2002

Simulation of NH3/CO2 Two-Stage Low Temperature Refrigeration System

H. Liu
Xi'an Jiaotong University

Z. Gu
Xi'an Jiaotong University

Y. Li
Xi'an Jiaotong University

Liu, H.; Gu, Z.; and Li, Y., "Simulation of NH3/CO2 Two-Stage Low Temperature Refrigeration System" (2002). *International Refrigeration and Air Conditioning Conference.* Paper 620.
http://docs.lib.purdue.edu/iracc/620

R18-5

SIMULATION OF NH_3/CO_2 TWO- STAGE LOW TEMPERATURE REFRIGERATION SYSTEM

Hongjuan Liu, M.S. student, Environment and Chemical Engineering School
Xi'an Jiaotong University, Xi'an 710049, P. R. China

*Zhaolin Gu, PhD, Professor, Environment and Chemical Engineering School
Xi'an Jiaotong University, Xi'an 710049, P. R. China; Tel: 86-29-3268980; Fax: 86-29-3237910;
E-mail: guzhaoln@mail.xjtu.edu.cn

Yun Li, PhD, Associate Professor, Environment and Chemical Engineering School
Xi'an Jiaotong University, Xi'an 710049, P. R. China

ABSTRACT

Studied is a kind of NH_3/CO_2 two-stage low temperature refrigeration system, in which NH_3 is used as the refrigerant in the high-temperature stage and CO_2 is used as the refrigerant in the low-temperature stage. The performance of the system is simulated and analyzed by changing the mean temperature of the evaporator-condenser and the temperature of the condenser. The mean temperatures of the evaporator-condenser are calculated and demonstrated to maximize the COP of the system at different condensing temperatures. It is shown that there exists an optimum mean temperature of evaporator-condenser to maximize the COP of the system and the maximal COP of the system is increased lowering the condensing temperature. The relationship between the optimum mean temperature and the condensing temperature is given in this paper.

INTRODUCTION

Due to the depletion of the ozonosphere and/or the global warming impact of CFCs, HCFCs and HFCs, natural refrigerants, such as carbon dioxide, ammonia and HCs, are discussed and used for refrigeration systems [1-9].

Carbon dioxide has many advantages as a natural refrigerant. It is non-flammable, inexpensive and abundant. It has great evaporating latent heat and refrigeration capacity per unit. Its kinetic viscidity is low and it's harmony with common lubricant. Carbon dioxide as a single refrigerant have been studied for automobile air-condition, heat pump and so on, mainly focusing on the transcritical refrigeration cycle [1-3]. The defect of carbon dioxide as a refrigerant is the high system pressure for air-conditioning, more than 100 bar.

Ammonia as a refrigerant has advantageous thermodynamic properties. But it is incompatible with ordinary mineral lubricant, so the evaporator must adopt full-liquid evaporator, which increases the filling capacity of ammonia. If there exists some water vapor in the system, ammonia becomes corrosive to copper, so the heat transfer rubes must use steel tube. The capability of heat transfer of steel is not as good as that of the copper, which increases the heat transfer area and the weight of the system. The disadvantages of ammonia as a refrigerant must be overcome in order to expand its market of ammonia refrigeration system [5-7].

Decreasing the filling capacity of ammonia refrigeration system and departing ammonia refrigeration system from the public or half-public area to reduce the fatalness of ammonia refrigeration system, have pressed the technique of ammonia refrigeration to be changed, one of which is using NH_3/CO_2 double working fluids two-stage refrigeration system instead of the traditional NH_3 low temperature refrigeration system. NH_3/CO_2 two-stage low-temperature refrigeration system takes advantages of the environmental qualities of ammonia and its safety issues [7-9]. This refrigeration system minimizes the charge quantity for NH_3 by filling only in the high-temperature stage, using CO_2 as a boiling secondary refrigerant/coolant in the low-temperature stage and possible to keep NH_3 away from safety requiring areas.

With the thermodynamic simulation, the performance of NH_3/CO_2 two-stage refrigeration system is calculated and analyzed by changing the mean temperature of evaporator-condenser between the two stage system and the temperature of the condenser, which provides a theoretic basis for the optimal designation and practical operation of cascading refrigeration system.

NH_3/CO_2 CASCADE REFRIGERATION SYSTEM AND SIMULATION

Figure 1 shows the schematic NH_3/CO_2 two-sage low temperature refrigeration system, including the high temperature loop with ammonia as the working fluid and the low temperature loop with carbon dioxide as the working fluid. The mean heat exchanger (Evaporator-Condenser) is an important heat exchanger, which operating parameters influence the COP of the system. Carbon dioxide as the refrigerant or coolant in the low temperature loop works as a common cascade refrigeration system. The filling capacity of ammonia in the high temperature loop is also decreased compared to ammonia low temperature refrigeration system and the problem of safety will be solved departing NH_3 loop from the public area. In addition, the volumetric refrigeration capacity of carbon dioxide is 8 times that of ammonia, which decreased the volumetric flow of coolant greatly. Because of the great phase-change latent of carbon dioxide, heat transfer is improved and the effective area of heat exchanger is reduced.

Figure 2 shows the *T-S* diagram of NH_3/CO_2 low temperature refrigeration system. The power input consists of two parts, the power of CO_2 compressor in the low temperature loop (N_1) and the power of NH_3 compressor in the high temperature loop (N_2). In the mean heat exchanger (Evaporator-Condenser), α kg NH_3 is needed to cool and condense 1 kg CO_2 assuming no heat loss and heat transfer only by the phase change of CO_2.

$$\alpha = q_c / q_g \tag{1}$$

Where, q_c is the CO_2 refrigeration capacity per unit mass, kJ/kg, q_g is the NH_3 refrigeration capacity per unit mass, kJ/kg.

The mass flow of CO_2 in the low temperature loop for the capacity of the system, Q_0 kW

$$G_d = Q_0 / q_d = Q_0 / (h_{1R}-h_5) \tag{2}$$

Therefore, the mass flow of ammonia in the high temperature loop is

$$G_g = \alpha\, G_d \qquad (3)$$

The coefficient of performance (COP) of this system is

$$\varepsilon = Q_0 / [G_d(\, h_{2R} - h_{1R}\,) + G_g(\, h_{7R} - h_{6R}\,)] \qquad (4)$$

The evaporating temperature of high temperature loop and the condensing temperature of low temperature loop must match to keep the good COP of the system. Therefore, the mean temperature of evaporator-condenser should be designed carefully. Generally speaking, there are two methods to choose the mean temperature of evaporator-condenser. One aims at maximizing the COP of the system, and the other aims at equalizing the pressure ratio of two compressors to save power input. The optimal mean temperature is determined to maximize the COP of the system in this paper.

Assuming pressure losses and isentropic efficiencies of two compressors in the system and given the temperature difference for heat transfer of heat exchanger (Condenser and Evaporator), the evaporating temperature of the low temperature loop and the condensing temperature of the high temperature loop, the performance of the system can be simulated using Eq.(1)-Eq.(4) and the state equations of CO_2 and NH_3.

RESULTS AND ANALYSIS

Calculation conditions: Evaporating temperature of low temperature loop, 233K; Condensing temperature, 308K; Refrigeration capacity, 1.5KW. The temperature difference for heat transfer of heat exchanger is given 5K. The isentropic efficiency of compressors and the mechanical efficiency of compressor shaft are given, 0.75 and 0.98, respectively.

Figure 3 shows the COP of the system changing of the mean temperature of the evaporator-condenser in the range of 250.65K ~265.65K. The mean temperature of the evaporator-condenser influences the COP and there is an optimum mean temperature to get the maximum COP. In Figure 3 the COP gets the maximum value, 1.486 at the mean temperature, 257.65K. However, the COP changes slightly within ±5K of the optimal mean temperature, which provides the flexibility for the design and operation of the NH_3/CO_2 low temperature refrigeration system.

After determining the optimal mean temperature, the state points of the cycle and heat capacities of heat exchangers can be calculated. Table 1and Table 2 demonstrate the results of every state point at the optical mean temperature, the heat load of heat exchangers and the power input of two compressors.

Changing the condensing temperatures of the high temperature loop, the COP is shown in Figure 4. The COP varies with the condensing temperatures. For example, when the condensing temperature is 298.15K, the COP reaches 1.75 at the condensing temperature 298.15K while the COP is only 1.19 at the condensing temperature, 323.15K.

Figure 5 shows the relation of the optimal mean temperatures of the evaporator-condenser with different condensing temperature. The higher the condensing temperature, the higher the optimal mean temperature is. The optimal mean temperature changes from the range of 255.65K to 260.65K while the condensing temperature changes from the range of the 298.15K to 323.15K. This curve is nearly linear and can be formulated as

$$T_M = 196.02 + 0.2 \times T_K \qquad (5)$$

Where, T_K changing from 298.15 to 323.15K

CONCLUSIONS

NH_3/CO_2 two-stage low-temperature refrigeration system takes advantages of the environmental qualities of ammonia and its safety issues. This refrigeration system minimizes the charge quantity for NH_3 by filling only in the high-temperature stage, using CO_2 as a boiling secondary refrigerant / coolant in the low-temperature stage and possible to keep NH_3 away from safety requiring areas. With the thermodynamic simulation, the performance of NH_3/CO_2 two-stage refrigeration system is calculated and analyzed by changing the mean temperature of evaporator-condenser between the two stage system and the temperature of the condenser, which provides a theoretic basis for the optimal designation and practical operation of cascading refrigeration system.

The mean temperature of the evaporator-condenser influences the COP and there is an optimum mean temperature to get the maximum COP. However, the COP changes slightly within ±5K of the optimal mean temperature, which provides the flexibility for the design and operation of the NH_3/CO_2 low temperature refrigeration system.

The COP varies with the condensing temperatures. For example, when the condensing temperature is 298.15K, the COP reaches 1.75 at the condensing temperature 298.15K while the COP is only 1.19 at the condensing temperature, 323.15K.

REFERENCES

[1] Lorentzen G, Pettersen J, A new, efficient and environmentally benign system for car air conditioning, Int J Refrig. 1993,16(1):4~12.

[2] Lorenten G, Revival of carbon dioxide as a refrigerant, Int J Refrig., 1994,17(4):292~301.

[3] Lin Gaopin, Gu Zhaolin, Performance of CO2 transcritical cycle, Journal of Xi'an Jiaotong University, 1998,32 (8): 35-38

[4] Klocker K, Schmidt E L, Steimle F, Carbon dioxide as a working fluid in a drying heat pumps, Int. J. Refrig., 2001, 24(1): 15-24

[5] Paul De Larminat, Expanding the use of ammonia, ASHRAE Journal, 2000, No. 3: 35-40

[6] Forbes S, Pearson, Ammonia refrigeration systems [J], ASHRAE Journal, 1999, No. 3: 24-29

[7] Miyasaki A, Ammonia, carbon dioxide and ice slurry--the commercial refrigerants of the future, Refrigeration of JSRAE, 1999,74(2): 120~125

[8] Enkemann T, Arnemann M, Investigation of CO_2 as a secondary refrigerant [A], Proc. Int. Conf. IIR, New Applications of Natural Working Fluids in Refrigeration and Air Conditioning [C], Hannover, Germany, 1994

[9] Infante Ferreira C A, Boukens R A, Carbon dioxide-secondary coolant or refrigerant for cascade systems? , Proc. Int. Conf. IIR, Applications for Natural Refrigerants, Aarhus, Denmark, 1996

Table 1 Results of the state points of the refrigeration system

Point	T/K	P/MPa	H/kJ kg^{-1}	S/kJ$(kg.\cdot k)^{-1}$
1	233.15	1.004	629.66	5.0288
1_R	232.89	0.984	629.66	5.0322
2_{AR}	280.86	2.474	668.39	5.0322
2	-------	2.426	681.29	--------
3	260.15	2.426	-------	--------
4	260.15	2.426	364.70	3.8718
5	-------	-------	364.70	--------
6	255.15	0.208	1639.86	8.8717
6_R	254.96	0.204	1639.86	8.8811
7_{AR}	395.82	1.375	1925.97	8.8811
7_R	-------	1.375	2021.34	--------
7	-------	1.3488	2021.34	--------
8	308.15	1.348	--------	--------
9	308.15	-------	565.16	4.5638
10	--------	-------	565.16	--------

Table 2 Loads of heat exchangers and compressors (COP=1.486)

Load	Q_e/KW	Q_c/kw	q/kg s^{-1}	N_e/KW
Low-temperature loop	1.50	1.792	5.66	0.318
High-temperature loop	1.792	2.428	1.67	0.692

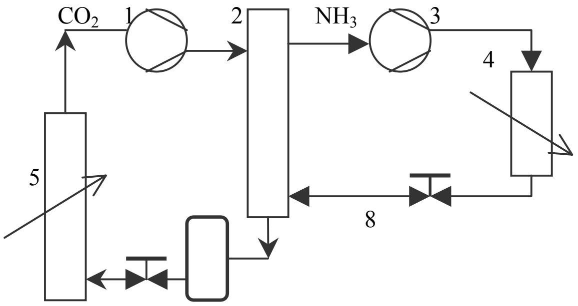

Figure 1 Schematic NH_3/CO_2 two-sage refrigeration system

1-CO_2 compressor; 2-Evaporator –Condenser; 3-NH_3 compressor; 4-Condenser; 5-Evaporator; 6 and 8-Throttle valve; 7-Accumlator

6 7

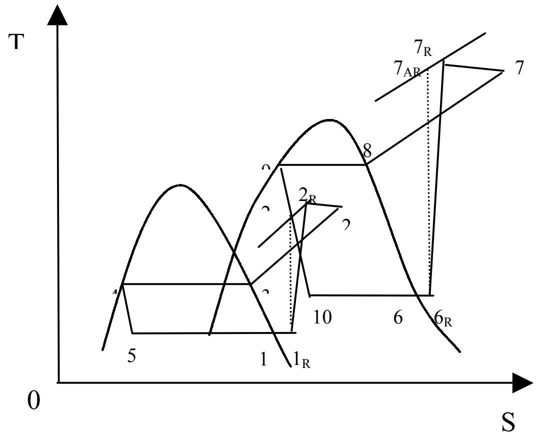

Figure 2 *T-S* diagram of the cycle

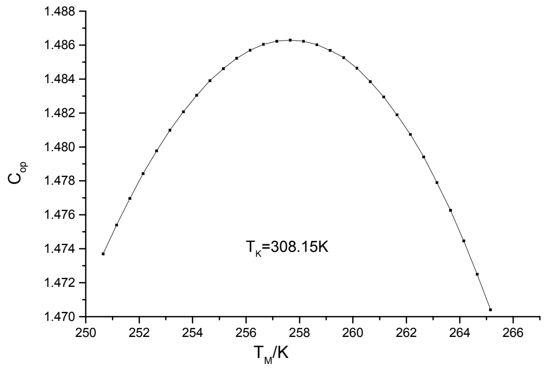

Figure 3 COP at different mean temperatures

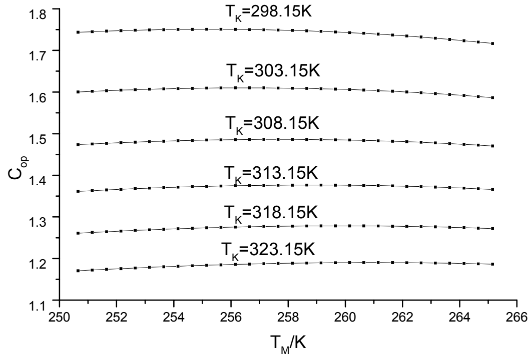

Figure 4 COP at different condensing temperatures

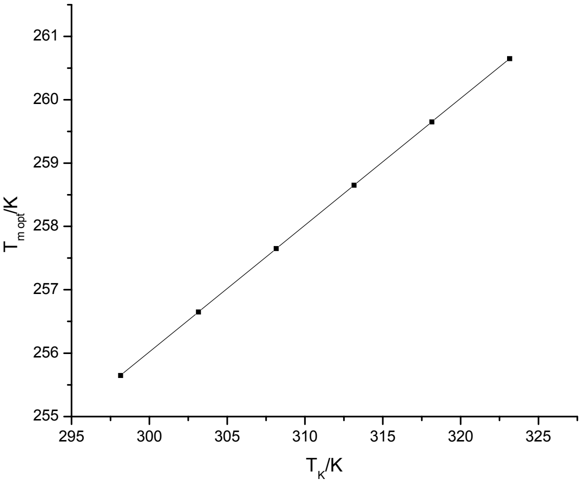

Figure 5 Relation of the optimal mean temperature and the condensing temperature