

Improvement of energy efficiency by cascade system with CO₂ refrigerant

Abstract

Global warming is commonly issue in worldwide. From point of global warming point of view, alternate HFC refrigerant to natural refrigerant is one of the means to reduce global warming gas which is used to refrigeration system in food store such as supermarket and convenience store. Carbon dioxide (hereinafter CO₂) is one of the dominant candidates as low Global Warming Potential (hereinafter GWP) refrigerant. However CO₂ refrigerant has lower efficiency for refrigeration because of low critical temperature. Also CO₂ refrigerant has higher system pressure than HFC refrigerant. Therefore typical cascade refrigeration system is equipped HFC refrigerant in higher temperature cycle. This report indicate the potential of CO₂-CO₂ cascade system even CO₂ refrigerant itself has lower efficiency than HFC refrigerant. And this report shows optimization of discharge pressure, suction pressure and internal heat exchanger are important to get better performance.

Keywords: CO₂, carbon dioxide, coefficient of performance, cascade

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Introduction

The showcase that is installed into most of supermarkets and convenience stores are operated with refrigeration system which is equipped HCFC or HFC refrigerant at this moment. However this HCFC and HFC refrigerant has high GWP. Therefore alternate refrigerant from HCFC and HFC to Low-GWP refrigerant is required. Some configurations of refrigeration system which uses CO₂ refrigerant are studied as solution for reducing global warming gases.¹ As well known, CO₂ refrigerant has lower efficiency at high ambient temperature. The refrigeration system is operated at super critical condition when ambient temperature exceeds 30.98°C. Therefore controlling discharge pressure is important to get better Coefficient of Performance (hereinafter COP). This report shows optimized pressure setting and configuration of internal heat

exchanger in CO₂-CO₂ cascade refrigeration system. And this report shows possibility to use CO₂ refrigerant in high ambient temperature condition in CO₂-CO₂ cascade refrigeration system. HFC-CO₂ cascade system and 2 stage refrigeration system with CO₂ refrigerant has been studied and these refrigeration system shows better performance than traditional HFC404 A refrigeration system. The other hand, CO₂-CO₂ cascade system has not been studied because of less efficiency of CO₂ refrigerant at high temperature cycle. However this study shows the possibility to apply CO₂ refrigerant to such higher temperature cycle with optimization of operating pressure and some system improvements. Also this study shows comprehensive analysis of CO₂-CO₂ cascade system and traditional HFC404A refrigeration system (Figure 1) by using annual temperature at relative medium temperature area and higher temperature area.

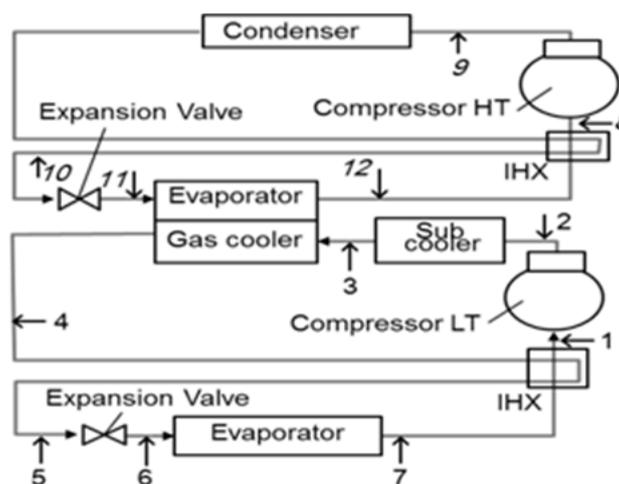


Figure 1 Configuration of CO₂-CO₂ cascade system.

Materials and methods

Configuration of CO₂ cascade system

A cascade system is structured from higher temperature cycle and lower temperature cycle, both cycles being combined thermodynamically by a cascade heat exchanger (Figure 1). Both refrigeration systems have compressor, gas cooler, expansion valve, evaporator, and internal heat exchanger (hereinafter IHX). The lower temperature cycle has additional heat exchanger between compressor and gas cooler as pre-cooler. Exhaust heat from lower temperature cycle is transferred to evaporator in higher temperature cycle. Figure 2 shows p-h diagram of the CO₂-CO₂ cascade system. CO₂-CO₂ cascade system which uses CO₂ refrigerant for both higher temperature cycle and lower temperature cycle has not been studied in the past, because of general acceptance of CO₂ refrigerant's lower efficiency at high ambient temperature condition due to lower critical temperature. However controlling the discharge pressure of CO₂ refrigeration system can make a significant improvement of COP. Both higher and lower temperature CO₂ refrigeration cycle has IHX to improve the efficiency. The effect of IHX is studied in this report.

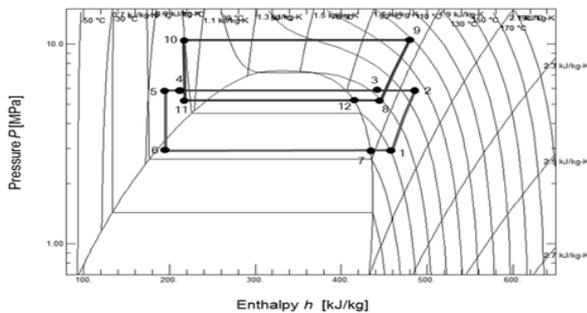


Figure 2 p-h diagram of CO₂-CO₂ cascade system.

Calculation of system efficiency

The condition for calculation and compressor efficiency was quoted from previous paper.¹ The system efficiency of cascade

refrigeration system is shown by formula (1) to (5). The total rate of heat transfer QLT can be expressed by mass flow of refrigerant in lower temperature cycle (MLT) and enthalpy difference between evaporator inlet (Figure 2 point 6, h₆) and outlet (Figure 2 point 7, h₇) of lower temperature cycle. The operation of higher temperature cycle depends on enthalpy difference at cascade evaporator (Figure 2 point 12, h₁₂ and point 11, h₁₁) and heat load at cascade heat exchanger (Figure 2 point 3, h₃ and point 4, h₄) of lower temperature cycle. In formula (2), capacity of higher temperature cycle is equivalent with heat load from lower temperature cycle.

Total COP of the CO₂-CO₂ cascade refrigeration system is shown by formula (5).

$$Q_{LT} = M_{LT} (h_7 - h_6) \tag{1}$$

$$Q_{HT} = M_{HT} \cdot (h_{12} - h_{11}) \tag{2}$$

$$= M_{LT} (h_3 - h_4)$$

$$W_{LT} = M_{LT} (h_2 - h_1) / \eta_{LT} \tag{3}$$

$$W_{HT} = M_{HT} (h_9 - h_8) / \eta_{HT} \tag{4}$$

$$COP = Q_{LT} / (W_{LT} + W_{HT}) \tag{5}$$

The parameters of calculation for CO₂-CO₂ cascade system are considered as Table 1. The discharge pressure at higher temperature cycle (P_{dHT}) is set from 7MPa to 12MPa with 1MPa pitch. The discharge pressure at lower temperature cycle (P_{dLT}) is from 4MPa to 12MPa with 1MPa pitch. The suction pressure at higher temperature cycle is set from 3MPa to 6MPa. T₈ and T₁ are considered as heat transfer at IHX.

Table 1 Test condition for refrigeration system

Ta [°C]	PdHT [MPa]	PsHT [MPa]	T8 [°C]	PdLT [MPa]	PsLT [MPa]	ΔP [MPa]	T1 [°C]	T3 [°C]
10	7.0, 8.0, 9.0, 10.0, 11.0, 12.0	3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	4.0 ~12.0 (1.0 pitch)	3	1.0, 2.0, 3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	10
20	7.0, 8.0, 9.0, 10.0, 11.0, 12.0	3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	4.0 ~12.0 (1.0 pitch)	3	1.0, 2.0, 3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	20
30	7.0, 8.0, 9.0, 10.0, 11.0, 12.0	3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	4.0 ~12.0 (1.0 pitch)	3	1.0, 2.0, 3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	30
40	7.0, 8.0, 9.0, 10.0, 11.0, 12.0	3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	4.0 ~12.0 (1.0 pitch)	3	1.0, 2.0, 3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	40
50	7.0, 8.0, 9.0, 10.0, 11.0, 12.0	3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	4.0 ~12.0 (1.0 pitch)	3	1.0, 2.0, 3.0, 4.0, 5.0, 6.0	-5.6, 0.2, 5.3, 10.0, 14.3, 18.3, 22.0	50

Results and discussion

Effect of internal heat exchanger

The IHX is installed to increase the COP. The configuration of IHX depends on operating condition.² However the configuration of IHX is not changeable. Therefore the configuration should be decided

by concerning annual operating condition. The percentage of ambient temperature range (Table 2) is calculated by environment data at Tokyo/Japan and Delhi/ India area for 8 years that was given by Japan Metrological Agency³ and Climate Zone.com.⁴ Figure 3 shows annual COP of high temperature cycle that is related to heat transfer at IHX. The suction pressure (PsHT) is fixed at 3MPa. The maximum

heat transfer rate (Figure 2 point 7-1 equivalent point 4-5) is set to enthalpy difference between h1 to h7 ($T1=-3.3$ to 22.0°C). In case of Tokyo condition, the heat transfer ratio with 0% gets highest COP. However in case of Delhi condition, the heat transfer ratio =100% gives highest COP. This result says. IHX is not needed for Middle temperature area such as Tokyo and optimized IHX is needed for hot temperature area such as Delhi. Thus optimization of IHX should be considered by region of installation area (Tables 1 & 2).

Table 2 Ratio of days in ambient temperature range

Temperature range [°C]	Ration of days [%]	
	Tokyo/ Japan	Delhi/ India
below 10	25	5.6
10 to 20	38.9	22.2
20 to 30	30.6	41.7
30 to 40	5.6	30.6

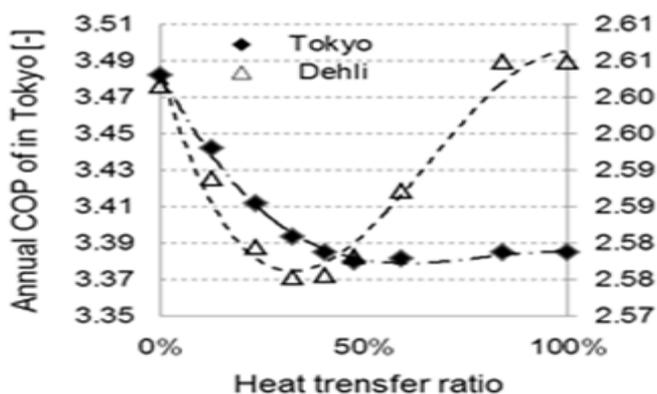


Figure 3 Annual COP of higher temperature cycle vs IHX heat transfer ratio in case of Tokyo and Delhi condition.

Optimization of pressure setting

As a result of calculations, the pressure setting for both refrigeration cycle of high pressure and low pressure that gives highest COP are given in Figure 4. Ambient temperature from 10oC ($T_a=10$) to 40°C ($T_a=40$) the discharge pressure at lower temperature cycle that gives highest COP is increased by ambient temperature. However at ambient temperature 50oC ($T_a=50$), the highest COP is given by discharge pressure (P_{dLT}) at 7MPa. The reason of this change comes from isentropic efficiency of compressor. The isentropic efficiency of this compressor has peak at compression ratio 2.3 (P_d/P_s) and this isentropic efficiency decreased above compression ratio 2.5. Smaller pressure difference between Suction pressure of lower temperature cycle (P_{dLT}) and discharge pressure of higher temperature cycle (P_{dHT}) provides smaller compression ratio of compressor. This smaller compression ratio makes effort to improve volumetric efficiency of the compressor. However highest COP at 30°C ($T_a=30$) and 40°C ($T_a=40$) are given from pressure difference (ΔP) at 3MPa. Because increased discharge pressure at lower temperature cycle makes effort to exhaust heat by pre-cooler. Thus heat load on higher temperature cycle is reduced (Figure 4).

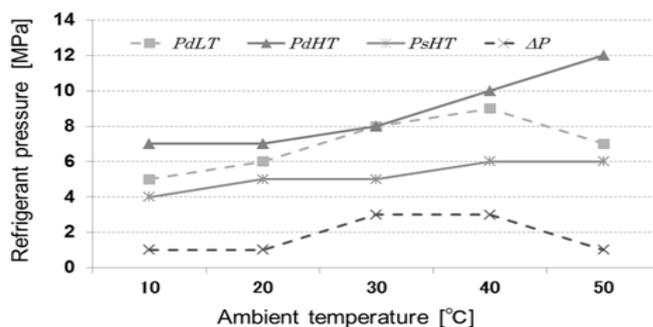


Figure 4 Optimized pressure for cascade system.

Experimental result

The refrigeration system was installed to test facility (Figure 5) and connected to actual refrigeration showcase that was same as typical convenience store in Japan (Figure 6). Figure 7 shows inside of CO₂-CO₂ cascade system which was installed to test facility. The efficiency was measured by input power of compressor and mass flow meter of refrigerant. The test condition is shown in Table 3. The pressure setting was optimized to get best efficiency at each temperature conditions. The efficiencies are compared as efficiency ratio between CO₂-CO₂ cascade system and HFC single refrigeration system (Figure 8). The calculation result shows better efficiency of CO₂-CO₂ cascade system around 15% to 20% and the test result shows around 15% better efficiency of CO₂-CO₂ cascade system from HFC single refrigeration system. And this compared COP insists that there are same result between calculation and experimental results (Figure 9).

Table 3 Test condition for refrigeration system

T_a [°C]	H_a [%RH]	TCVS[°C]	HCVS[%RH]	T_{sc} [°C]
2	45	22	35	4
7	50	22	35	4
20	55	25	45	4
28	60	26	50	4
35	65	27	50	4
40	65	27	50	4



Figure 5 CO₂ Cascade system.

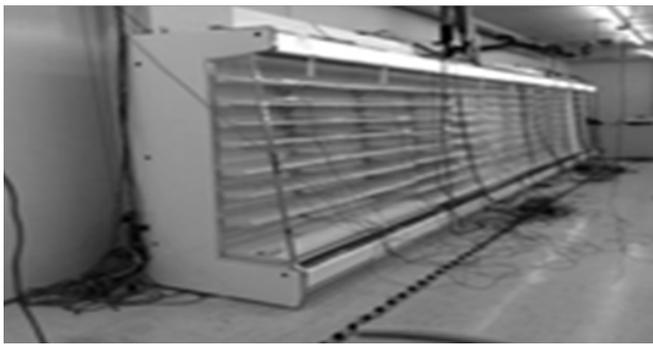


Figure 6 Showcases.

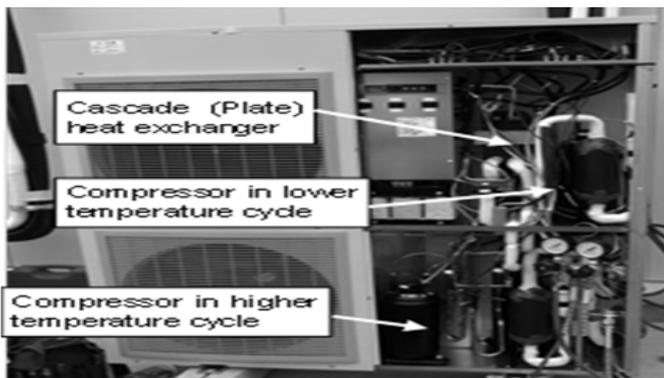


Figure 7 Inside of CO_2 - CO_2 cascade system.

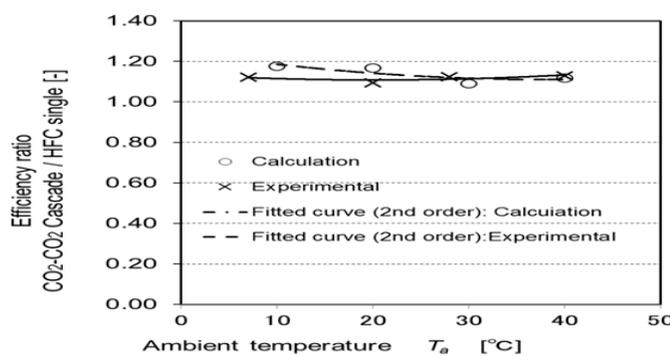


Figure 8 Efficiency ratio between CO_2 - CO_2 refrigeration system and HFC refrigeration system.

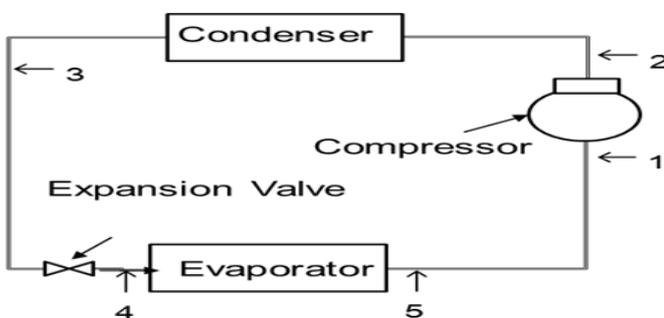


Figure 9 Configuration of single refrigeration system for HFC404A refrigerant.

Effect of CO_2 - CO_2 cascade system

The Figure 10 shows calculated COP that is compared with CO_2 - CO_2 cascade system and typical refrigeration system that is equipped R404A (Figure 10) at several ambient temperatures. CO_2 - CO_2 cascade system has better efficiency from 10% to 20% than HFC404A refrigeration system. The advantage is bigger at lower ambient temperature side. This means annual efficiency of CO_2 - CO_2 cascade system is around 20% better than typical R404A refrigeration system, because average ambient temperature is below 20°C at most of city around the world.² Figures 11 & 12 are calculated COP that were given from approximate expression of COP in Figure 10. The COP is described by second order formula of ambient temperature. Monthly COP is calculated by using average temperature, monthly average highest temperature, and monthly minimum temperature. The times of those temperatures are 8 hours each. And the environment data at Tokyo/Japan and Delhi/India area for 8 years that was given by Japan Metrological Agency³ and Climate Zone.com.⁴ CO_2 - CO_2 cascade system has better COP than HFC404A single refrigeration system throughout the year at Tokyo condition and Delhi condition. In case of Tokyo condition, the advantage of CO_2 - CO_2 cascade system is around 16% and Delhi condition is 13% better than HFC404A single cycle. Even summer condition CO_2 - CO_2 cascade system has better COP. In case of Delhi condition, the average of monthly maximum ambient temperature is nearly 40°C. Although such high ambient temperature CO_2 - CO_2 cascade system has better COP than HFC404A single system around 11%.

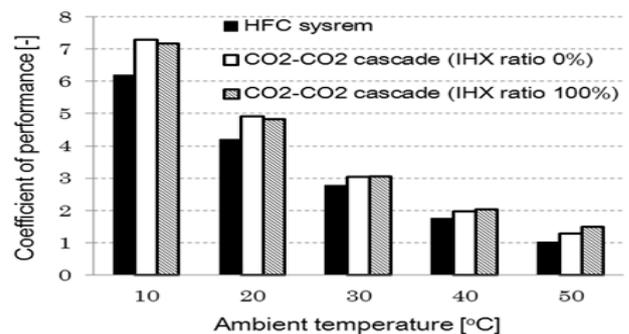


Figure 10 Compared COP with CO_2 - CO_2 cascade system and HFC404A system.

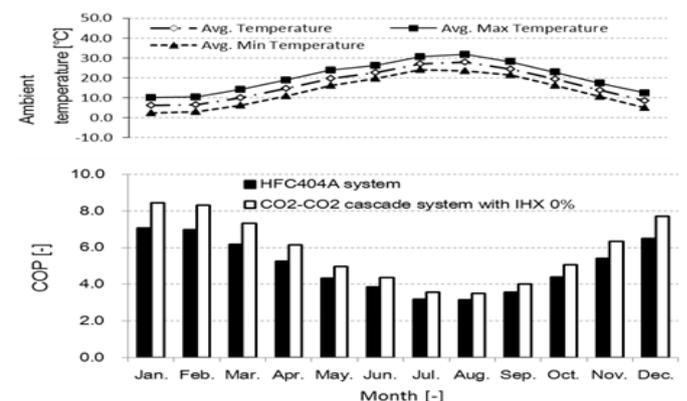


Figure 11 Comparison of calculated Monthly COP at Tokyo condition between CO_2 - CO_2 cascade system and HFC404A system.

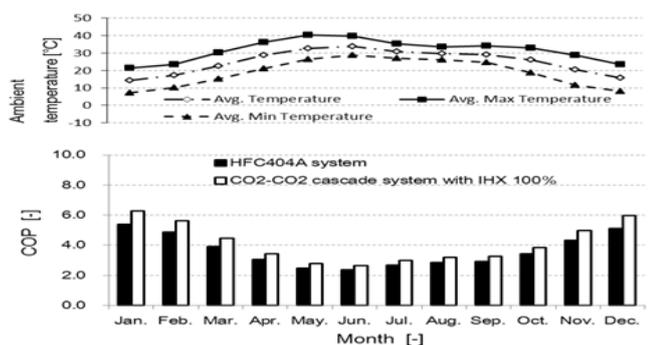


Figure 12 Comparison of calculated Monthly COP at Delhi condition between CO_2 - CO_2 cascade system and HFC404A system.

Conclusion

HFC refrigerant is widely used in refrigeration industry at this moment. The market stock of refrigerant which is charged to refrigeration system is increased because of replacement of CFC and HCFC refrigerant. CO_2 refrigerant is considering as alternative solution to reduce not only global warming gas but also CO_2 emission from energy usage of refrigeration system. Because energy usage of refrigeration system at typical convenience store is around 50% of total energy usage of its store.⁴ Therefore reducing energy usage on refrigeration system makes significant effort to reduce total CO_2 emission from food store such as convenience store. CO_2 refrigerant have been not used to higher temperature cycle of cascade system because of its lower critical temperature.^{5,6} However this report shows the possibility of CO_2 - CO_2 cascade system that has better efficiency than existing HFC refrigeration system for retail industry. In addition optimization of IHX configuration for higher ambient temperature area should be considered to get better efficiency.

Nomenclature

T_a : Ambient temperature [°C]

P_d : Discharge pressure [MPa]

P_s : Suction Pressure [MPa]

h : Enthalpy [kJ/kg]

Q : Rate of heat transfer [W]

W : Power [W]

M : Mass flow rate [kg/hr]

T_n : Refrigerant temperature at point n in T - h diagram [°C]

COP: Coefficient of performance

η : Isentropic efficiency

HT: Higher temperature cycle

LT: Lower temperature cycle

ΔP : Pressure difference between suction on higher cycle and discharge on lower cycle.

Acknowledgments

None.

Conflicts of interest

None.

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