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ABSTRACT

This paper analyzes a CO_2/NH_3 cascade refrigeration system of two temperature range applied in the cold storage. A mathematical model is presented to determine the optimal middle temperatures of the cascade-condenser for obtaining the maximum coefficient of performance (COP) under different operation conditions. Three main parameters including the evaporation temperature in the cold storage, the evaporation temperature in the refrigerated storage and the condensation temperature in the high temperature stage are used to study the optimal middle temperature of CO_2 in the cascade-condenser. The results show that the optimal middle temperature increases with the increment of three main parameters. Moreover, under specific conditions, the optimal temperature is equal to the evaporation temperature of refrigerated storage. The results shown in this paper is helpful to the control strategy of CO_2/NH_3 cascade refrigeration systems for two temperature ranges.

1. INTRODUCTION

In the field of rapid freezing and the storage of frozen food, the required temperature ranges from -30℃ to -55℃ and the ambient temperature is generally from 20℃ to 35℃. A large temperature difference leads to a large pressure difference, which results in the low efficiency of the single- stage refrigeration system. Instead, the cascade refrigeration system can be a better choice and the CO_2/NH_3 cascade refrigeration system is generally applied in the cold storage. Compared with other refrigerants, CO_2 and NH_3 whose global warming potential (GWP) are 1 and 0 respectively are environmental-friendly. Additionally, in this system, NH_3 operates at high temperature stage, which is away from refrigeration storage and NH3 charge is low. Due to non-toxic, non-flammable, CO_2 is safe enough and operates at low temperature stage, which can be closed to the foods. Hence, this system has good reliability and safety.

Many studies have been conducted on the CO_2/NH_3 cascade refrigeration system. Getu and Bansal (2008) presented a thermodynamic analysis on the CO_2/NH_3 cascade refrigeration system to determine the operation parameters and design. Lee *et al.* (2006) thermodynamically analyzed a CO_2/NH_3 cascade refrigeration system to determine the optimal condensing temperature of the cascade-condenser under different design parameters and displayed an equation about the maximum COP. Wang *et al.* (2009) evaluated the performance of a CO_2/NH_3 cascade refrigeration system experimentally and analyzed the effects of different parameters, which showed that the experimental values were close to those predicted by Lee's expression. Messineo (2012) showed a comparison between the CO_2/NH_3 cascade refrigeration system and a hydrofluorocarbon two stage system, which showed that while energy, security and environmental reasons are taken into account, the CO_2/NH_3 cascade refrigeration system is an interesting alternative to R404A two-stage refrigeration system for low evaporating temperatures in commercial refrigeration. Aminyavari *et al.* (2014) analyzed the performances of a CO_2/NH_3 cascade refrigeration system from exergetic, economic and environmental points of view and a genetic algorithm method was used to design the parameters of this system.

In order to meet the temperature demands of different foods, the low temperature stage is generally divided into several circuits for different temperature ranges in the practical applications and these relatively complicated system have received little attention. In this paper, the low temperature stage of the CO_2/NH_3 cascade refrigeration system is divided into two circuits, one for refrigerated storage with the evaporation temperature from −5℃ to -15℃ and the other for cold storage with the evaporation temperature from -30℃ to -50℃. A mathematical model is applied to obtain the optimal middle temperatures of the cascade-condenser under different operation conditions which include the evaporation temperature in the cold storage, the evaporation temperature in the refrigerated storage and the condensation temperature at the high temperature stage. A discussion about the effect of different parameters on the COP is presented in detail.

2. CYCLE SYSTEM DESCRIPTION

A schematic diagram of the CO_2/NH_3 cascade refrigeration system which has been considered in the present study is displayed in Figure 1. This CO2/NH3 cascade refrigeration system is constituted by two single stages, high temperature stage using NH_3 as working fluid and low temperature stage using CO_2 as working fluid, connected by a cascade-condenser. The high temperature stage is aimed at cooling CO_2. The low temperature stage is used to cooling different foods.

It can be clearly seen from Figure 1 that the low temperature stage is constituted by two circuits, one for refrigerated storage (dotted line) and the other for cold storage (chain dotted line). CO_2 at the outlet of cascade-condenser is divided in to two fluids. Fluid A passes though the throttling valve A and changes into two-phase fluid which flows into separator A. The liquid of fluid A in the separator is driven by the pump and flows into evaporator A where the liquid of fluid A absorbs heat for cooling the refrigerated storage. The gas of fluid A is compressed by the compressor A and then back to the cascade-condenser where the gas of fluid A releases heat. The evaporation temperature in the refrigerated storage is assumed to be between -5℃ and -15℃. The cooling capacity of refrigerated storage is assumed as 100kW. The flow process of fluid B is similar to the flow process of fluid A and the evaporation temperature in the cold storage is assumed to be between -30℃ and -50℃. Besides, the cooling capacity of cold storage is also assumed as 100kW.

The main components of the high temperature stage are condenser, NH_3 compressor, throttling valve and cascade-condenser. The condensation temperature is assumed to be between 25℃ and 40℃ and the temperature difference in the cascade-condenser is assumed as 5℃.

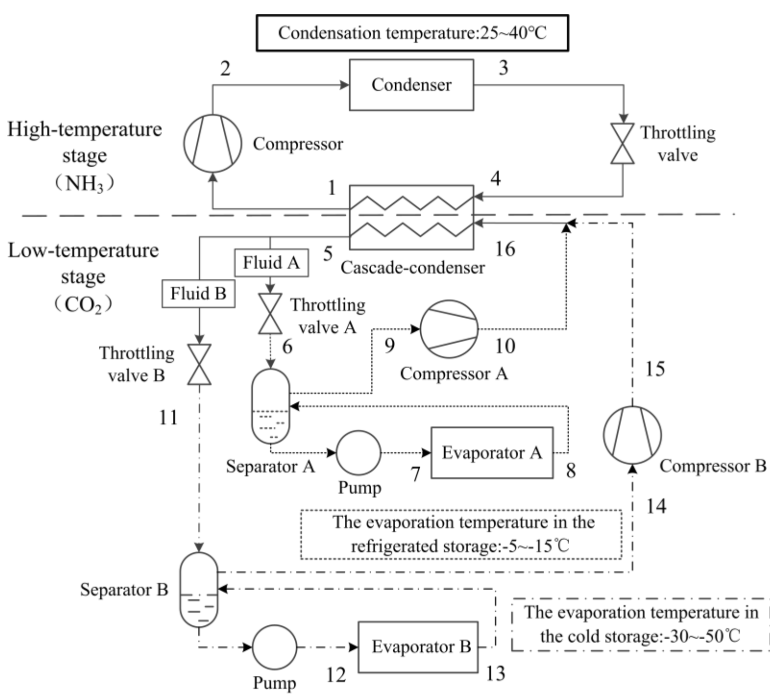

Figure 1: Schematic diagram of the CO_2/NH_3 cascade refrigeration systems for two temperature ranges

3. CYCLE MATHEMATICAL MODELING

3.1 Assumptions for Mathematical Model

To obtain the maximum COP under different operation conditions including the evaporation temperature in the cold storage, the evaporation temperature in the refrigerated storage and the condensation temperature in the high temperature stage, a mathematical model based on mass conservation and energy conservation is proposed. Some assumptions for simplifying analysis are considered as follows:

(1) All components of this system operate under steady conditions.
(2) The pressure drop and heat loss of the refrigerant flowing inside the heat exchangers and pipelines are neglected.
(3) All throttling processes are isenthalpic.
(4) The compression process is irreversible and the isentropic efficiencies of the compressors related to the pressure ratio are taken into account (Stoecker, 1998; Petter *et al.*, 2004).
(5) The superheat degree and the subcooling degree are assumed as 0℃.
(6) The power consumption of pump can be neglected.

3.2 Energy Analysis

For the high temperature stage:

$$W_{\mathrm{com,NH_3}} = m_{\mathrm{NH_3}}(h_{2s} - h_1)/\eta_{\mathrm{com,NH_3}} \tag{1}$$

$$h_2 = \frac{h_{2s} - h_1}{\eta_{\mathrm{com,NH_3}}} + h_1 \tag{2}$$

$$Q_{\mathrm{cc,NH_3}} = m_{\mathrm{NH_3}}(h_1 - h_4) \tag{3}$$

For the low temperature stage:

$$Q_{\mathrm{cc,CO_2}} = (m_{\mathrm{A}} + m_{\mathrm{B}}) * (h_{16} - h_5) \tag{4}$$

$$W_{\mathrm{comA}} = m_{\mathrm{A}}(h_{10s} - h_9)/\eta_{\mathrm{com,CO_2}} \tag{5}$$

$$W_{\mathrm{comB}} = m_{\mathrm{B}}(h_{15s} - h_{14})/\eta_{\mathrm{com,CO_2}} \tag{6}$$

$$Q_{\mathrm{eA}} = m_{\mathrm{A}}(h_6 - h_9) \tag{7}$$

$$Q_{\mathrm{eB}} = m_{\mathrm{B}}(h_{11} - h_{14}) \tag{8}$$

In this system, the COP can be obtained as follows:

$$\mathrm{COP} = (Q_{\mathrm{eA}} + Q_{\mathrm{eB}})/(W_{\mathrm{com,NH_3}} + W_{\mathrm{comA}} + W_{\mathrm{comB}}) \tag{9}$$

The isentropic efficiencies of the compressors are strongly related with the pressure ratio, which are shown as follows (Stoecker, 1998; Petter *et al.*, 2004):

$$\eta_{\mathrm{com,NH_3}} = -0.00097\left(\frac{p_2}{p_1}\right)^2 - 0.01026\left(\frac{p_2}{p_1}\right) + 0.83955 \tag{10}$$

$$\eta_{\mathrm{com,CO_2}} = 0.00476\left(\frac{p_2}{p_1}\right)^2 - 0.09238\left(\frac{p_2}{p_1}\right) + 0.89810 \tag{11}$$

Additionally, the thermodynamic properties of CO_2 and NH_3 are calculated by using REFPROP and the calculation procedure is written by using MATLAB.

4. RESULTS AND DISCUSSION

4.1 Effect of the Condensation Temperature in the High Temperature Stage

Figure 2 displays the results of COP versus middle temperature of CO_2 in the cascade-condenser at the six different condensation temperatures of high temperature stage and the optimal middle temperatures of six curves are marked by arrows. The evaporation temperature in the cold storage and the evaporation temperature in the refrigerated storage are -30℃ and -10℃, respectively. the condensation temperature is from 25℃ to 40℃. It is clearly seen that the condensation temperature has a significant effect on the COP which decreases dramatically with the condensation temperature. The major reason is that an increment of temperature difference in the high temperature stage leads to a pressure difference lift. Hence, the power consumption of high temperature stage increases greatly. In addition, with the increase of middle temperature, six curves of COP increase firstly and then drop. The range of great COP is obtained at the corresponding range of middle temperature. Moreover, the optimal middle temperature increases with the increase of condensation temperature, which is helpful to the control of this refrigeration system.

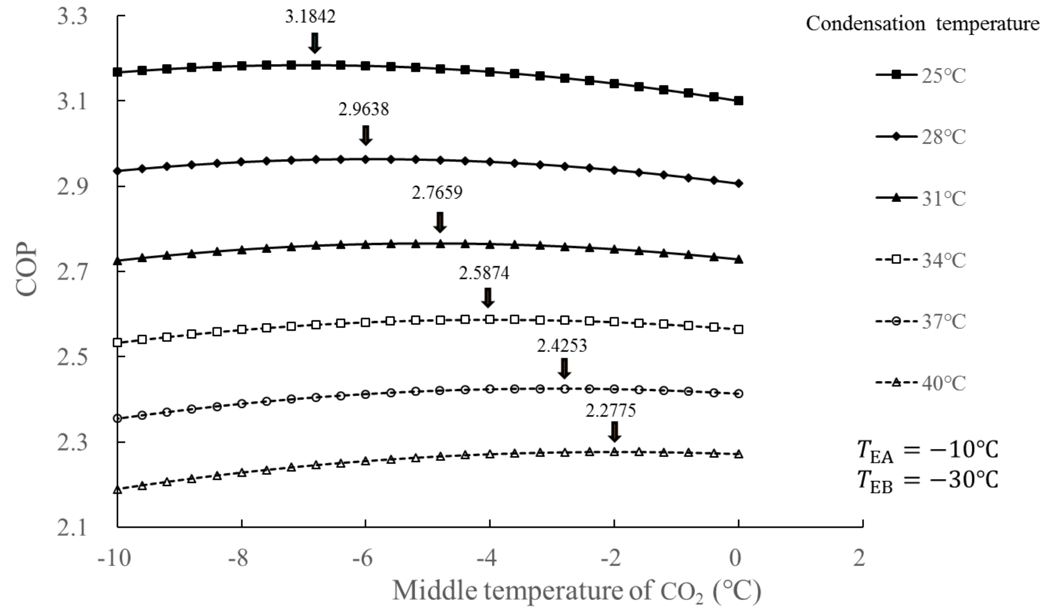

Figure 2: COP versus middle temperature of CO_2 at different condensation temperature

4.2 Effect of the Evaporation Temperature in the Refrigerated Storage

Figure 3 shows the results of COP versus middle temperature of CO_2 in the cascade-condenser at the six different evaporation temperature of refrigerated storage. The optimal middle temperatures of six curves are marked by arrows. The condensation temperature of high temperature stage is 30℃ and the evaporation temperature of cold storage is -30℃. Besides, the evaporation temperature of refrigerated storage is from -5℃ to -15℃ and the middle temperature should be equal or larger than the evaporation temperature of the refrigerated storage. It is clearly seen that an increment of evaporation temperature in the refrigerated storage results in a COP lift. And the optimal middle temperature increases with the evaporation temperature of the refrigerated storage. In addition, with the increase of middle temperature, six curves of COP increase firstly and then drop. The range of great COP is obtained at the corresponding range of middle temperature.

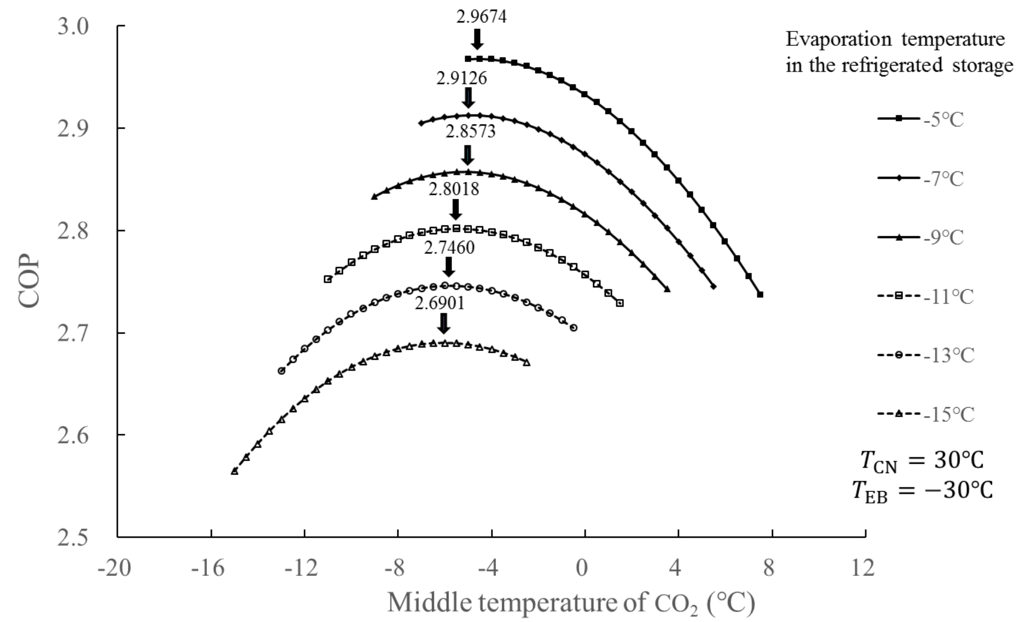

Figure 3: COP versus middle temperature of CO_2 at different evaporation temperature of refrigerated storage

4.3 Effect of the Evaporation Temperature in the Cold Storage

Fig. 4 displays the variations of COP with middle temperature of CO2 in the cascade-condenser at the six different evaporation temperature of cold storage and the optimal middle temperatures of six curves are marked by arrows and dots. The condensation temperature and the evaporation temperature of refrigerated storage are 30℃ and -10℃, respectively. The evaporation temperature of cold storage is from -30 ℃ to -50℃, which is important to the COP. With the increase of evaporation temperature in the cold storage, the pressure difference of compressor B decreases and the power consumption of low temperature stage drops, which leads to the increase of COP. Middle temperature has a relatively slight effect on COP and the fluctuations of six curves are small. When the evaporation temperature of cold storage is from -30℃ to -42℃, the curves of COP increase firstly and then decrease. However, when the evaporation temperature of cold storage is from -46℃ to -50℃, the optimal middle temperature for COP is equal to the evaporation temperature of refrigerated storage. The reason is that the middle temperature should be equal or larger than the evaporation temperature of the refrigerated storage. Besides, the optimal middle temperature grows with the growth of evaporation temperature in the cold storage.

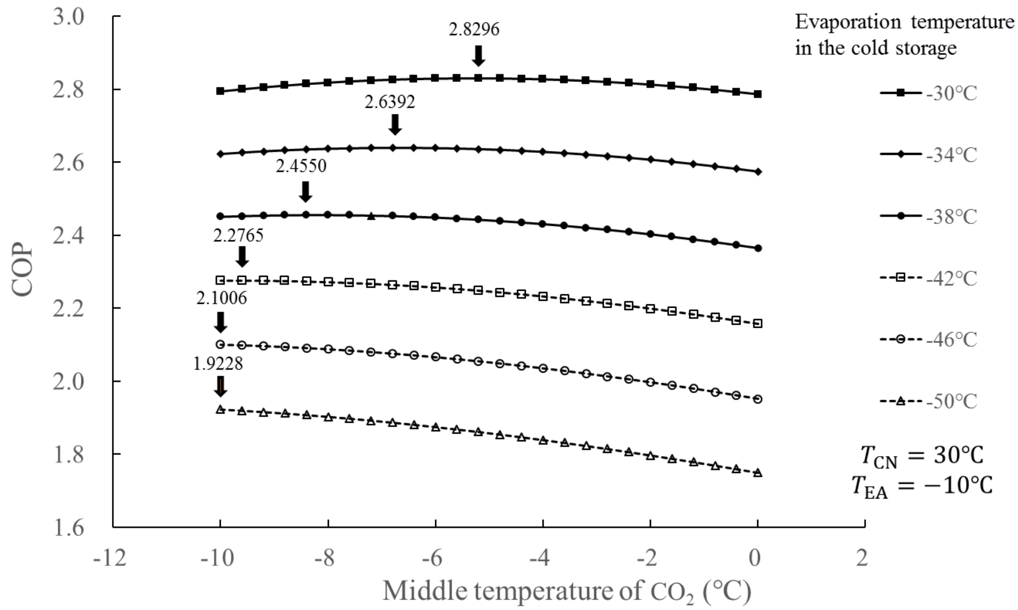

Figure 4: COP versus middle temperature of CO_2 at different evaporation temperature of cold storage

5. CONCLUSIONS

This paper analyzes the performance of a CO_2/NH_3 cascade refrigeration system with two temperature ranges. Three main parameters including the evaporation temperature in the cold storage, the evaporation temperature in the refrigerated storage and the condensation temperature in the high temperature stage are used to study the optimal middle temperature of CO_2 in the cascade-condenser. The conclusions from analytical results for obtaining the optimal performance are shown in detail as follows:

- COP increases with the decrease of condensation temperature in the high temperature stage and the optimal middle temperature increases with the increase of condensation temperature
- Both COP and optimal middle temperature increase with the increment of evaporation temperature in the refrigerated storage.
- Both COP and optimal middle temperature increase with the increment of evaporation temperature in the cold storage.

NOMENCLATURE

W	work or power	(kW)

m	mass flow rate	($kg{\cdot}s^{-1}$)
h	specific enthalpy	($kJ{\cdot}kg^{-1}$)
η	efficiency	(–)
Q	heating capacity	(kW)
COP	coefficient of performance	(–)
p	pressure	(MPa)

Subscript

com	compressor
1,2,3···	points shown in Figure 1
s	isentropic
cc	cascade-condenser
comA	compressor A
comB	compressor B
A	Fluid A
B	Fluid B
eA	evaporator A
eB	evaporator B

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