Comparative Assessment of Alternative Refrigerants in Cascade Refrigeration System

[1]Prasanna Songire, [2]Kundalik V. Mali

[1,2]Department of Mechanical Engineering, Sinhgad college of Engineering, Vadgaon (Bk.), Pune-411041
Email: [1]songireprasanna11@gmail.com, [2]kvmali.scoe@sinhgad.edu

Abstract- This study deals with comparison of synthetic and natural refrigerants in cascade refrigeration system for low temperature application. Synthetic refrigerants have been prominently used in all refrigeration applications due to their favourable thermodynamic properties. CFCs are phased out; HCFCs are scheduled to phase out following to Montreal protocol and its subsequent amendments. Thus, natural refrigerants being eco-friendly are reviewed as alternative to synthetic refrigerants. Theoretical analysis and comparison of various refrigerant pairs such as R507-R23,R717-R23, R290-N_2O, R717- N_2O. System performance is estimated with variation in evaporating, condensing temperature, isentropic efficiency of compressor, temperature overlap in cascade condenser. Results shows that, COP of the system with R717-N_2O is higher than other pairs and it can be used as alternative refrigerant for low temperature application in cascade refrigeration system.

Keywords-COP, Cascade refrigeration system, Synthetic refrigerants, Alternative refrigerant.

I. INTRODUCTION

Many industrial applications requires low temperature refrigeration such as quick freezing, biomedical preservations, manufacturing of dry ice, liquefaction of petroleum vapours, pharmaceutical reactions etc. where evaporating temperature requires between -40ºC to-80ºC. Condensing temperature is governed by temperature of cooling tower water which is about 35 ºC. Thus, system has to work for wide range of temperature. Single stage vapour compression system is not feasible for such application and its performance decreases below -35 ºC. Multistage or compound systems can be useful but no refrigerants available to work efficiently for high temperature lift. Also, it will be difficult to balance the oil level in compressor because of large difference in suction pressures of low stage and higher stage compressors. Cascade refrigeration system has two different stages which permits appropriate selection refrigerants to maximise system performance. Synthetic refrigerants prominently used in till now due to their excellent thermodynamic properties but owing to higher ODP (Ozone Depletion Potential), GWP (Global worming Potential) they are contributor to ozone depletion and global warming. Following to Montreal protocol and subsequent amendments CFCs are already phased out and HCFCs are scheduled to phase out[1]. Thus, there is need to seek alternative refrigerants with no ODP and minimum GWP.

Very few refrigerants are suitable to use upto -80 ºC. HFC-23 considered as replacement of CFC-13. CO_2 is also preferred at low temperature cycle but, the triple point of it is -56 ºC hence can't be used below it. M.Alhamid et. al. performed exergy and energy analysis using mixture of R744+R170 which enable for system to work upto -85 ºC with reduction in flammability risk associate with hydrocarbons [2].A. D. Parekh and P. R. Tailor did thermodynamic analysis for R507A-R23 pair in cascade refrigeration system[3]. HFC-507A is suggested as replacement of HCFC-502 by manufacturer [4]. The HFC pair selected as ozone friendly refrigerants. The analysis is carried out to optimise design and operating parameters. In energy and irreversibility analysis of a cascade refrigeration system for various refrigerant couples by Kilicarslan et.al., refrigerantpairs R717–R23 has the highest COP and lowest irreversibility, while R507–R23 has the lowest COP and highest irreversibility among the selected refrigerant pairs[5]. In the paper by Kruse et.al. theoretical investigation performed for cascade refrigerating systems using existing refrigerant pair R23-R134a verses N_2O in LTC and various natural refrigerants -ammonia, propane, carbon dioxide and nitrous oxide itself for HTC[6]. From the results, N_2O at LTC and ammonia or hydrocarbons as refrigerants at the secondary stage in refrigerating systems achieves similar COPcompared to the R23-R134a cascade refrigerating system. Yingbaiet.al.shows that conventional synthetic refrigerant pair R22-R23 can be replaced by natural hydrocarbon refrigerant pair R290-R170 in cascade refrigeration cycle[7].Souvik Bhattacharyya et.al. investigated thermodynamic analysis of N_2O-CO_2 cascade refrigeration system. Result shows that internal heat exchanger has marginal influence on system performance also since N_2O and CO_2 has similar thermodynamic properties, performance of the system is same when fluids swapped in LT and HT cycle[8].

In present paper thermodynamic analysis is done for refrigerant pairs mentioned early in order to find substitute for synthetic refrigerants. Variable parameters selected condenser, evaporator temperature, temperature difference in cascade condenser and isentropic efficiency of compressors.

II. CASCADE REFRIGERATION SYSTEM

Two stage cascade refrigeration system is represented by a line diagram, P-h and T-s diagram in Fig.3.1, 3.2 and 3.3 respectively. In the system both Low Temperature Cycle (LTC) and High Temperature Cycle (HTC) work with different refrigerants and thermally connected to each other through a heat exchanger which acts as an evaporator for the HTC and a condenser for the LTC. HTC operates with refrigerant having high boiling point and high critical temperature and LTC operates with refrigerant having low boiling point. Properties of refrigerants are given in Table I. Fig.1 shows that the condenser rejects heat Q_{HT} from the condenser at condensing temperature of Tc to its condensing medium or environment. The useful refrigerating effect is produce in evaporator of LTC by absorbing the cooling load Q_{LT} from the cooling space at the evaporating temperature Te. Heat absorbed by LTC evaporator and work input to LTC compressor equals the heat absorbed by HTC evaporator that is cascade condenser. Tc,cas and Te,cas represent the condensing and evaporating temperatures respectively. The temperature difference between them, ΔT= Tc,cas – Te,cas is called temperature overlap which is necessary for heat transfer.

III. THERMODYNAMIC ANALYSIS

Thermodynamic analysis of the system didbase on the following general assumptions:

i) Compression process in both stage is adiabatic with isentropic efficiency is 0.8.

ii) Pressure drop in piping, heat exchanger or system components is negligible.

iii) Expansion processes are isenthalpic in both stages.

Table I.Thermodynamic properties of refrigerants

	N_2O	R23	R507A	R717	R290
Boiling Point(°C)	-88.5	-82.1	-46.7	-33.3	-42.1
Critical Temp(°C)	36.4	25.6	70.9	132.3	96.8
Critical Pressure (MPa)	7.25	4.86	3.79	1.33	4.97
Molecular mass (kg/Kmol)	44.01	70.01	98.86	77.65	44.1
ODP	0	0	0	0	0.05
GWP	280	14800	3985	0	3

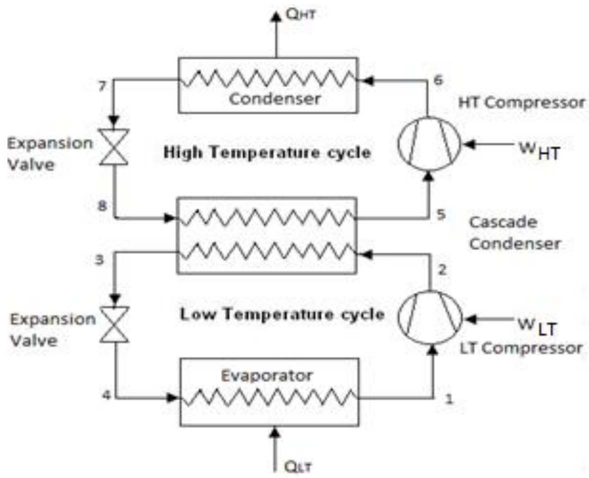

Fig.3.1 Cascade refrigeration system

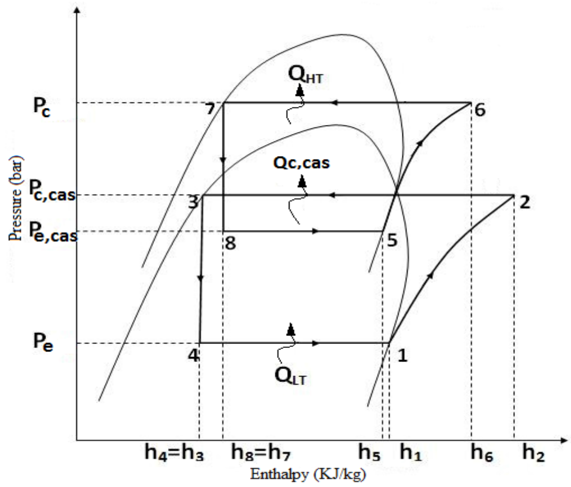

Fig. 3.2 Pressure-Enthalpy diagram

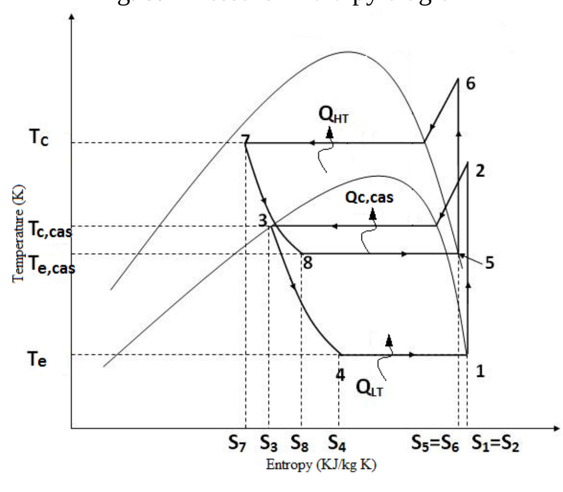

Fig.3.3 Temperature-Entropy diagram

Steady flow energy equation for calculations is given as:

Rate of heat absorbed by LT evaporator,$Q_{LT}=m_{LT}(h_1-h_4)$ (1)

Compressor powerby HT cycle, $W_{HT}=m_{HT}(h_6-h_5)$ (2)

Compressor power by LT cycle,$W_{LT}= m_{LT}(h_2- h_1)$ (3)

Rate of heat transfer in the cascade heat exchanger,

$Q_{CAS}= m_{LT} (h_2-h_3)= m_{HT}(h_5-h_8)$ (4)

Heat rejection rate by the condenser,$Q_{HT}= m_{HT}\ (h_6-h_7)$(5)

COP of low temperature cycle,$COP_{LT}=Q_{LT}\ /W_{LT}$ (6)

COP of high temperature cycle,$COP_{HT}=Q_{CAS}/W_{HT}$ (7)

Overall COP of the system,$COP=Q_{LT}/(W_{HT}+W_{LT})$ (8)

Above equations formulated in excel sheet and COP of system can be calculated with the help of thermodynamic properties like enthalpy at corresponding saturation pressure and temperature.

IV. RESULTS AND DISCUSSION

Effect of particular parameter on the performance of system is investigated by varying only that parameter keeping rest of parameters constant. Variable parameters considered as evaporating, condensing temperature, temperature difference in cascade condenser that is temperature overlap and isentropic efficiency of compressors. It varied in the ranges given below:

- Low temperature cycle evaporator temperature varied from -80°c to -55°c.
- High temperature cycle condensing temperature varied from 30°c to 40°c.
- Temperature overlap, ΔT is varied from 3°c to 15°c.
- Isentropic efficiency of both stage compressors varied as, ηisen = 0.7 to 0.9.

For variation in any temperature, cascade condenser temperature is considered at optimum value, $[Tc, cas]opt = \sqrt{Tc * Te}$ at which COP of the system is maximum. Parameters assumed constant while varying particular one considered for the computation are given below:

- Cooling capacity=1000W
- LTC evaporating temperature, Te = -80°c.
- HTC condensing temperature, Tc =35°c.
- Cascade condenser temperature, Tc,cas = -30°c.
- Temperature overlap in cascade condenser, (ΔT) = 8°c.
- Degree of superheating, (ΔT)sup = 0°C in both HT and LT cycles.
- Degree of subcooling, (ΔT)sub = 0°C in both HT and LT cycles.
- Isentropic efficiency = 0.8 in both HT and LT compressors.

Effect on COP of system with variation of evaporating temperature, keeping rest of parameters constant is shown in Fig.4.1. As evaporating temperature increases from -80°C to -55°C, pressure ratio in LTC compressor decreases which decreases specific work of compressor and power consumption. COP of system increases for all refrigerant pairs. Similarly, effect of variation of condensing temperature on the cop of system can be shown in Fig. 4.2. As condensing temperature increases from 30°C to 40°C, pressure ratio of HTC compressor increases which increases compressor work and COP of the system decreases for all refrigerant pairs. In both cases that is variation in condensing and evaporating temperature, COP is highest for R717-N_2O pair and lowest for R507A-R23 pair. COP is slightly lower for R290-N_2O than R717-N_2O pair.

Effect of variation of temperature difference in cascade condenser on COP is as shown in Fig.4.3 keeping rest of

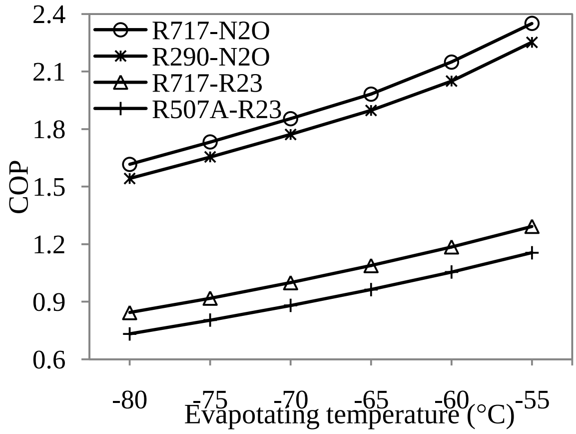

Fig.4.1 Effect of evaporating temperature on COP of system

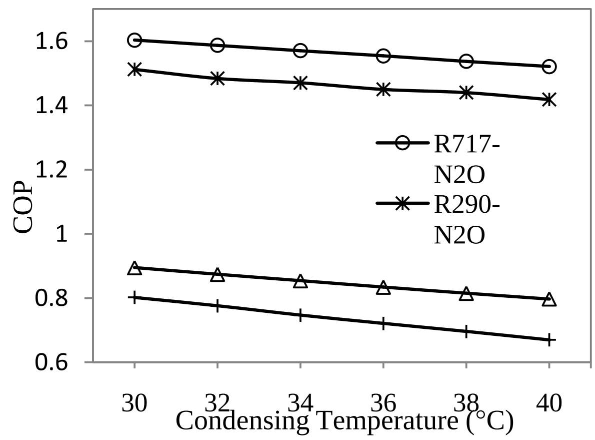

Fig.4.2 Effect of condensing temperature on COP of system

parameter constant. Temperature difference can be vary by decreasing cascade evaporator temperature and keeping cascade condenser constant. As cascade evaporator temperature decreases compressor work for HTC increases per unit amount of refrigerant which decreases COP of the system with no change in refrigerating effect in LTC evaporator. Again, R717-N_2O pair has highest COP; R290-N_2O has slightly lesser than it while R507-R23 pair has lowest among all selected pairs. This trend is found to be similar for variation of all parameters.

Further, for variation of isentropic efficiency of compressors, refrigerant pairs R290-N_2O and R717-R23 don't taken into account. As R717-N_2O has highest COP and objective of study to find replacement of R507-R23.

Thus performance of both pairs compared for change in isentropic efficiency of compressors. Actual compression process always accomplished by friction due to which process becomes irreversible. Isentropic efficiency measures deviation of actual process from ideal. As friction occurs during compression, isentropic efficiency of compressor decreases and enthalpy of gas at end of compression increases.

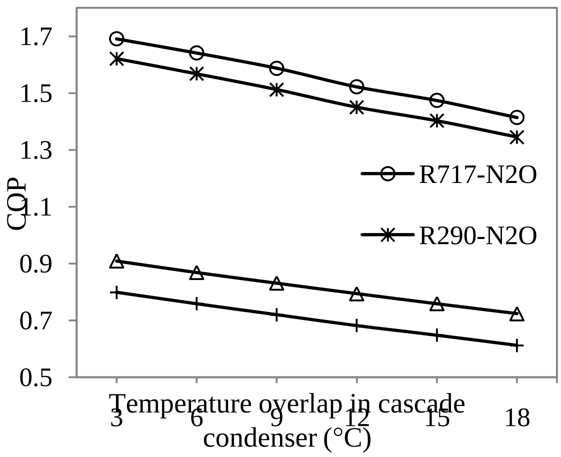

Fig.4.3 Effect of temperature difference in cascade condenser on COP of system

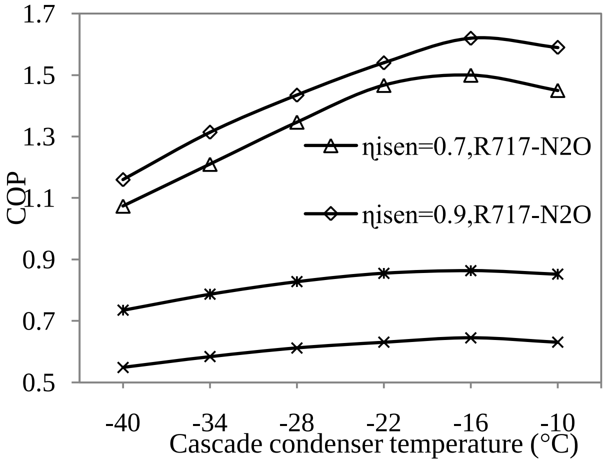

Fig.4.4 Effect of variation of isentropic efficiency on COP of system

Fig.4.4 shows effect of variation of isentropic efficiency of both LTC and HTC compressors on COP of system with variation of cascade condenser temperature from -10°C to -40°C. COP of the system increases with increase in isentropic efficiency for both pairs. COP is higher using R717-N_2O at 70% isentropic efficiency than using R507A-R23 pair at 90% isentropic efficiency. COP increases progressively with increase in cascade condenser temperature. It reaches to maximum at optimum value and decreases further with increase in cascade condenser temperature.

V. CONCLUSIONS

- ♦ Performances of various refrigerant pairs are compered by varying key operating parameters to check alternatives for synthetic refrigerants in cascade refrigeration system. R717-N_2O pair has the maximum COP for variation of all parameters consideredand R507A-R23 has lowest COP among refrigerant pairs considered.
- ♦ It can be stated that by substituting R507A-R23 having GWP about 3985 and 14,800 respectively by R717-N_2O having 240 GWP for N_2O,higher energetic performances can be achieved. However, R717 is toxic, R290 is flammable in nature and safety properties of N_2O are yet to study thoroughly hence these refrigerants must be used considering all safety precautions and suitability for particular applications.
- ♦ COP of cascade refrigeration system increases with increase in evaporation temperature. It decreases with increase in condensation temperature and increase in temperature difference in cascade condenser.
- ♦ COP of the system increases with increase in isentropic efficiency of the compressor and it is maximum at optimum cascade temperature.

NOMENCLATURE

CFC Chloro fluorocarbon

COP Coefficient of performance

GWP Global warming potential

HCFC Hydrochloro fluorocarbon

HFC Hydro fluorocarbon

HTC High temperature cycle

h Enthalpy (kJ/kg)

LTC Low temperature cycle

m mass flow rate (kg/min)

ODP Ozone depletion potential

P Power (W)

Q Heat transfer rate (W)

R Refrigerant

ΔT Temperature difference (°C)

T Temperature (°C)

S Entropy (kJ/kg K)

W Work done (W)

η Eefficiency

Subscript

cCondenser

eEvapoartor

CAS Cascade heat exchanger

HTHigh temperature stage

LT Low temperature stage

isenIsentropic

sub Subcooling

sup Superheating

REFERENCES

[1] Montreal protocol on substances that deplete the ozone layer. United Nations Environment programme (UNEP), 1987.

[2] Alhamid M., Syaka D and Nasruddin,' Exergy and Energy Analysis of a Cascade Refrigeration System Using R744+R170 for Low Temperature Applications' International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS Vol: 10 No: 06.

[3] DuPont Suva refrigerants-Technical information of properties and applications of R-507 Refrigerant.

[4] Parekh A. D. and Tailor P. R.,' Thermodynamic Analysis of R507A-R23 Cascade Refrigeration System' International Journal of Aerospace and Mechanical Engineering 6:1 2012

[5] Kilicarslan A. andHosoz M.,'Energy and irreversibility analysis of a cascade refrigeration system for various refrigerant couples' Energy Conversion and Management 51 (2010) 2947–2954.

[6] Kruse H. and RussmannH.,' The natural fluid nitrous oxide-an option as substitute for low temperature synthetic refrigerants' International Journal of Refrigeration 29 (2006) 799-806.

[7] Xie Y., Liu C., LunL. and XuedongZ.,"Use of R290/R170 in Lieu of R22/R23 in Cascade Refrigeration Cycle". International Refrigeration and Air Conditioning Conference.Paper 939, 2008.

[8] Bhattacharyya S., Gorai A., SarkarJ.,' Thermodynamic analysis and optimisation of a novel N_2O-CO_2 cascade system for refrigeration and heating' International Journal of refrigeration 32(2009) 1077-1084.