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# Thermodynamic Analysis and Optimization of $N_2O$-$NH_3$ Cascade System for Low Temperature Refrigeration

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**Abstract**—The present work deals with thermodynamic analysis of cascade refrigeration system for low temperature range -50 °C to -85 °C. First comparison is done regarding COP by using nitrous oxide in low temperature circuit and different fluids such as R717, R290, R1290, R134a and an azeotropic mixture R507A in high temperature circuit as refrigerants. In this study thermodynamic analysis of $N_2O$-$NH_3$ cascade refrigeration system is done, to optimize the design and operating parameters of the system for fixed temperature difference in the cascade heat exchanger. The design and operating parameters include condenser, evaporator, subcooling and superheating temperatures in the, HTC and LTC. At the end, three useful correlations that yield the optimal condensing temperature in cascade heat exchanger, the associated maximum COP, and optimum mass flow rate ratio are presented.

**Index Terms**—Natural Refrigerant, Cascade System, Nitrous Oxide, Ammonia

## 1 Introduction

MANY industrial applications like food storage, liquefaction of petroleum vapour, manufacturing of dry ice and storage of blood etc., required low temperature refrigeration in the temperature range from -30 °C to -100 °C. So for in this temperature range refrigeration not possible by single stage vapour compression system, multistage compression systems is other option but it is not efficient due to limitations of refrigerants and some other factor. These limitations can be overcome by using cascade systems which employ more than one refrigerant. A refrigeration system in which series of single stage units are thermally coupled through cascade heat exchanger is known as cascade refrigeration system. In cascade system each cycle has a different refrigerant, the lower temperature units progressively using lower boiling point refrigerants. There have been many analytical and experimental studies done so far on the performance analysis of the cascade system and most of them are on R744/R717 refrigerants pair [1-6].

Triple point of Carbon dioxide is -56.56 °C so evaporation below this temperature is not possible by using $CO_2$. Further low temperature can be achieved by the natural fluid $N_2O$ which has a triple point (-90.82 °C) and NBP (-88.47 °C) [7]. Another advantage with $N_2O$ is that it is five times lower toxicity than $CO_2$. However its GWP is significantly higher than $CO_2$, it still falls under the low GWP category that is currently considered to have an upper limit of 300 [8].

For refrigeration below -50 °C, a theoretical investigation reported on cascade refrigeration systems by Kruse and Russmann (2006) using $N_2O$ as refrigerant in LTC and various natural refrigerants in HTC. Bhattacharyya et al. (2009) reported a thermodynamics analysis of Cascade system with carbon dioxide in HTC as transcritical cycle and nitrous oxide in LTC for simultaneous heating and cooling applications. Other transcritical cascade system with R290-R744 and R1270-R744 are also available in literature [10, 11].

The present study argues that if heating load is not required use of transcritical cascade system is not justified due to high working pressure, temperature and cost. In this work thermodynamic analysis of the cascade system for low temperature range -50 °C to -85 °C is presented.

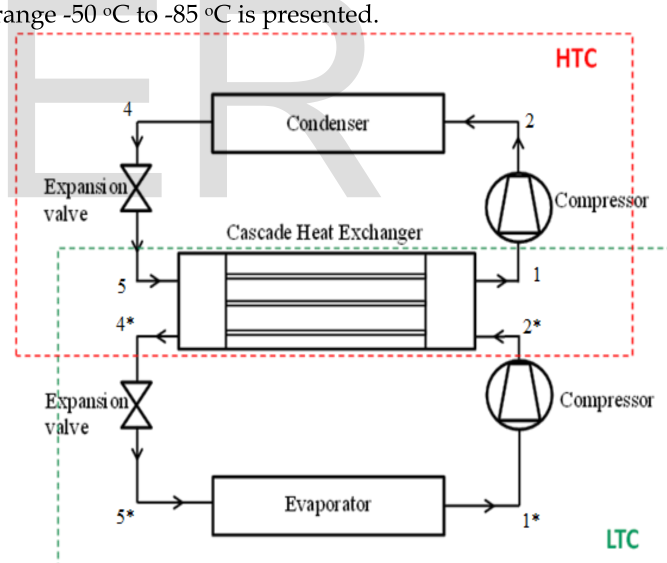


Fig. 1. Schematic diagram of a $N_2O$-$NH_3$ cascade refrigeration system

## 2. System Description

A schematic diagram of a two-stage cascade system for refrigeration is shown in Fig. 1. This system comprises two separate circuits: the high temperature circuit (HTC) using $NH_3$ and low temperature circuit (LTC) using $N_2O$ as a refrigerant, which are thermally coupled by cascade heat exchanger. In cascade heat exchanger, ammonia evaporates and nitrous oxide condenses by transferring their heat to each other. The corresponding T–s diagram for the cascade refrigeration system is shown in Fig. 2.

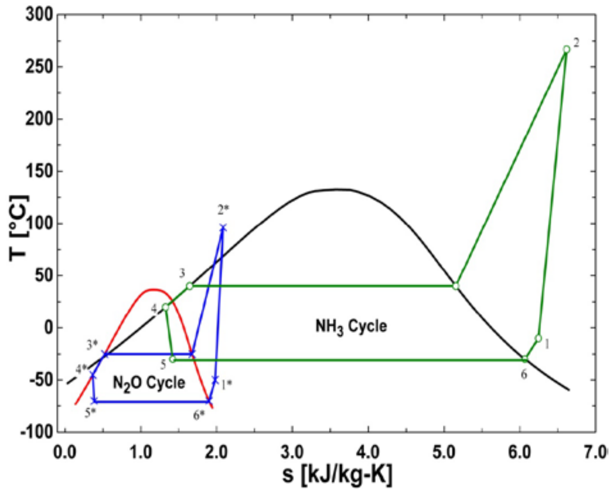


Fig. 2. T-s diagram of a N2O-NH3 cascade refrigeration system

## 3. THERMODYNAMIC ANALYSIS

The thermodynamic analysis of $N_2O$-$NH_3$ cascade refrigeration system performed based on the following assumptions:

- Compression process is adiabatic with an isentropic efficiency of 0.70;
- The expansion process is isenthalpic;
- Negligible heat losses or gains in the pipe lines or system components;
- Negligible changes in kinetic and potential energy;
- Temperature difference in the cascade heat exchanger is 5 $^oC$.

The thermo-physical properties of $N_2O$ and $NH_3$ specified in this work were calculating using a software package called engineering equation solver (EES) [12].

The cycle modelled by applying steady flow energy equation and mass balance equation for each individual process of the cycle. The equations for the different components of the cascade refrigeration system are given in the Table 1.

TABLE 1
MASS AND ENERGY BALANCE FOR THE CASCADE SYSTEM

| Component | Mass balance | Energy balance |
|---|---|---|
| LTC expansion valve | $\dot{m}_L = \dot{m}_{4*} = \dot{m}_{5*}$ | $h_{4*} = h_{5*}$ |
| Evaporator | $\dot{m}_L = \dot{m}_{5*} = \dot{m}_{1*}$ | $\dot{Q}_E = \dot{m}_L(h_{1*} - h_{5*})$ |
| LTC compressor | $\dot{m}_L = \dot{m}_{1*} = \dot{m}_{2*}$ | $\dot{W}_L = \dot{m}_L(h_{2*} - h_{1*})$ |
| Cascade heat exchanger | $\dot{m}_L = \dot{m}_{2*} = \dot{m}_{4*}$<br>$\dot{m}_H = \dot{m}_5 = \dot{m}_2$ | $\dot{Q}_{hx} = \dot{m}_L(h_{2*} - h_{4*})$<br>$= \dot{m}_H(h_1 - h_5)$ |
| HTC compressor | $\dot{m}_H = \dot{m}_1 = \dot{m}_2$ | $\dot{W}_H = \dot{m}_H(h_2 - h_1)$ |
| Condenser | $\dot{m}_H = \dot{m}_2 = \dot{m}_4$ | $\dot{Q}_C = \dot{m}_H(h_2 - h_4)$ |
| HTC expansion valve | $\dot{m}_H = \dot{m}_4 = \dot{m}_5$ | $h_4 = h_5$ |

The COP of the system can be express as:

$$COP=\frac{(COP_{LTC})(COP_{HTC})}{1+COP_{LTC}+COP_{HTC}} \quad (1)$$

$$COP_{LTC}=\frac{\dot{Q}_E}{\dot{W}_L} \quad (2)$$

$$COP_{HTC}=\frac{\dot{Q}_{hx}}{\dot{W}_H} \quad (3)$$

## 4. RESULTS AND DISCUSSION

The results of the present analysis have been given in the following sections.

### 4.1 Selection of high temperature circuit refrigerant

Five refrigerants are chosen namely: R717, R290, R1270, R134a and R507A for HTC of cascade system. Among these refrigerants, first three are natural refrigerants and rest two are the synthetic refrigerants. These refrigerants are compared at evaporator temperature of -80 $^oC$ and condenser temperature 40 $^oC$. COP of the cascade system with different refrigerant in HTC is compared at different coupling temperature ($T_{CT}$) i.e. condensing temperature of $N_2O$ in cascade heat exchanger, for different degree of subcooling and superheating.

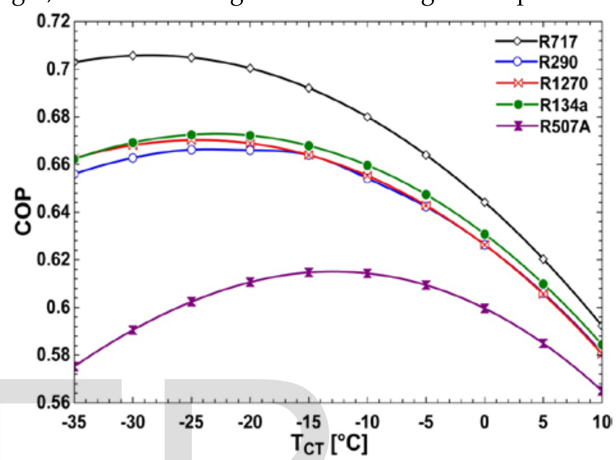


Fig. 3. Variation of COP with coupling temperature for a subcooling of 0 K and superheating of 0 K



Variation of COP with coupling temperature for 0 degree subcooling and superheating in both cycle (HTC and LTC) has been shown in Fig. 3. It is observed from the figure that the COP of the system with ammonia (R717) is the highest among all the refrigerants and with R507A, it is lowest. For rest three refrigerants COP lies in between, performance with R290, R1290 is nearly similar, which is slightly low than R134a.

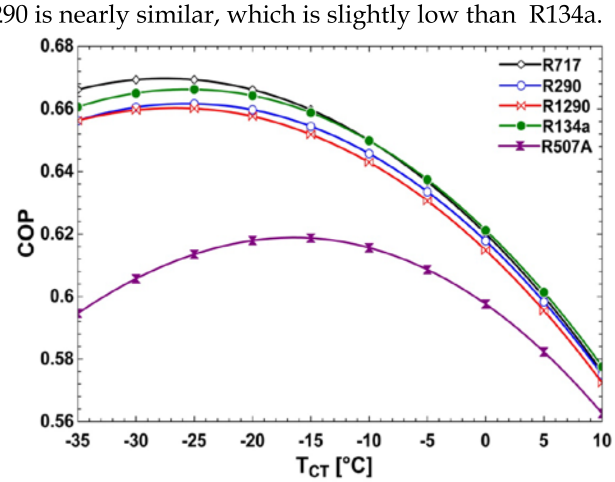


Fig.4. Variation of COP with coupling temperature for subcooling of 0 K and superheating of 20 K in both circuits

Variation in COP with 0 degree subcooling and 20 degree superheating is shown in Fig. 4. Performance of the system is still higher for ammonia followed by R134a but for coupling temperature above -15 oC, the performance of the system is almost same for both refrigerants. With 20 degree of superheating, performance of all refrigerants except R507A, decreases as compared to 0 degree superheating. However, R507A still has lowest performance in cascade system.

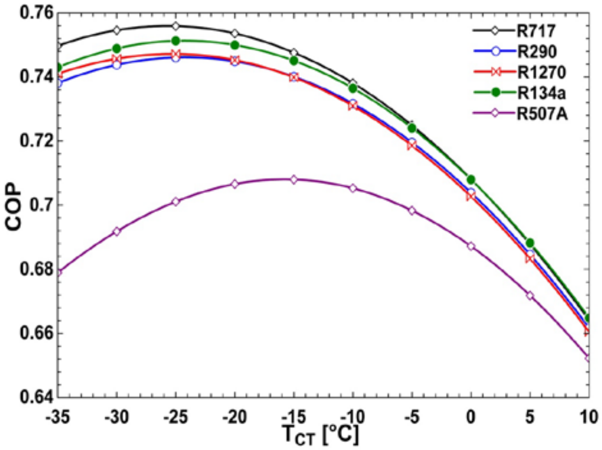


Fig. 5. Variation of COP with coupling temperature for a subcooling of 10 K and superheating of 0 K

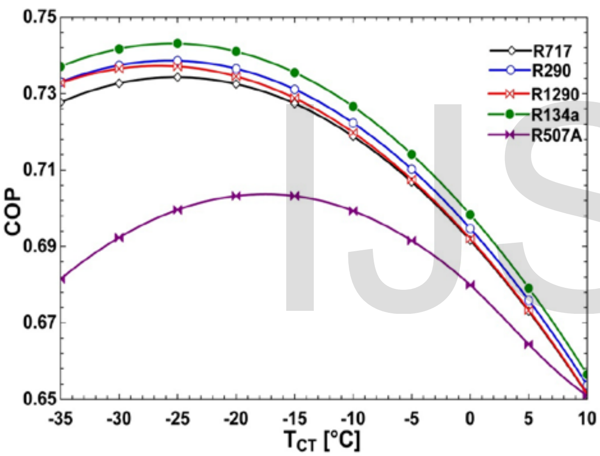


Fig.6. Variation of COP with coupling temperature for a subcooling of 10 K and superheating of 10 K

Fig.5 shows that subcooling in both the circuit has increasing effect on COP of the cascade system for all refrigerants. For this set of condition, performance of system is highest for ammonia and lowest for R507A. Fig. 6 shows the variation of COP with coupling temperature for 10 degree subcooling and 10 degree superheating. For given operating condition, performance of cascade system is highest for R134a followed by R717, R1270, R290, R507A.

From the above discussion, it can be concluded that, performance of cascade system with R507A in HTC is worse, while R717 and R134a have excellent performance. Hydrocarbons (Propane and Propylene) performed similarly in the cascade system and the performance of hydrocarbons based system is near about to R134a. However, the hydrocarbons are flammable and R134a has very high value of GWP (1300), while ammonia has zero ODP and zero GWP. Therefore ammonia is selected for further analysis in HTC with $N_2O$ in LTC of a cascade refrigeration system. Ammonia has excellent thermodynamic properties, and high heat transfer coefficients. However ammonia is toxic in nature, but this risk is somewhat mitigated by its pungent smell, even at lower concentrations (5 ppm) it is self-alarming in the event of a leak [13].

### 4.2 $N_2O$-$NH_3$ Cascade System

The analysis for the $N_2O$-$NH_3$ cascade refrigeration system is done at various operating conditions such as coupling temperature, various degree of subcooling /superheating, evaporating, and condensing temperature.

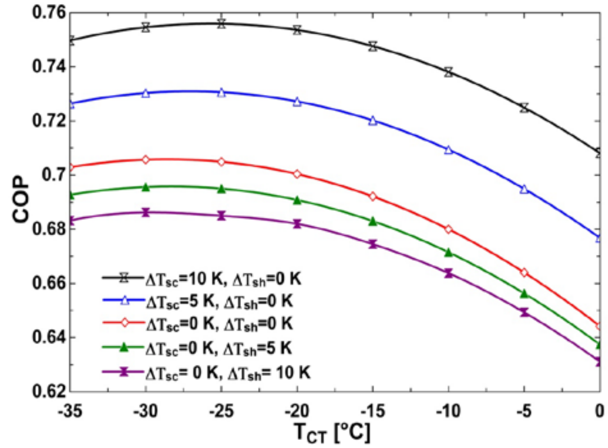


Fig. 7.Variation of COP with coupling temperature



Fig. 7 shows variation of COP with coupling temperature at different degree of subcooling and superheating at TC=40 $^{o}$C and TE= -80 $^{o}$C. It is observed from the figure that system performance first increases with coupling temperature; attains maximum value and after that it decreases. Therefore, for given operating condition COP is maximum at a particular coupling temperature which is known as optimum coupling temperature. Fig. 7 also shows the effect of degree of subcooling and superheating on the performance of cascade system. As the degree of subcooling increases, the optimum coupling temperature (for maximum COP) shifts rightward. The degree of superheating has negligible effect on the optimum coupling temperature.

Fig. 8 shows the variation of maximum COP (at optimum coupling temperature) of the system with condenser temperature for various degrees of subcooling and superheating at fixed evaporator temperature of -80 $^{o}$C. As the condenser temperature increases, the pressure ratio in HTC increases which results in increase of optimum coupling temperature so the pressure ratio in both cycles increases hence the maximum COP of the system decreases for fixed evaporator temperature. With increases in degree of subcooling the maximum COP of the system increases, however increase in degree of superheating the maximum COP of the system decreases, with respect to 0 degree of subcooling and 0 degree of superheating.

Fig. 9 shows the variation in maximum COP of the system with evaporator temperature for various degree of subcooling and superheating at fixed condenser temperature of 40 $^{o}$C. As the evaporator temperature increases, the pressure ratio in LTC decreases which results in increase of optimum coupling temperature so pressure ratio in both cycle decreases hence the maximum COP of the system increases for fixed condensing temperature. With increases in degree of subcooling the maximum COP of the system increases, however increase in degree of superheating the maximum COP of the system decreases, with respect to 0 degree of subcooling and 0 degree of superheating.

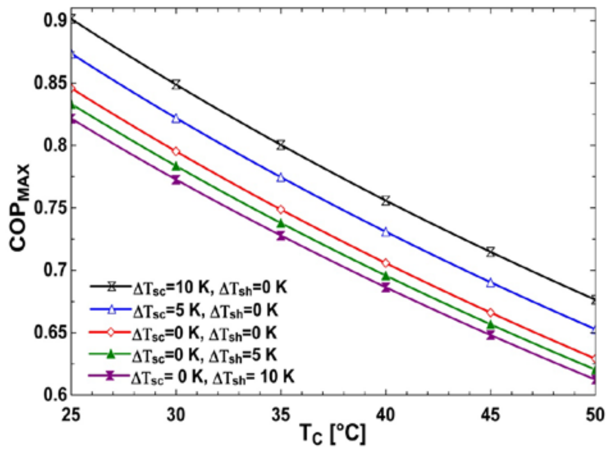


Fig. 8. Variation of maximum COP with condenser temperature

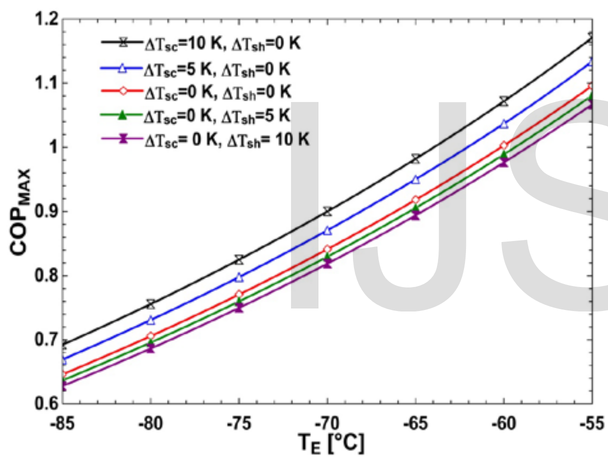


Fig. 9. Variation in maximum COP with evaporator temperature

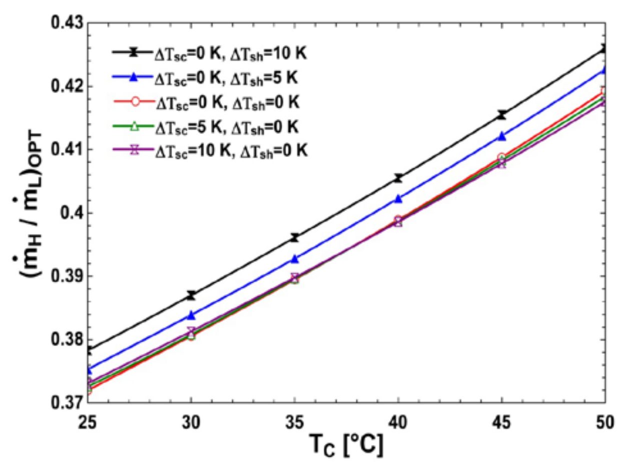


Fig. 10. Variation in optimum mass flow rate with condenser temperature

Fig. 10 shows the variation in optimum mass flow rate ratio with condensing temperature. The mass flow rate ratio at optimum coupling temperature (or maximum COP) is known as optimum mass flow rate ratio. It is an important design parameter. It becomes more important for this system because ammonia is toxic in nature. As the condenser temperature increases, the optimum mass flow rate ratio increases for fixed evaporator temperature -80 °C. Fig. 11 shows the variation in optimum mass flow rate ratio with evaporator temperature. As the evaporator temperature increases, the optimum mass flow rate ratio decreases for fixed condenser temperature 40 °C.

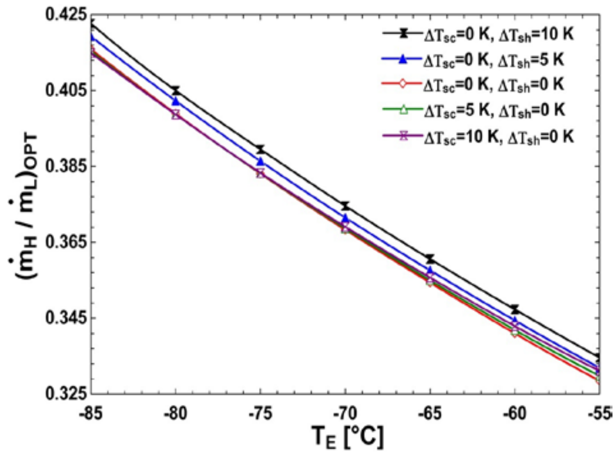


Fig. 11. Variation in optimum mass flow rate ratio with evaporator temperature



It can be observed from Fig. 10 and Fig. 11 that the optimum mass flow rate ratio increases with increase in degree of superheating however degree of subcooling has negligible effect.

### 4.3 Optimization

To establish a correlation for optimum conditions, a large database has been generated by cyclic simulation for fixed temperature difference in cascade heat exchanger and compressor isentropic efficiency of 70 % for a range of evaporation temperatures from –55 °C to -85 °C and condenser from 25 °C to 45 °C. Performing a regression analysis on the data, the following relations have been established to predict estimates of the optimum design parameter.

$$T_{CT,OPT} = a_o + a_1 T_E + a_2 T_C + a_3 \Delta T_{Sh} + a_4 \Delta T_{sc} \quad (4)$$

$$COP_{MAX} = a_o + a_1 T_E + a_2 T_C + a_3 \Delta T_{Sh} + a_4 \Delta T_{sc} \quad (5)$$

$$(\dot{m}_H/\dot{m}_L)_{OPT} = a_o + a_1 T_E + a_2 T_C + a_3 \Delta T_{Sh} + a_4 \Delta T_{sc} \quad (6)$$

TABLE 2
REGRESSION COEFFICIENT

| | $T_{CT,OPT}$ | $COP_{MAX}$ | $(\dot{m}_H/\dot{m}_L)_{OPT}$ |
|---|---|---|---|
| $a_o$ | -3.200325 | 2.499316 | 0.1109522 |
| $a_1$ | 0.4562245 | 0.01667982 | -0.002777139 |
| $a_2$ | 0.2762245 | -0.0118638 | 0.001618531 |
| $a_3$ | 0.04650216 | -0.002622088 | 0.000628450 |
| $a_4$ | 0.2982165 | .006107283 | 0.000120359 |
| $R^2$ | 99.62% | 98.20% | 99.74% |
| rms | 0.31447 | 0.025564 | 0.0021995 |

## 5. CONCLUSION

In this study, thermodynamic analysis of cascade refrigeration system is presented at -80 °C evaporating temperature and 40 °C condensing temperature by using nitrous oxide in low temperature circuit (LTC) and different fluids such as R717, R290, R1290, R134a and an azeotropic mixture R507A in high temperature circuit (HTC) as refrigerants. This analysis lead to a result that performance of ammonia in HTC with nitrous oxide in LTC is best whereas the azeotropic mixture R507A perform worse.

The analysis of $N_2O$-$NH_3$ cascade system is concluded that the subcooling in both cycles is always desirable, since it increases COP of the system and does not affect the mass flow ratio considerably. Superheating has adverse effect on COP and increase the mass flow ration thus it is undesirable. To optimize the COP, a regression analysis has been developed that could be useful to refrigeration engineers for setting optimum thermodynamic parameters of nitrous oxide–ammonia cascade system.

Nomenclature:

| | | |
|---|---|---|
| h | specific enthalpy | (kJ/ kg) |
| $\dot{m}$ | mass flow rate | (kg/ s) |
| Q | heat transfer | (kJ) |
| s | specific entropy | (kJ/kg- K) |
| T | temperature | (°C) |
| $\dot{w}$ | specific work | (kJ/s) |
| Greek | | |
| η | efficiency | |
| Δ | change | |
| Subscripts | | |
| 1–6 | refrigerant state points in HTC | |
| 1*–6* | refrigerant state points in LTC | |
| C | condenser | |
| CT | coupling temperature | |
| E | evaporator | |
| hx | cascade heat exchanger | |
| H | higher | |
| HTC | high temperature circuit | |
| L | lower | |
| LTC | low temperature circuit | |
| MAX | maximum | |
| OPT | optimum | |
| sh | superheating | |
| sc | subcooling | |

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