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Thermodynamic Analysis of a Cascade Refrigeration System Based On Carbon Dioxide and Ammonia

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Abstract
Thermodynamic analysis of a cascade refrigeration system that uses carbon dioxide-ammonia (R744-R717) as refrigerant is presented in this paper to determine the optimum condensing temperature of the cascade condenser at given design parameters, to maximize the COP of the system. The design and operating parameters considered in this study include (1) condensing, sub cooling, evaporating and super heating temperatures in the ammonia (R717) high-temperature circuit, (2) temperature difference in the cascade heat exchanger, and (3) evaporating, superheating, condensing and sub cooling in the carbon dioxide (R744) low-temperature circuit. A multilinear regression analysis was employed in order to develop two useful correlations for maximum COP, and optimum condensing temperature.

Keywords: Cascade refrigeration system, Optimization, Coefficient of performance, Operating parameters, Correlation,

I. INTRODUCTION

In low temperature applications, including rapid freezing and the storage of frozen food, the required evaporating temperature of the refrigeration system ranges from -40°C to -55°C, so a single stage vapor-compression refrigeration system is insufficient. Two stage or cascade refrigeration systems are used for low temperature applications. The high and low pressure sides of a two stage refrigeration system are filled same refrigerant but the high and low temperature circuits of a cascade system are filled separately with appropriate refrigerants with respect to global environmental protection, the use of natural refrigerant in refrigeration systems has been demonstrated to be a complete solution to permanent alternative fluorocarbon-based refrigerant [1,2].Therefore, using natural refrigerants in both two stage and cascade refrigeration system helps to satisfy the obligation of environmental treaties. Ammonia (R717) is a natural refrigerant that is most commonly adopted in low-temperature two stage refrigeration systems, but it has disadvantages. For instance, ammonia has a pungent smell; it is toxic and moderately flammable.

Additionally, the evaporating pressure of ammonia system is below atmospheric pressure, when the evaporating temperature is below -35°C, causing air to leak in to the system, leading to short-term inefficiency and the long term unreliability of the system. Hence, a non toxic, non flammable and dense refrigerant gas with a positive evaporating pressure should be chosen for evaporation below -35°C.A cascade refrigeration system with natural refrigerants CO_2 and NH_3 meets these requirements. Hence, a non toxic, non flammable and dense refrigerant gas with a positive evaporating pressure should be chosen for evaporation below -35°C. A cascade refrigeration system with natural refrigerants CO_2 and NH_3 meets these requirements. A CO_2/NH_3 cascade refrigeration system uses carbon dioxide and ammonia as refrigerants in low and high temperature circuits, respectively. Some of the characteristics of CO_2 make it a good alternative to ammonia for use in large -scale refrigeration plants operated at low temperature. The most obvious advantages of carbon dioxide are that it is non toxic, incombustible and has no odour. More over as compared with ammonia two stage refrigeration- systems, the CO_2/NH_3 cascade refrigeration system has significantly lower amount of ammonia, and the COP of the cascade system exceeds that of a two stage system at low –temperatures [3,5,7]. Therefore CO_2/NH_3 cascade refrigeration systems are attracting attention all over the world [3,6-10].

In the design phase of a CO_2/NH_3 cascade refrigeration system, an important issue is the means of determining the optimum condensing temperature of a cascade –condenser under particular design conditions, such as condensing temperature, evaporating temperature and the temperature difference between the high and low temperature circuits in cascade condenser.

A small concern with cascade refrigeration system is the initial installation cost which is about 10% higher than the traditional direct expansion systems(Wilson and Maier,2006) . But this cost can be negated with less refrigerant charge requirements and environmental advantage of the cascade system due to less direct emissions as compared to single stage system. An improvement, which can also be observed in cascade system is the reduced amount of superheat in the discharge temperature of the high temperature circuit that results in a reduced capacity of the high temperature condenser and an increased refrigeration effect (Ratts and Brown,2000).

The isentropic efficiency and volumetric efficiency are regarded here as a function of the pressure ratio of the compressor. Two correlations that are useful for determining the optimum condensing temperature of a cascade condenser and its corresponding maximum COP are presented, to support the development of CO_2/NH_3 cascade refrigeration systems and related equipment in the design phase.

II. System description

Fig.1 schematically depicts a CO_2/NH_3 cascade refrigeration system. Fig.2 presents the corresponding T-s and P-h diagrams. This refrigeration system comprises of two separate refrigeration circuits, the high temperature circuit (HTC) and the low temperature circuit (LTC). Ammonia is the refrigerant in HTC, where as carbon dioxide is the refrigerant in LTC. The circuits are thermally connected to each other through a cascade condenser, which acts as an evaporator for the HTC & as a condenser for LTC. Fig.2. Shows that the condensing and evaporating pressure in the NH3 circuit are both lower than those in the CO_2 circuit. So the NH_3 circuit is called HTC rather than the high pressure circuit, and CO_2 circuit is called the LTC rather than the low pressure circuit.

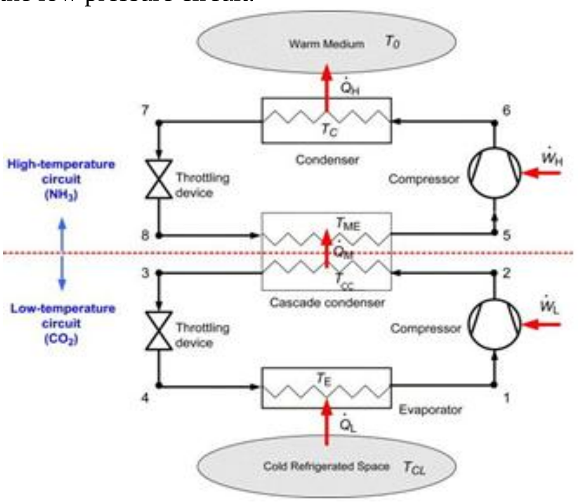

Fig.1.Schematic diagram of a CO_2/NH_3 cascade refrigeration system

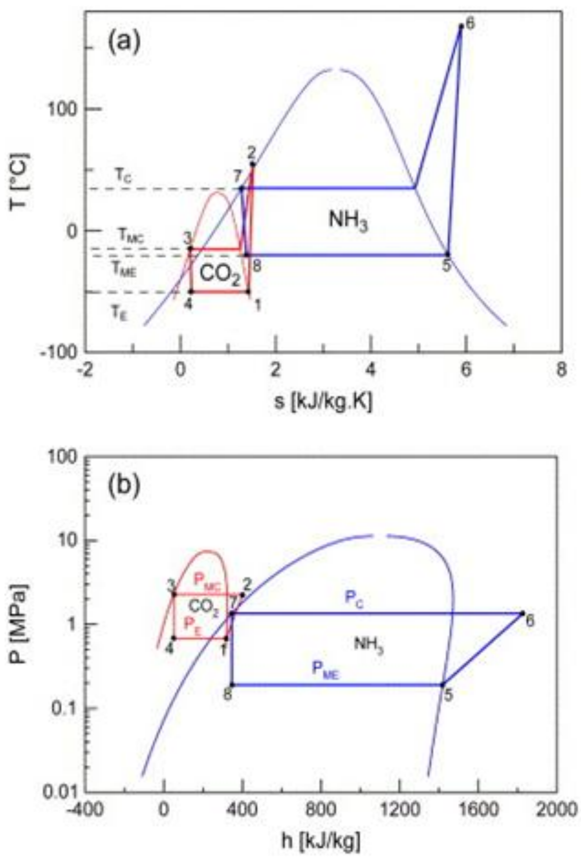

Fig.2 (a): The T-s and (b) the p-h diagrams of a CO_2/NH_3 cascade refrigeration system.

Fig.1.indicates that the condenser in the cascade refrigeration system rejects a heat of QH at condensing temperature of Tc, to its warm coolant or environment at temperature To. The evaporator of this cascade system absorbs a refrigerated load QL from the cold refrigerated space at TcL to the evaporating temperature TE .The heat absorbed by the evaporator of the LTC plus the work to the LTC compressor equals the heat absorbed by the evaporator of the HTC. TCC and TME represent the condensing and evaporating temperature of the cascade-condenser, respectively. ΔT= (TCC-TME) 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 CO2/NH3 cascade refrigeration system.

III. Thermodynamic analysis

A parametric study with fixed cooling capacity, and various condensing temperatures, evaporating temperatures and temperature differences in the cascade-condenser has been conducted to determine the optimum condensing temperature of a cascade-condenser in a CO2/NH3 cascade refrigeration system operating at low temperatures. The condensing temperatures used in the parametric study are 35°C, 40°C and 45°C.The evaporating temperatures are -45°C, -50 °C, and -55°C. The temperature differences in the cascade-condenser are 3°C, 4°C and 5°C.Each component in the cascade refrigeration system, shown in fig.1 can be treated as a control volume.

Assumptions

The following assumptions are made to simplify the thermodynamic analysis.

1. All components are assumed to operate at a steady state .The changes in the potential and the kinetic energy of the working fluids across each components are negligible.
2. The high and low temperature circuit compressors are adiabatic but non isentropic, and their isentropic efficiency can be expressed as a function of pressure ratio.
3. Combined motor and mechanical efficiency of each compressor is assumed to be 0.93.
4. The heat loss and pressure drops in the piping connecting the components are negligible.
5. All throttling devices are isenthalpic.
6. The outlet states of the condenser and the cascade condenser are at sub cooled state and that of the evaporator is at superheated state.

Based on the assumptions described above, the balanced equations are applied to find the mass flow rate of each cycle, the work input to the compressor, the heat transfer rates of condenser, cascade condenser and evaporator.

Mass balance

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \tag{1}$$

Energy balance

$$\dot{Q} - \dot{W} + \sum_{in} \dot{m}h - \sum_{out} \dot{m}h = 0 \tag{2}$$

Table-1. Mass and energy balance equations of different components.

Component	mass balance
Energy balance	
HTC compressor	$\dot{m}_5 = \dot{m}_6 = \dot{m}_H$
$\dot{W}_H = \frac{\dot{m}_H(h_{6s} - h_5)}{\eta_s \eta_m \eta_e} = \frac{\dot{m}_H(h_6 - h_5)}{\eta_m \eta_e}$	
Condenser	$\dot{m}_6 = \dot{m}_7 = \dot{m}_H$
$\dot{Q}_H = \dot{m}_H(h_6 - h_7)$	
HTC throttling Device	$\dot{m}_7 = \dot{m}_8 = \dot{m}_H$
$h_7 = h_8$	
Cascade condenser	$\dot{m}_2 = \dot{m}_3 = \dot{m}_L$ $\dot{m}_8 = \dot{m}_5 = \dot{m}_H$
$\dot{Q}_M = \dot{m}_H(h_5 - h_8) = \dot{m}_L(h_2 - h_3)$	
LTC compressor	$\dot{m}_1 = \dot{m}_2 = \dot{m}_L$
$\dot{W}_L = \frac{\dot{m}_L(h_{2s} - h_1)}{\eta_s \eta_m \eta_e} = \frac{\dot{m}_L(h_2 - h_1)}{\eta_m \eta_e}$	
LTC throttling device	$\dot{m}_3 = \dot{m}_4 = \dot{m}_L$
$h_3 = h_4$	
Evaporator	$\dot{m}_4 = \dot{m}_1 = \dot{m}_L$
$\dot{Q}_L = \dot{m}_L(h_1 - h_4)$ $\dot{m}_L = \frac{\dot{Q}_L}{(h_1 - h_4)}$	

Compressor Efficiency

The isentropic and volumetric efficiencies of ammonia and carbon dioxide compressors can be expressed in terms of compression ratio, as

NH_3 Compressor [20] η_s = -0.00097 R_p^2 -0.01026 R_p +0.83955 (3)

η_V = -0.00076 R_p^2 -0.05080 R_p +1.03231 (4)

CO2 Compressor [21]

η_s = -0.00476 R_p^2 -0.0923 R_p +0.89810 (5)

η_V = -0.00816 R_p^2 -0.15293 R_p +1.13413 (6)

System performances

The overall coefficient of performance, or the first law efficiency, of the cascade refrigeration system is given by,

$$COP = \frac{\dot{Q}_L}{\dot{W}_H + \dot{W}_L} = \frac{(COP_{LTC})(COP_{HTC})}{1 + COP_{LTC} + COP_{HTC}}, \quad (7)$$

where

$$COP_{LTC} = \frac{\dot{Q}_L}{\dot{W}_L} \quad (8)$$

$$COP_{HTC} = \frac{\dot{Q}_M}{\dot{W}_H} \quad (9)$$

The refrigeration capacity $\dot{Q}_L$, the heat transfer rate in the cascade condenser $\dot{Q}_M$, the work in put to the HTC compressor $\dot{W}_H$ and the work input to the LTC compressor $\dot{W}_L$ can all be determined using the relationship given in the table.

IV. Results and discussion

The thermodynamic properties of ammonia and carbon dioxide used here such as specific volume, enthalpy and entropy, are determined using the software EES.

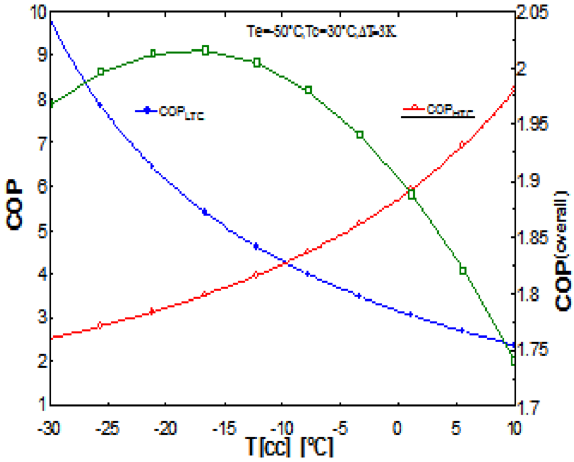

Fig-3 Effect of TCC on the COP of HTC and LTC

Fig.3 plots the curves of COP versus TCC at TC =30°C, TE = -50°C and ΔT= 3K .It gives the effect of TCC on the COP of HTC and LTC, as determined by equations (8) and (9).The COP of HTC increases with TCC where as that the COP of LTC decreases as TCC increases. Hence an optimum TCC and its corresponding maximum COP exist, similar to a two stage system with a single refrigerant. It gives that the COP is maximum 2.01 at the optimum Tcc of -17°C.

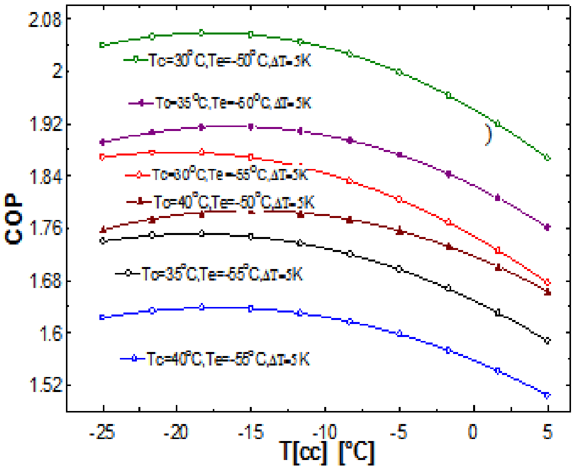

Fig. 4. plots the curves of overall COP versus Tcc at different design parameters

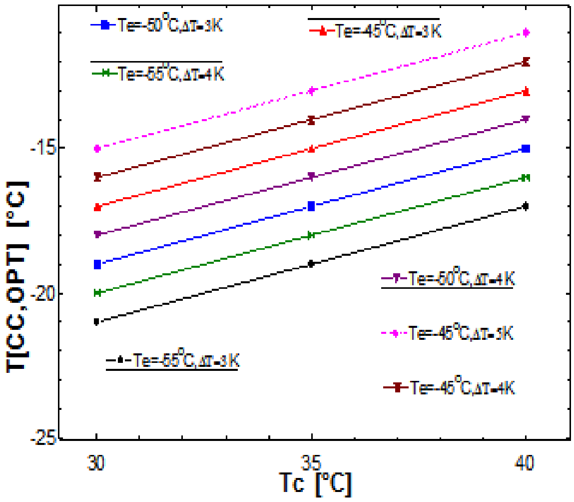

Fig. 5 The influence of Tc on the Tcc,opt of a CO_2/NH_3 cascade refrigeration system.

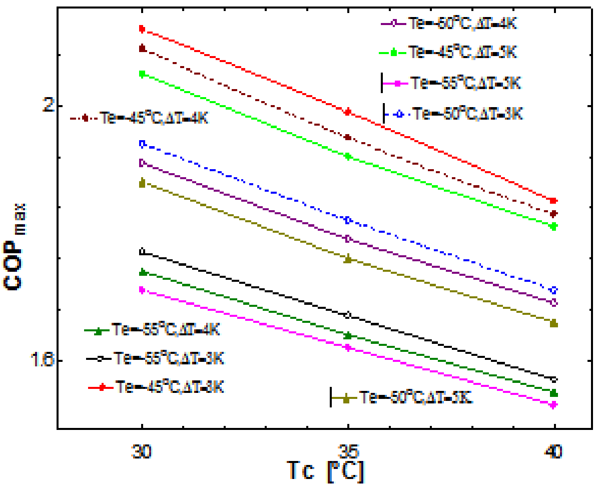

Fig-6 The influence of Tc on the COP_{max} of a CO_2/NH_3 cascade refrigeration system.

Fig - 6 and 7 show the effect of the condensing temperature Tc on the TCC,OPT and the corresponding COPmax at various evaporating temperatures TE and various temperature differences in the cascade condenser ΔT. The figures show that increasing T_C increases TCCOPT and reduces COPmax.

Fig-6 gives that TCCOPT is linearly related to the parameters of TC,TE, and ΔT. Fig-7 plots the same linear relationships between COP_{max} and the parameters of TC, TE, and ΔT. The following regression equations are obtained from the above data.

$$T_{CC,OPT} = 41.48 + 0.4T_C + 0.4T_E + 0.78\Delta T \quad (10)$$

$$COP_{MAX} = 2.083 - 0.0231T_C + 0.0312T_E - 0.03167\Delta T \quad (11)$$

The unit used in Equations (10) and (11) is Kelvin (K).

V. Conclusions

This paper presents the optimum condensing temperature $T_{CC,OPT}$,and the corresponding maximum coefficient of performance $COP_{\max}$ for CO_2/NH_3 cascade refrigeration systems reference to three design parameters – condensing temperature Tc, evaporating temperature T_E, and the temperature difference in the cascade condenser ΔT.

1. The optimum condensing temperature of a cascade condenser increases Tc, T_E and ΔT, where as the maximum COP increases with only T_E, but decreases as Tc or ΔT increases.
2. Two correlations used to determine the optimum condensing temperature of a cascade –condenser and the corresponding maximum COP are obtained with reference to three design parameters.

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