameter that decides the COP of the entire system.

Zubair [1] simulated the performance of an actual single stage vapor compression system using a thermodynamic model. The model was used to study the performance of a variable-speed refrigeration system in which the evaporator capacity was varied by changing the mass-flow rate of the refrigerant for the constant inlet chilled-water temperature. Cabell et al. [2] proposed a simplified steady-state model to predict the energy performance of a single stage vapor compression plant. Kilicarslan [3] did the theoretical and experimental investigation of a two-stage vapor compression cascade refrigeration system using R-134a as the refrigerant. Bhattacharya et al. [4] did the analysis of an endo-reversible two-stage cascade cycle and obtained optimum intermediate temperature for maximum exergy and refrigeration effect. A comprehensive numerical model of a transcritical CO2-C3H8 cascade system was developed with intent to verify the theoretical results. Getu and Bansal [5] did the thermodynamic analysis of carbon dioxide–ammonia (R744–R717) cascade refrigeration system to optimize the design and operating parameters of the system. A multilinear regression analysis was employed in terms of operating parameters in order to develop mathematical expressions for maximum COP, optimum evaporating temperature and optimum mass flow ratio of R717 to that of R744 in the cascade system. Lee et al.

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[6] studied thermodynamically a cascade refrigeration system that uses carbon dioxide and ammonia as refrigerants, to determine the optimal condensing temperature of the cascade-condenser to maximize the COP and minimize the exergy destruction of the system. Bhattacharyya et al. [7] studied the performance of a cascade refrigeration–heat pump system based on a model incorporating both internal and external irreversibilities. It further explored the optimum allocation of heat exchanger inventories in cascade refrigeration cycles for the maximization of performance and minimization of system cost. Calm [8] reviewed the progression of refrigerants, from early uses to the present, and then addresses future directions and candidates. The paper examines the outlook for current options in the contexts of existing international agreements, including the Montreal and Kyoto Protocols to avert stratospheric ozone depletion and global climate change, respectively. Bansal and Jain [9] showed that carbon dioxide is a potential low temperature refrigerant for temperature down to -50°C due to its low cost, easy availability, and favorable properties. Maj [10] did the design and development of two stage cascade refrigeration system using CO_2 as LTC refrigerant and Propane as HTC refrigerant.

Literature shows that several refrigerants have emerged to substitute R12, the most widely used fluorocarbon refrigerant in the world. These include the environmental friendly refrigerants i.e. R717, R290, R744, R404A and R1270. In the present work thermodynamic analysis is carried out to evaluate the performance of cascade systems at varying design parameters using a few selected refrigerants. For this study, CO2 is selected as the low-temperature refrigerant due to its many inherent advantages. Propane, propylene, R404A, and ammonia are selected as the high-temperature refrigerants. Carbon dioxide is not considered as a high-temperature refrigerant as it results in very high pressure and transcritical operation of the system. Table I shows the physical and environmental characteristics of these refrigerants.

TABLE I
PHYSICAL AND ENVIRONMENTAL CHARACTERISTICS OF SELECTED REFRIGERANTS

Properties	R717	R290	R12	R744	R404A	R 1270
Chemical Formula	NH_3	C_3H_8	CCl_2F_2	CO_2	CHF_2CF_3/ CH_3CF3/ CH_2FCF_3	C_3H_6
Molecular Weight (kg / Kmol)	17.03	44.09	120.91	44.01	97.6	42.08
B.P. at 1.013 bar [°C]	33.32	-42.09	29.75	-56.55	-46.6	-47.69
Critical temperature [°C]	135.2	96.67	111.97	30.97	72.1	92.42
Critical pressure [bar]	113.3	42.47	41.36	73.77	37.4	46.6
O.D.P	0	0	0.9	0	0	0
G.W.P	0	3	8500	1	3260	3

II. THERMODYNAMIC MODEL OF CASCADE REFRIGERATION SYSTEM

A schematic diagram of a two stage cascade refrigeration system is illustrated in Fig. 1. The P-h diagram (Fig. 2) depicts all the state points corresponding to Fig. 1 including saturated lines. The thermodynamic analysis of the two-stage cascade refrigeration system was performed based on the following general assumptions:

(i) Adiabatic compression with given isentropic efficiency for both high- and low-temperature compressors,
(ii) Negligible pressure loss in pipe networks or system components,
(iii) Isenthalpic expansion of refrigerants in expansion valves,
(iv) Negligible changes in kinetic and potential energy.

The thermophysical properties of the refrigerants studied in this work are determined using REFPROP. The following sequence of equations was applied for the analysis.

The heat-transfer rate to the cycle in the evaporator is

$$Q_{evap} = (\varepsilon C)_{evap}\left[T_{in,evap} - T_{evap}\right] \quad (1)$$

Temperature of the cascade condenser for low temperature circuit is given by

$$T_6 = T_o + dT_{overlap} \quad (2)$$

Isentropic efficiency of LTC compressor is given by

$$\eta_{isen} = (h_{5s} - h_4)/(h_5 - h_4) \quad (3)$$

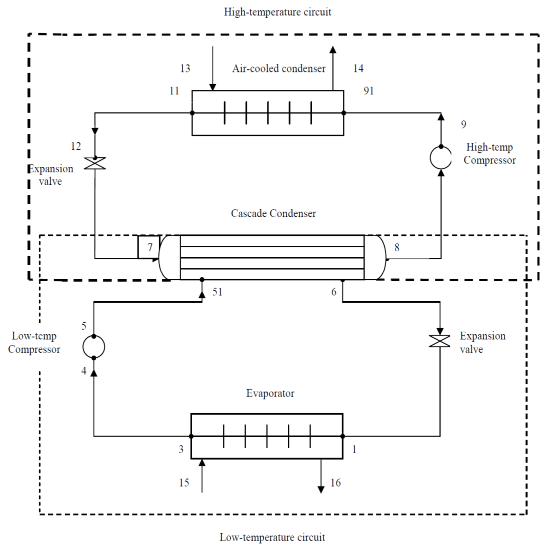


Fig. 1 Pressure-enthalpy diagram for cascade refrigeration system

Since process 1-6 is isenthalpic process,

$$h_1 = h_6 \quad (4)$$

Mass flow rate of refrigerant in LTC can be calculated from following equation,

$$Q_{evap} = m_{LTC}(h_3 - h_1) \tag{5}$$

Assuming n % of heat loss in the discharge line of compressor, hence heat loss is

$$Q_{loss} = n\, Q_{evap} \tag{6}$$

Heat leakage into the discharge line of LTC is also given by the formulation:

$$Q_{loss} = m_{LTC}(h_5 - h_{51}) \tag{7}$$

Isentropic efficiency of HTC compressor is given by

$$\eta_{isen} = (h_{9s} - h_8)/(h_9 - h_8) \tag{8}$$

Process 12-7 is isenthalpic. So

$$h_7 = h_{12} \tag{9}$$

Mass flow rate of refrigerant in HTC is given by

$$m_{HTC} = m_{LTC}[(h_{51} - h_6)/(h_8 - h_7)] \tag{10}$$

Heat leakage into the discharge line of HTC is,

$$Q_{loss} = m_{HTC}(h_9 - h_{91}) \tag{11}$$

Heat transferred in condenser is

$$Q_{cond} = (\varepsilon C)_{evap}[T_{in,evap} - T_{evap}] = m_{HTC}(h_{91} - h_{11}) \tag{12}$$

Total power input to the compressor is given by

$$W = [m_{LTC}(h_5 - h_4)] + m_{HTC}(h_9 - h_8) \tag{13}$$

Hence,

$$COP = Q_{evap}/W \tag{14}$$

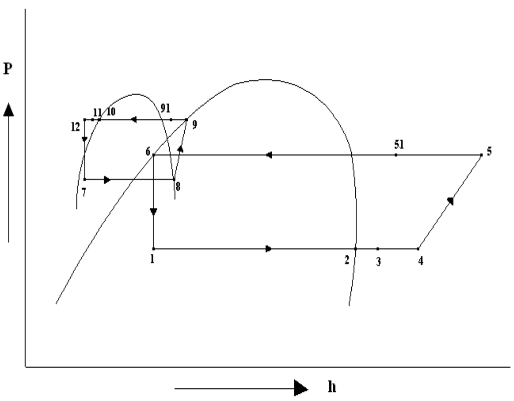


Fig. 2 Schematic diagram of a cascade system

III. Results and Discussion

A. Input Conditions

It should be noted that for an actual vapor compression system the efficiency of the reciprocating compressor and effectiveness of the heat exchangers do not remain constant with the variation in refrigeration capacity. However for the present investigation, we have considered these parameters to be constant. The values of inputs at design condition are given in Table II.

TABLE II
VALUES OF INPUTS AT DESIGN POINT

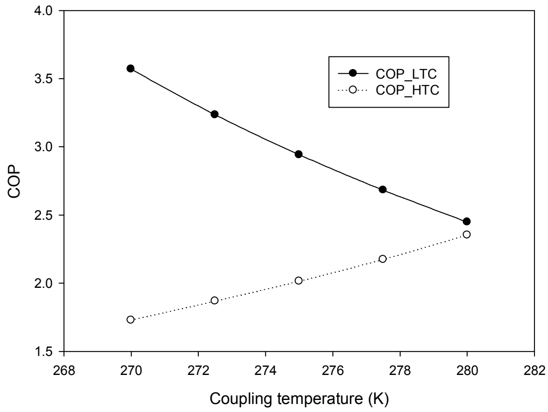
Parameters	Values
Evaporator coolant inlet temperature ($T_{in, evap}$)	250 K
Condenser coolant inlet temperature ($T_{in, cond}$)	313 K
Rate of heat absorbed by evaporator (Q_{evap})	66.67 kW
Product of condenser effectiveness and capacitance rate of external fluid [$(\varepsilon C)_{cond}$]	9.39 kW/K
Product of evaporator effectiveness and capacitance rate of external fluid [$(\varepsilon C)_{evap}$]	8.2 kW/K
Efficiency of compressors (η_{isen})	0.65
Degree of overlap or approach ($T_{overlap}$)	5 K

B. Optimum Coupling Temperature

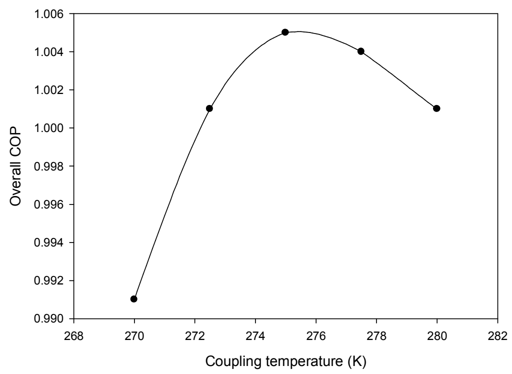
The temperature of the evaporator of HTC is called the coupling temperature. The optimum condensing temperature and the degree of approach of a cascade condenser is an important issue while designing a cascade refrigeration system to run under specific conditions such as fixed condensing and evaporating temperature and the temperature difference between the high- and low-circuits in cascade-condenser.

For reversible cycles, assuming no temperature difference between the two fluids in cascade condenser, the optimum cascade temperature is the geometric mean of the condenser and evaporator temperatures of the cascade system, i.e., to have the same temperature ratio in each circuit, the assumption of a single temperature for both the fluids is not practically feasible, as it would demand infinite area of the cascade condenser. The optimum temperature difference between the two fluids would depend not only on the heat transfer characteristics of the refrigerants in the two circuits but also on the economics of the design (operating versus capital cost). The larger the temperature difference, the lower the COP of the system.

Fig. 3 (a) shows the variation of COP of LTC and HTC and Fig. 3 (b) shows the overall COP of system with coupling temperature for CO2 and R404A system at the design point conditions.



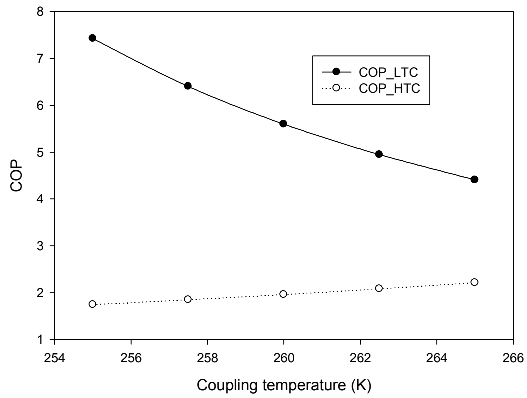
(a)



(b)

Fig. 3 COP v/s coupling temperature for R404A (a) LTC and HTC (b) overall system

Fig. 3 (a) depicts that COP of HTC increases while the COP of LTC decreases with coupling temperature. Hence an optimal coupling temperature and its corresponding maximum COP exist. Fig. 3 (b) reveals that the overall COP of system is maximum at 275 K. Hence it is taken as optimal coupling temperature for CO2 and R404A system.



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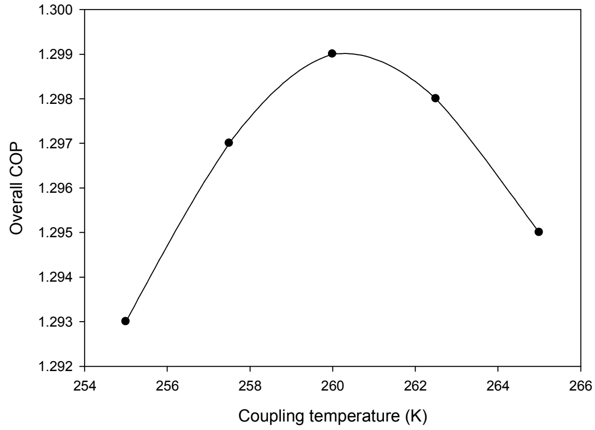
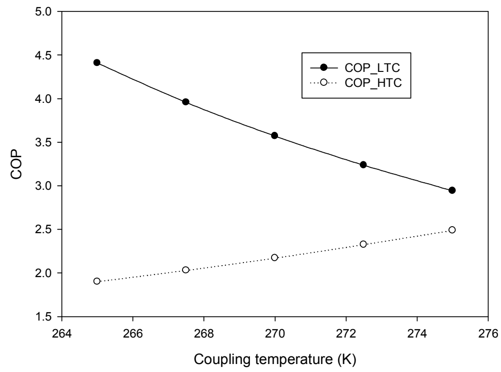


(b)

Fig. 4 COP v/s coupling temperature for Ammonia (a) LTC and HTC (b) overall system



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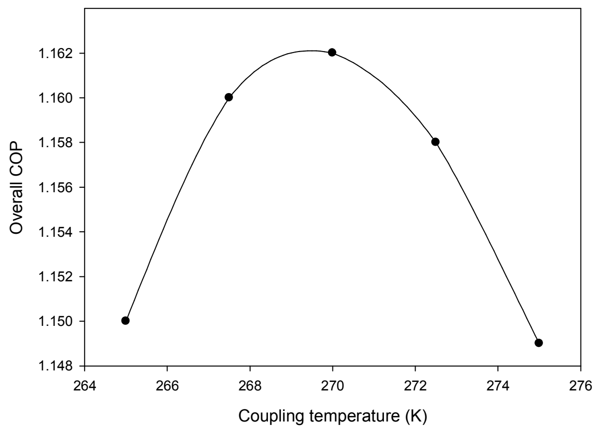
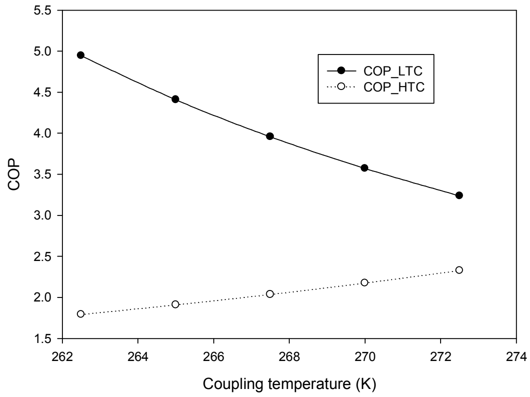


(b)

Fig. 5 COP v/s coupling temperature for Propane (a) LTC and HTC (b) overall system



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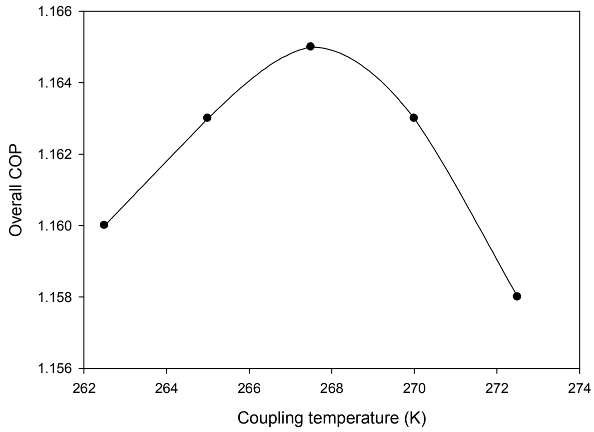
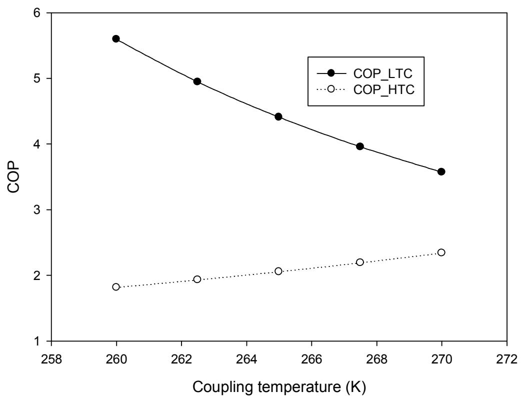
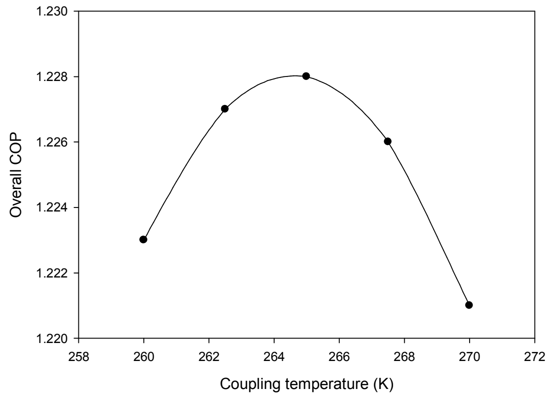


(b)

Fig. 6 COP v/s coupling temperature for Propylene (a) LTC and HTC (b) overall system



(a)



(b)

Fig. 7 COP v/s coupling temperature for R12 (a) LTC and HTC (b) overall system

Figs. 4 to 7 represent the variation of COP of LTC, HTC and overall system with coupling temperature for Ammonia, Propane, Propylene and R12 respectively. The optimum coupling temperature for R404A, Ammonia, Propane, Propylene and R12 system is 275 K, 260 K, 270 K, 267.5 K, and 265 K respectively.

C. Effect of Degree of Approach

The temperature difference between the two fluids of the cascade condenser is called degree of overlap or approach. Fig. 8 shows the variation of overall COP with degree of overlap. It can be seen that the larger the temperature difference, the lower the COP of the system. At the design point COP for Ammonia system is maximum and minimum for R404A.

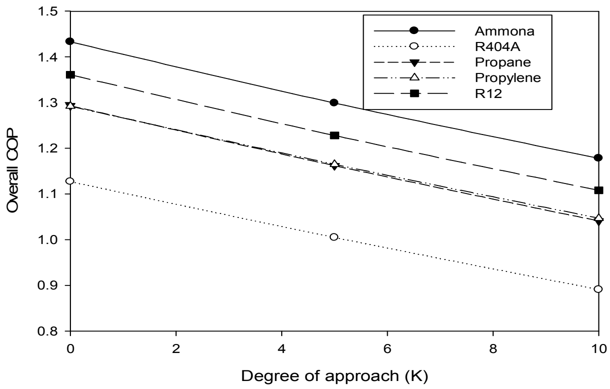
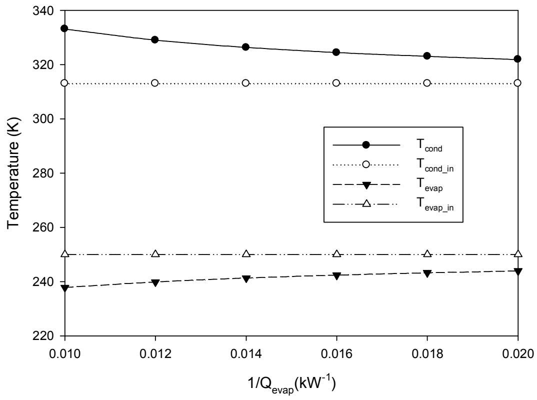


Fig. 8 Variation of overall system COP with degree of approach

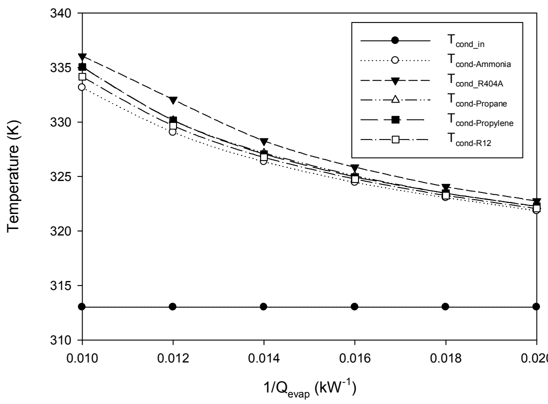
D. Effect of Refrigeration Capacity

Fig. 9 (a) shows the effect of variation of inverse of cooling capacity on system temperatures for CO2-Ammonia fluid pair. Evaporator temperature increases and condenser temperature decreases with the decrease in cooling capacity while keeping the inlet temperature of external fluid i.e. $T_{in,cond}$, $T_{in,evap}$ as constant. There is a slight difference in the condenser temperature (T_{cond}) for different refrigerants as revealed in Fig. 9 (b).

The amount of heat transfer in condenser for ammonia, R404A, Propane, Propylene and R12 are 92.49 k W, 100 kW, 95.30 kW, 95.3 kW and 94.36 kW respectively.



(a)



(b)

Fig. 9 (a) Variation of temperature with inverse of cooling capacity for Ammonia system and (b) Variation of condenser temperature with inverse of cooling capacity

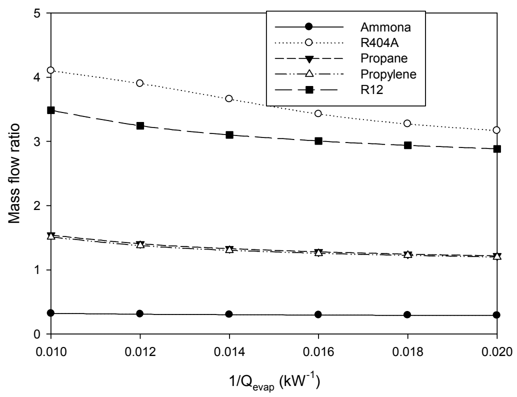


Fig. 10 Variation of mass flow ratio with inverse of cooling capacity

It can be seen from Fig. 10 that at high evaporator capacity, mass flow ratio is high and, thus, the temperature difference in the heat exchangers is also high.

Therefore, the losses due to finite-temperature difference in the heat exchangers are high; hence, the COP is reduced as shown in Fig. 11. But as the capacity is decreased, the temperature difference in heat exchangers also decreases, therefore the losses due to the finite rate of heat transfer also decreases and the COP of the system increases. At the designed point mass flow ratio is maximum for R404A and is minimum for Ammonia. At the designed point COP is maximum for Ammonia and is minimum for R404A system. The variation of mass flow ratio and COP for Propane and Propylene systems is almost same.

IV. Summary of Relative Comparison of Refrigerants with R12 System

On the basis of results obtained; Ammonia, Propane, Propylene and R404A are compared with R12 at the design point. The comparison in tabular form is given in Table III.

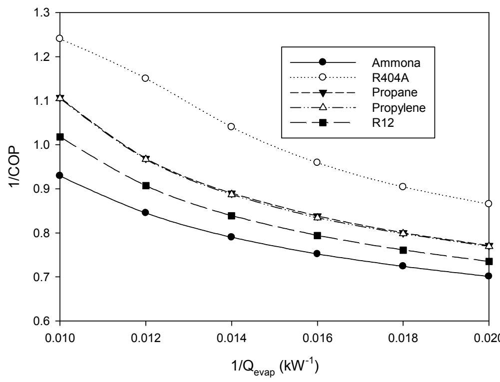


Fig. 11 Variation of inverse of COP with inverse of cooling capacity

TABLE III
Summary of Relative Comparison of Various Refrigerants with R12

FACTORS	Ammonia	Propane	Propylene	R404A
HTC Pressure ratio	152.52 %	76.53 %	80.09 %	70.38 %
HTC Refrigerant charge	9.37 %	44.83 %	43.02 %	128.84 %
COP	105.78 %	94.62 %	94.86 %	81.84 %
Compressor Work	94.53 %	105.71 %	105.43 %	122.23 %
Coupling Temperature	98.11 %	101.88 %	100.94 %	103.77 %
Condenser Heat Transfer	97.62 %	102.73 %	102.37 %	110.27 %

The coupling temperature and condenser heat transfer is maximum for R404A and is minimum for Ammonia. The system cost will be more for R404A and will be least for Ammonia. Propane is high capacity and high pressure refrigerant compared to R12. Because of these characteristics; Propane air conditioner of same capacity requires a larger displacement compressor, large evaporator, condenser and tubing. Hence, Propane system costs more to build and operate than an equivalent R404A system. Propylene has similar characteristic as Propane, hence propane system can easily be replaced with propylene. R404A is higher pressure refrigerant as compared to R12. It has lower COP and high refrigerant charge as compare to R12 system. So it requires large displacement compressor, evaporator, condenser and tubing. Hence it has high cost.

Ammonia is high pressure and high capacity refrigerant than R12.Ammonia is the best high-temperature refrigerant among propane, propylene, R12 and R-404A considered in this study. It gives a theoretical optimum COP of about 1.299 with the lowest mass flow rate of 0.078 kg/s in HTC at a condenser temperature of 52.2°C, an evaporator temperature of –31.28°C, and a cascade heat exchanger approach of 5 K. It also has the lowest optimum operating temperatures and pressures for maximum COP in the cascade heat exchanger.

NOMENCLATURE

COP	Coefficient of performance
C	Capacitance rate for the external fluids (kW/K)
h	Specific enthalpy of refrigerant (kJ/kg)
m	Mass flow rate (kg/s)
Q_{cond}	Rate of heat rejection in condenser (kW)
$Q_{loss, cond}$	Rate of heat leak from the hot refrigerant (kW)
Q_{evap}	Rate of heat absorbed by the evaporator (kW)
$Q_{loss, evap}$	Rate of heat leak from the ambient to the cold refrigerant (kW)
T	Temperature (K)
To	Coupling temperature (K)
W	Electrical power input to compressor (kW)

Greek symbols

ε	Effectiveness of heat exchanger
η	Efficiency

Subscripts

1,2,3,……	state points
comp	compressor
cond	condenser
evap	evaporator
HTC	High temperature circuit
isen	isentropic
in	Inlet
LTC	low temperature circuit
ref	refrigerant

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