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Performance Characteristics of the R404A Indirect Refrigeration System Using CO_2 as a Secondary Refrigerant

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Abstracts

In this study, to investigate the performance characteristics of the R404A indirect refrigeration system using CO_2 as a secondary refrigerant, the R404A refrigeration system was analyzed experimentally. Under given experimental conditions, the lower the difference between the R404A condensing temperature and the CO_2 cooling temperature was, the greater the coefficient of performance (COP) of the total indirect refrigeration system was, including the R404A refrigeration and CO_2 refrigerant circulation systems. These findings indicate that the COP of the R404A indirect refrigeration system using CO_2 as a secondary refrigerant and the COP of the total refrigeration system are influenced by such variables as the R404A condensation temperature, the efficiency of the R404A internal heat exchanger, and the difference in CO_2 cooling temperature, CO_2 evaporation temperature, and CO_2 mass flow. Therefore, the optimum R404A indirect refrigeration system using CO_2 as a secondary refrigerant could be designed by considering these variables

Keywords: CO_2, R404A, Secondary refrigerant, COP (coefficient of performance), Performance characteristics.

Nomenclature

Symbols

COP	: Coefficient of performance	[-]
G	: Mass flow rate	[kg/min]
h	: Enthalpy	[kJ/kg]
Q	: Heat capacity	[kW]
P	: Pressure, pumping power	[kPa], [kW]
T	: Temperature	[ºC]
W	: Compressor electrical power	[kW]

Greek symbols

Δ	: Difference	[-]
η	: Efficiency	[-]

Subscripts

cooler	: Cooler
c	: Condenser
e	: Evaporator
in	: Inlet
out	: Outlet
R404A	: R404A refrigeration system
CO_2	: CO_2 secondary system
total	: Indirect refrigeration system
sub	: Subcooling degree
sup	: Superheating degree

Introduction

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The indirect refrigeration system is mainly used in large discount stores or refrigeration warehouses, and single-phase brine has been used as the secondary refrigerant for cooling goods. Single-phase brine, however, greatly increases the pump power consumption due to the rising viscosity as the cooling temperature falls. Therefore, research is being conducted to enable CO_2 to be used as the secondary refrigerant because its viscosity does not become large even at low temperatures like -30 ºC~-50 ºC. [1-2]

There are a number of published studies on the indirect refrigeration systems using CO_2 as a secondary refrigerant. Hinde et al. [3] compared the pipe diameter of the indirect refrigeration system using CO_2 as a secondary refrigerant with that of the indirect refrigeration system using the conventional brine. As a result, the use of CO_2 as the secondary refrigerant provided large evaporative latent heat and excellent heat transfer efficiency. Thus, the pipe diameter can be reduced because the CO_2 mass flow is smaller than that in the conventional indirect refrigeration system that uses brine. Accordingly, the pressure drop due to pipe friction also decreased.

Kaga et al. [4] developed a compact R600a indirect refrigeration system that circulates CO_2 as a secondary refrigerant through the thermosiphon effect. Consequently, it was reported that at the same refrigeration capacity, this refrigeration system could reduce energy consumption by 95% over the direct refrigeration system using R134a.

Kruse [5] compared the energy consumptions of the indirect and direct refrigeration systems. He revealed that using CO_2 as a secondary refrigerant of an indirect refrigeration system could reduce the energy consumption by about 40%.

Kawamura et al. [6] compared the indirect refrigeration systems using CO_2 and a representative brine (29.9% $CaCl_2$, 54.1% propylene glycol (PG), 52.8% ethylene glycol (EG)) as a secondary refrigerant at the refrigeration capacity of 120 kW. They found that the use of CO_2 reduced the pump power consumption by 90% over each brine while the CO_2 pipe size was fivefold smaller than that of each brine.

Most of these studies were about the pump power of the CO_2 secondary circulation system and heat transfer, and there is almost no study that analyzed the performance characteristics of the indirect refrigeration system. Furthermore, there is no established theory about this.
Therefore, in this study, to investigate the performance characteristics of the R404A indirect refrigeration system using CO_2 as a secondary refrigerant, the R404A refrigeration system was analyzed under the same experimental conditions. As a result, the basic data for the optimum design of the R404A indirect refrigeration system using CO_2 as a secondary refrigerant would be provided.

Experimental apparatus and method

Experimental apparatus and method

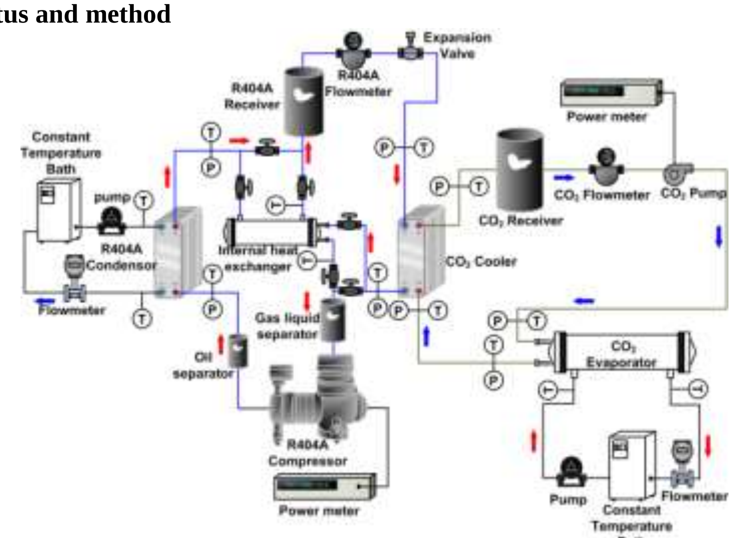

Fig. 1 Schematic diagram of the experimental apparatus.

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Photo 1 Photograph of the experimental apparatus.

This experimental apparatus (Fig. 1) was designed to investigate the performance characteristics of the indirect refrigeration system using CO_2 as a secondary refrigerant. To obtain the data required for the calculation and evaluation of the refrigeration capacity, compressor power consumption, pump power consumption, and coefficient of performance (COP), the refrigeration temperature, pressure, and flow at various points of the indirect refrigeration system as well as the power consumptions of the compressor and pump were measured. The measurement points are shown in Fig. 1. Photo 1 is a photograph of this refrigeration system. As shown in Fig. 1, this indirect refrigeration system largely consists of a one-step compression and one-step expansion vapor compression refrigeration system using R404A as a refrigerant and a system using CO_2 as a secondary refrigerant. The R404A refrigeration system consists of a compressor, an R404A condenser, an internal heat exchanger, a liquid receiver, an expansion valve, a CO_2 cooler (R404A evaporator, CO_2 condenser), and an accumulator. Furthermore, the CO_2 secondary refrigerant system consists of a CO_2 cooler, a liquid receiver, a pump, and a CO_2 evaporator.

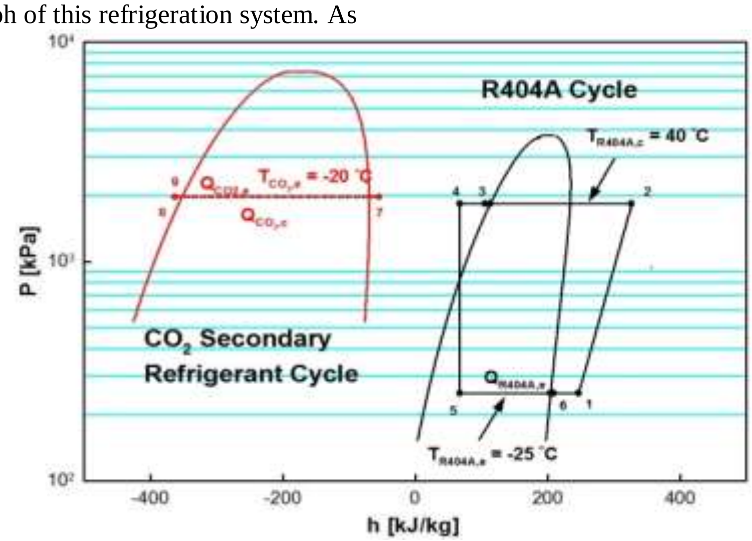

Fig. 2 Pressure-enthalpy diagram of the R404A refrigeration system using CO_2 as a secondary refrigerant.

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Fig. 2 shows various points of this refrigeration system on a P-h diagram. First, the one-step compression and one-step expansion refrigeration system for R404A (right side of Fig. 2) is identical to the basic one-step compression and one-step expansion refrigeration system. Furthermore, the CO_2 secondary refrigerant system (left side of Fig. 2) appears as a straight line on the P-h diagram. This is because due to the very small pump power consumption, they appear to be at the same position on the P-h diagram, but there is actually a small difference.

Table 1 Experimental conditions.

Parameter	Value
Condensing temperature of R404A [℃]	30-50
Subcooling degree of R404A [℃]	5
Superheating degree of R404A [℃]	5
Efficiency of internal heat exchanger	0.3-0.7
Mass flow rate of R404A [kg/min]	2.9-3.5
Temperature difference of CO_2 cooler [℃]	1-9
Subcooling degree of CO_2 [℃]	5
Superheating degree of CO_2 [℃]	5
Evaporation temperature of CO_2 [℃]	-30~-10
Mass flow rate of CO_2 [kg/min]	1-1.5

The coolant circulation process of this indirect refrigeration system can be described in two steps. First, in the R404A circulation process in the one-step compression and one-step expansion vapor compression refrigeration system, the R404A that the liquid receiver was filled in under subcooling conditions is circulated by the compressor. Then, after the refrigerant from the liquid receiver passes through the mass flow meter, the flow rate and density of the refrigerant are measured to check the state of the R404A refrigerant, and the refrigerant is expanded by the expansion valve. After this, the CO_2 cooler exchanges heat with the CO_2, and it flows into the heat exchanger (if there is no bypass). At this moment, the internal heat exchanger exchanges heat with the refrigerant from the condenser, and the superheated steam enters the accumulator and compressor. The high-temperature, high-pressure refrigerant steam compressed in the compressor exchanges heat with the heat source water in the condenser before passing through the internal heat exchanger and entering the liquid receiver. Next, to explain the CO_2 circulation in the CO_2 secondary refrigerant system, the refrigerant from the CO_2 liquid receiver flows into the mass flow meter by pump operation, and the refrigerant from the mass flow meter enters the CO_2 evaporator. In the CO_2 evaporator, the refrigerant exchanges heat with the heat source water (brine) and is sent to the CO_2 cooler and liquefied. Then the refrigerant is subcooled again and returns to the liquid receiver again for CO_2.

The temperatures of the refrigerants and secondary refrigerants in the various heat exchangers used in this system were measured with T thermocouples. The absolute pressure meters were installed at the inlets and outlets of various heat exchangers and evaporators, and the power consumptions of the compressor and pump were measured with a wattmeter. For data collection and system control, a data acquisition system and a computer were used.

Once this system assumes a normal state, the instruments are activated and the temperature, pressure, and mass flow data of the measuring unit are sent to the computer via GPIB communication. When the system reaches the normal state (the system is regarded as having reached the normal state when the temperature measurement variation for 15 min is within ±0.5℃, when the pressure measurement variation is within ±5 kPa, and when the mass flow variation is within ±0.05 kg/min), the refrigerant temperature, pressure mass flow, compressor power consumption, and pump power consumption are measured three times in 5 min intervals. Table 1 shows the operation conditions of the indirect refrigeration system in this study. And, Table 2

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presents the specification for the experimental apparatus.

Table 2 Specification of each component.

R404A compressor			
Model		HGX34P/380-4S	
Max working current (A)		26.1	
Max power consumption (kW)		9.1	
Max permissible pressure (bar)		38	
R404A condenser			
Model		ACH-70X	
Heat capacity (kW)		16.29	
Total heat transfer area (m^2)		3.35	
R404A internal heat changer			
Inner tube	Inner diameter (mm)	11.1	
	Outer diameter (mm)	12.7	
Out tube	Inner diameter (mm)	19.8	
	Outer diameter (mm)	22.2	
Total length (mm)		6000	
R404A expansion valve			
Model		Series L	
Orifice diameter (mm)		3.25	
Maximum allowable pressure (bar)		68.9	
CO_2 cooler			
Model		ACH-70X	
Heat capacity (kW)		10.86	
Total heat transfer area (m^2)		2.45	
Refrigerant pump			
Model		Micropump, Series 5000	
Flow rate (L/min)		0 ~ 13.5	
Maximum allowable pressure (bar)		103	

Data Reduction

The thermal properties of the R404A and CO_2 that were used in this study were calculated using REFPROP (version 8.01) [7], a refrigerant properties calculation program developed by NIST, and the following calculation formulas were used for analysis of the experimental data for identification of the performance of the CO_2 secondary refrigerant system. First, the condensation heat capacity of the R404A refrigeration system was calculated using the following equation:

$$Q_{R404A,c} = G_{R404A} \cdot (h_{R404A,c,in} - h_{R404A,c,out}) \qquad (1)$$

Where G_{R404A} is the mass flow rate [kg/s] of R404A. Furthermore, $h_{R404A,c,in}$ and $h_{R404A,c,out}$ denote the

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refrigerant enthalpies [kJ/kg] at the inlet and outlet of the R404A condenser, respectively.

The evaporation heat capacity (Q_{R404Ae}) of R404A was calculated using the following equation:

$$Q_{R404Ae} = G_{R404A} \cdot (h_{R404Ae,out} - h_{R404Ae,in}) \quad (2)$$

where $h_{R404Ae,in}$ and $h_{R404Ae,out}$ denote the refrigerant enthalpies [kJ/kg] at the inlet and outlet of the R404A evaporator, respectively.

The condensation heat capacity ($Q_{CO_2,c}$) of the CO_2 secondary refrigeration system was calculated using the following equation:

$$Q_{CO_2,c} = G_{CO_2} \cdot (h_{CO_2,c,in} - h_{CO_2,c,out}) \quad (3)$$

where G_{CO_2} is the mass flow rate [kg/s] of CO_2. Furthermore, $h_{CO_2,c,in}$ and $h_{CO_2,c,out}$ denote the refrigerant enthalpies [kJ/kg] at the inlet and outlet of the CO_2 condenser, respectively.

As with the R404A condenser, the heat exchange rate of the CO_2 evaporator was calculated using the following equation:

$$Q_{CO_2,e} = G_{CO_2} \cdot (h_{CO_2,e,out} - h_{CO_2,e,in}) \quad (4)$$

where $h_{CO_2,e,in}$ and $h_{CO_2,e,out}$ denote the refrigerant enthalpies [kJ/kg] at the inlet and outlet of the CO_2 evaporator, respectively.

There are two kinds of COPs for this system: R404A refrigeration system COP (COP_{R404A}) and the total system COP ($COP_{R404A+CO_2}$). The R404A refrigeration system COP is generally identical to that of the one-step compression one-step expansion refrigeration system and was calculated using equation (5). The total system COP was calculated using equation (6) by Lin and Jiang. [8] The reason for using the CO_2 evaporator heat capacity for the numerator value, as in equation (6), is that the CO_2 evaporator heat capacity value in the total system is the heat capacity that is directly used and is the most important parameter in the total system. The reason for using the power consumptions of the compressor and pump is that the work inputted to the total refrigeration system includes the pump as well as the compressor.

$$COP_{R404A} = \left(\frac{Q_{R404Ae}}{W_{com}} \right) \quad (5)$$

$$COP_{R404A+CO_2} = \left(\frac{Q_{CO_2,e}}{W_{com} + P_{pump}} \right) \quad (6)$$

Here, W_{com} is the compressor power consumption [kW] and P_{pump} is the pump power consumption [kW].

This study estimated the uncertainties of the experiment results. The forecast was calculated using the equation proposed by Kline and McClintock [9], and the estimated results are shown in Table 3.

Table 3 Parameters and estimated uncertainties.

Parameters	Uncertainty
Temperature [°C]	± 0.2
CO_2 cooler temperature difference [°C]	± 0.4
Heat flux [kW/m^2]	± 0.0905
Pressure [Pa]	± 0.743
Mass flow rate [kg/min]	± 0.01
Compressor power consumption [kW]	± 0.035
Pump power consumption [W]	± 0.235
COP of R404A refrigeration system	± 0.0128
COP of indirect refrigeration system	± 0.0135

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Results and discussion

This section analyzes the performance characteristics of the indirect refrigeration system using CO_2 as a secondary refrigerant so as to provide the basic design data. Hence, the COP was examined according to the changes in the R404A condensation temperature, the efficiency of the R404A internal heat exchanger, and the difference in CO_2 cooler temperature, CO_2 evaporation temperature, and CO_2 mass flow rate.

Effects of the R404A Condensation Temperature

Fig. 3 shows the COP of the R404A refrigeration system (COP_{R404A}), the COP of the indirect refrigeration system ($COP_{R404A+CO_2}$), R404A evaporation heat capacity ($Q_{R404A,e}$), compressor power consumption (W_{com}), and pump power consumption (P_{pump}) according to the changes in the R404A condensation temperature under the same conditions (CO_2 evaporation temperature ($T_{CO_2,e}$=-20℃), CO_2 cooler temperature difference ($\Delta T_{CO_2,cooler}$=5℃), internal heat exchanger efficiency (η_{IHX}=0.58), subcooling degrees ($T_{R404A,sub}$, $T_{CO_2,sub}$=5℃), and superheating degrees ($T_{R404A,sup}$, $T_{CO_2,sup}$=5℃) of the R404A refrigeration system and CO_2 secondary refrigeration system). As can be seen in Fig. 3, as the R404A condensation temperature increased by 5℃, and the COP of the R404A refrigeration system and the COP of the indirect refrigeration system decreased by about 9.5 and 9.3%, respectively. This is because each time the R404A condensation temperature increases by 5℃, the R404A evaporation heat capacity decreases by about 3.4% and the compressor power consumption increases by about 9.9%. The reason that the R404A evaporation heat capacity decreases here is that the condenser outlet enthalpy (h_3) increases as the R404A condensation temperature increases under the same conditions. The reason for the increase in the compressor power consumption is that the compression ratio increases as the R404A condensation temperature increases under the same condition. Furthermore, COP_{R404A} is greater than $COP_{R404A+CO_2}$ at the same R404A condensation temperature. The reason for this seems to be the heat loss of the CO_2 cooler and pipe. Furthermore, the pump power consumption appears constant because it is very small compared to the compressor power consumption. As the R404A condensation temperature changes from 30 to 50℃, however, the pump power consumption decreases by about 14% due to the lower mass flow rate of CO_2. As described above, the R404A evaporation heat capacity decreases because the R404A condenser outlet enthalpy (h_3) increases as the R404A condensation temperature increases under the same conditions. Furthermore, the CO_2 mass flow rate also decreases due to the energy balance because the conditions are identical to those in the CO_2 secondary refrigeration system. As shown in Fig. 4, it decreased by about 13.6% when the R404A condensation temperature increased from 30 to 50℃.

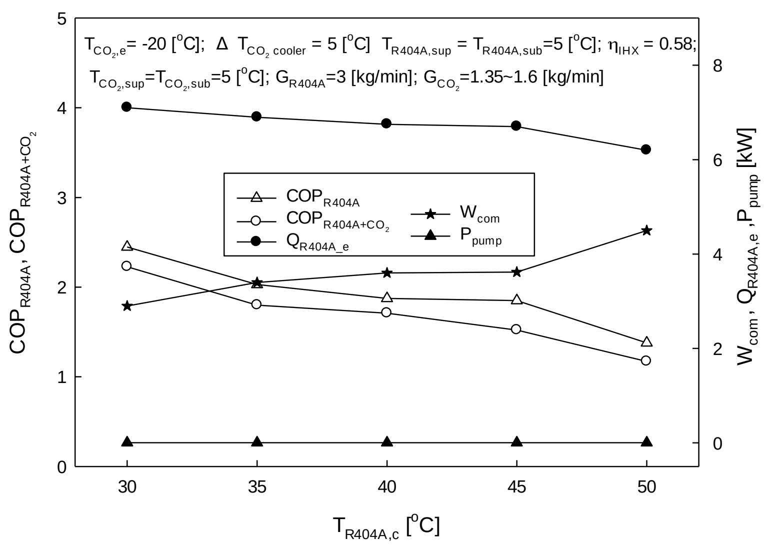

Fig. 3 COP, Q, and W of the indirect refrigeration system with respect to the variation of the R404A condensation temperatures.

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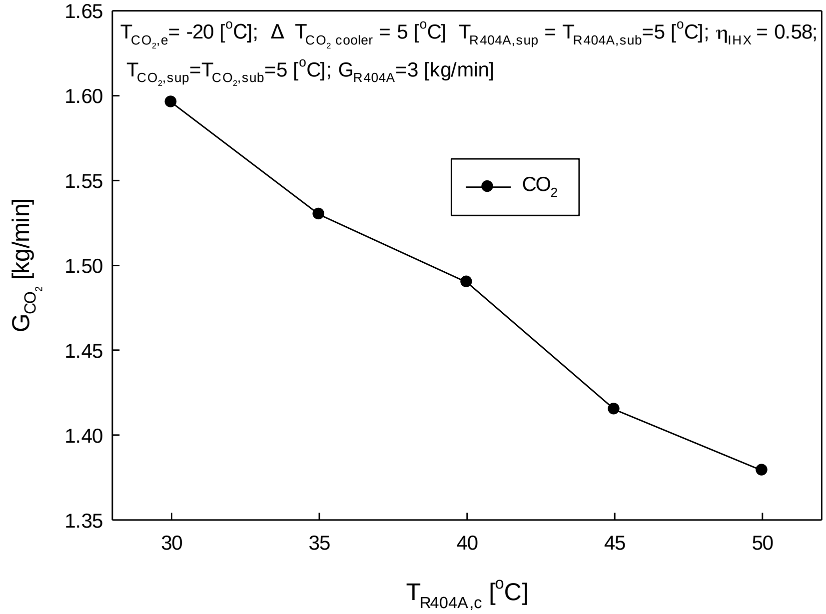

Fig. 4 Change of the CO_2 mass flow rate with respect to the R404A condensation temperature

Effects of Internal Heat Exchanger Efficiency

Fig. 5 presents the COP of the R404A refrigeration system (COP_{R404A}), the COP of the indirect refrigeration system ($COP_{R404A+CO_2}$), the R404A evaporation heat capacity ($Q_{R404A,e}$), compressor power consumption (W_{com}), and pump power consumption (P_{pump}) with respect to the increasing efficiency of the internal heat exchanger under the same conditions (R404A condensation temperature ($T_{R404A,c}$=40°C), CO_2 evaporation temperature ($T_{CO_2,e}$=-20°C), CO_2 cooler temperature difference ($\Delta T_{CO_2,cooler}$=5°C), subcooling degrees ($T_{R404A,sub}$, $T_{CO_2,sub}$=5°C), and superheating degrees ($T_{R404A,sup}$, $T_{CO_2,sup}$=5°C) of the R404A refrigeration system and CO_2 secondary refrigeration system. The internal heat exchanger efficiency (η_{IHX}) can be defined as follows:

$$\eta_{IHX} = \left(\frac{(T_1 - T_6)}{(T_3 - T_6)}\right) \qquad (7)$$

The case where the internal heat exchanger efficiency in Fig. 5 is 0 implies no internal heat exchanger (complete bypass), and the case where it is 1 implies an infinitely large heat exchanger with no heat loss. As the heat transfer area cannot be infinite, however, and as there is heat loss in actuality, the maximum value of the internal heat exchanger under the conditions of this experiment is 0.58.

As can be seen in Fig. 5, as the internal heat exchanger efficiency increased from 0 to 0.58 under the same conditions, and the COPs of the R404A refrigeration system and the indirect refrigeration system increased by 6.4 and 6.7%, respectively. This is because as the internal heat exchanger efficiency increased from 0 to 0.58, the R404A evaporation heat capacity increased by about 22.8%, and the compressor power consumption increased by about 15.4%. The R404A evaporation heat capacity increased here because the subcooling degree of the refrigerant at the inlet of the expansion valve increased as a result of the improved efficiency of the internal heat exchanger under the same conditions, thereby increasing the refrigeration effect. The reason that the compressor power consumption increased is that as the internal heat exchanger efficiency was improved under the same conditions, which increased the superheating degree of the refrigerant steam sucked into the compressor, thereby increasing the specific volume. Furthermore, due to the heat loss in the CO_2 cooler and pipe, the COP of the R404A refrigeration system becomes greater than that of the indirect refrigeration system. In addition, although the pump power consumption appears constant in Fig. 5 because it is very small compared to the compressor power consumption, the pump power consumption increases by 23.2% as the internal heat exchanger efficiency increases from 0 to 0.58. This is because the R404A evaporation heat capacity increases due to the improved efficiency of the internal heat exchanger under the same conditions, and the CO_2 mass flow rate increases due to the energy balance of the R404A refrigeration system and the CO_2 secondary refrigerant system. The changes in the CO_2 mass flow rate according to the increasing efficiency of the internal heat exchanger are shown in Fig. 6. The efficiency increased by about 21.6% when the internal heat exchanger efficiency increased from 0 to 0.58.

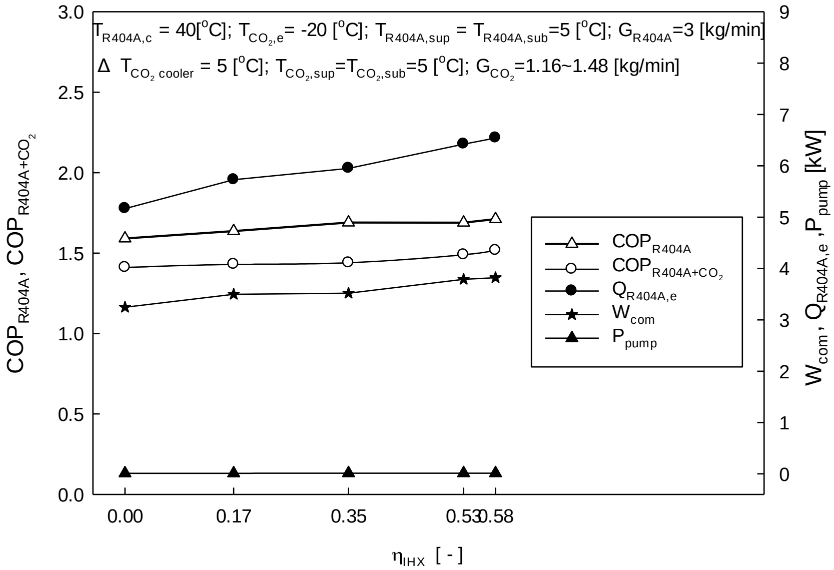

Fig. 5 COP, Q, and W of the indirect refrigeration system with respect to the variation of the efficiency of the internal heat exchanger.

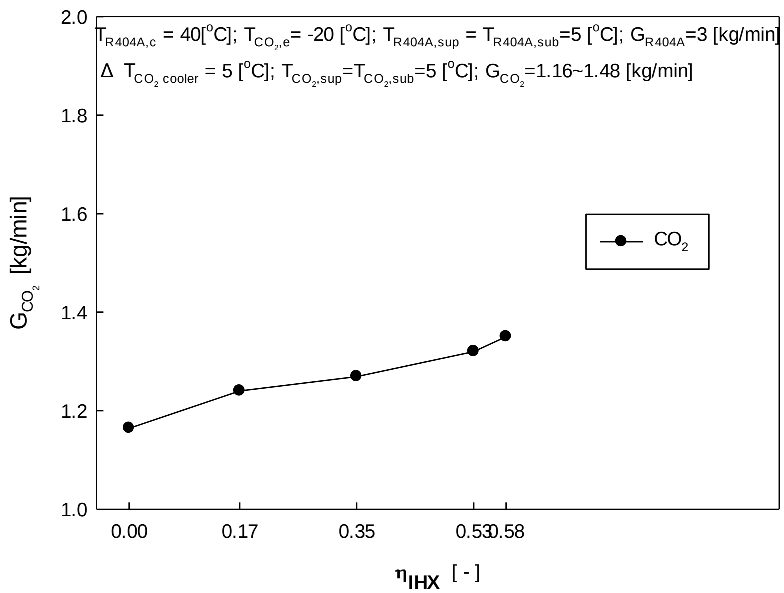

Fig. 6 Variation of the CO_2 mass flow rate with respect to the efficiency of the internal heat exchanger.

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Effects of the CO_2 Cooler Temperature Difference

Fig. 7 displays the COP of the R404A refrigeration system (COP_{R404A}), the COP of the indirect refrigeration system ($COP_{R404A+CO_2}$), R404A evaporation heat capacity ($Q_{R404A,e}$), compressor power consumption (W_{com}), and pump power consumption (P_{pump}) according to the increasing difference in the CO_2 cooler temperature under the same conditions (R404A condensation temperature ($T_{R404A,c}$=40℃), CO_2 evaporation temperature ($T_{CO_2,e}$=-20℃), internal heat exchanger efficiency (η_{IHX}=0.58), subcooling degrees ($T_{R404A,sub}$, $T_{CO_2,sub}$=5℃), and superheating degrees ($T_{R404A,sup}$, $T_{CO_2,sup}$=5℃) of the R404A refrigeration system and CO_2 secondary refrigeration system (=5℃). As can be seen in Fig. 7, as the CO_2 cooler temperature difference increased by 2℃, the COP of the R404A refrigeration system and that of the indirect refrigeration system decreased by about 4.3 and 4.2%, respectively. This is because each time the R404A condensation temperature increases by 5℃, the R404A evaporation heat capacity decreases by about 0.1% and the compressor power consumption increases by about 4.9%. The reason for the decrease in the R404A evaporation heat capacity herein is that the R404A evaporator outlet enthalpy (h_6) decreases as the R404A evaporation temperature decreases under the same conditions. The compressor power consumption increases because the compression ratio increases. Furthermore, although the pump power consumption appears constant in Fig. 7 because it is very small compared to the compressor power consumption, the pump power consumption decreased by about 0.6% when the CO_2 cooler temperature difference increased from 1 to 9℃. This is because the R404A evaporation heat capacity increases due to the increased difference in the CO_2 cooler temperature under the same conditions, and the CO_2 mass flow rate decreases due to the energy balance of the R404A refrigeration system and the CO_2 secondary refrigerant system. As can be seen in Fig. 8, the CO_2 mass flow rate decreased by about 0.5% when the CO_2 cooler temperature difference increased from 1 to 9℃.

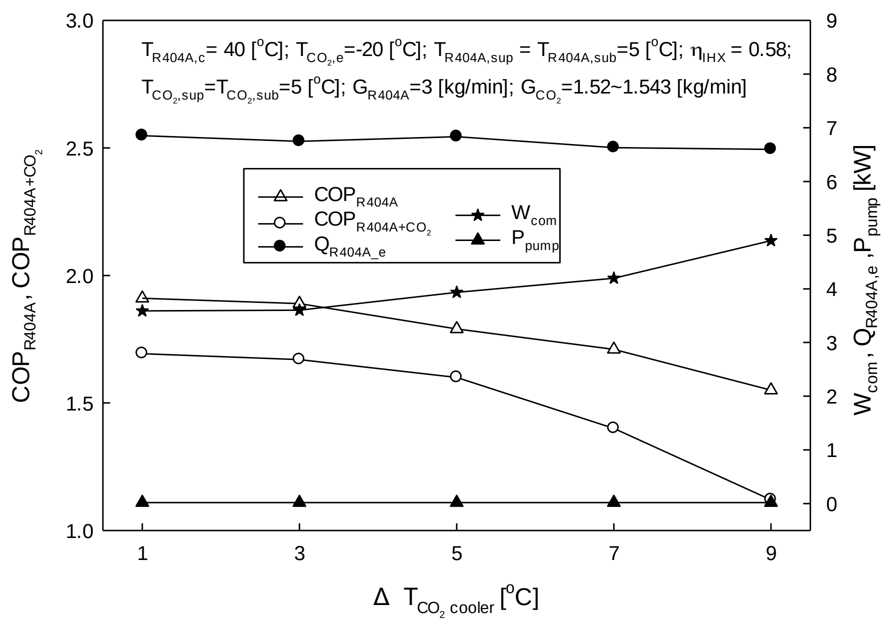

Fig. 7 COP, Q, and W of the indirect refrigeration system with respect to the variation of the temperature difference of the CO_2 cooler.

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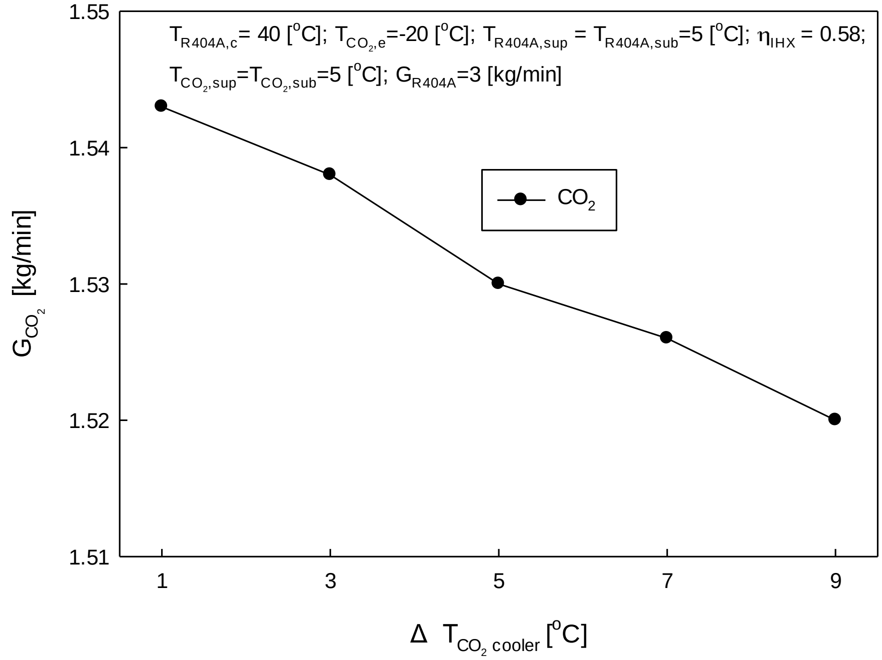

Fig. 8 Variation of the CO_2 mass flow rate with respect to the CO_2 cooler temperature difference

Effects of CO_2 Evaporation Temperature

Fig. 9 depicts the COP of the R404A refrigeration system (COP_{R404A}), the COP of the indirect refrigeration system ($COP_{R404A+CO_2}$), the R404A evaporation heat capacity ($Q_{R404A,e}$), compressor power consumption (W_{com}), and pump power consumption (P_{pump}) with respect to the increasing CO_2 evaporation temperature under the same conditions (R404A condensation temperature ($T_{R404A,c}$ =-40℃), CO_2 cooler temperature difference ($\Delta T_{CO_2,cooler}$=5℃), internal heat exchanger efficiency (η_{IHX} =0.58), subcooling degrees ($T_{R404A,sub}$, $T_{CO_2,sub}$=5℃), and superheating degrees ($T_{R404A,sup}$, $T_{CO_2,sup}$=5℃) of the R404A refrigeration system and CO_2 secondary refrigeration system.

As can be seen in Fig. 9, as the CO_2 evaporation temperature increased by 5℃, the COP of the R404A refrigeration system and that of the indirect refrigeration system increased by about 15.1 and 14.9%, respectively. This is because each time the CO_2 evaporation temperature increases by 5℃, the R404A evaporation heat capacity decreases by about 3.3% and the compressor power consumption decreases by about 11.4%. The reason for the decrease of the R404A evaporation heat capacity herein is that even though the R404A evaporation temperature increases as the CO_2 evaporation temperature increases under the same conditions, and the enthalpy (h_6) at the output of the R404A evaporator increases in the P-h diagram of Fig. 2, the decrease in the R404A mass flow rate due to the energy balance (as shown in Fig. 9) has a greater effect. Furthermore, the compressor power consumption decreases because as the R404A evaporation temperature increases, the compression ratio and the R404A mass flow rate decrease. In addition, the pump power consumption decreases by about 9.5% when the CO_2 evaporation temperature changes from -30 to -10℃. The reason for this seems to be that as shown in Fig. 10, when the CO_2 mass flow rate is constant, the viscosity decreases due to the increasing CO_2 evaporation temperature.

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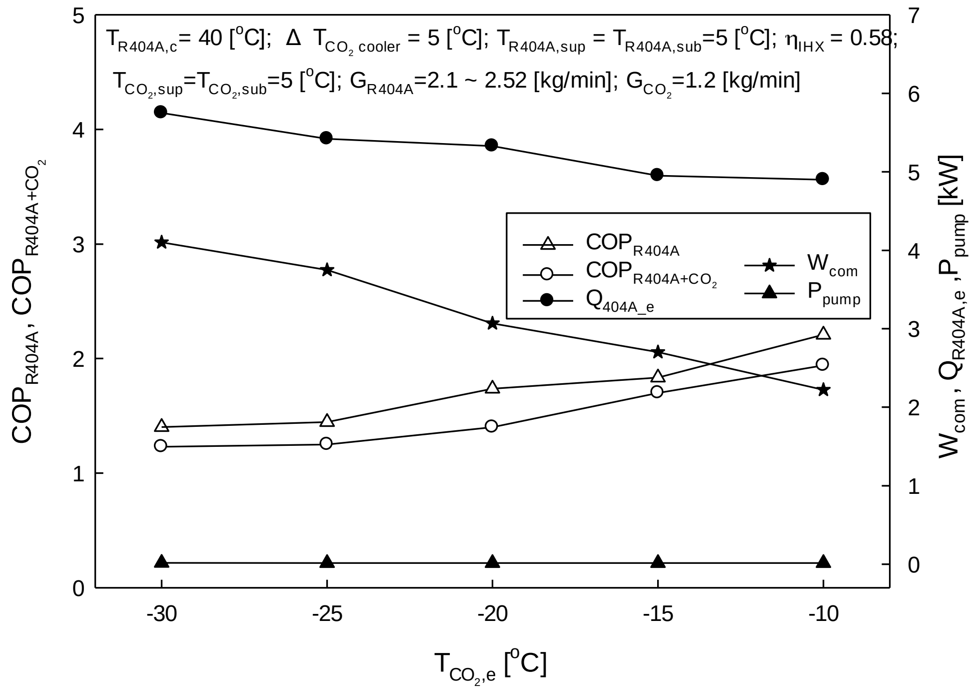

Fig. 9 COP, Q, and W of the indirect refrigeration system with respect to the variation of the CO_2 evaporation temperature.

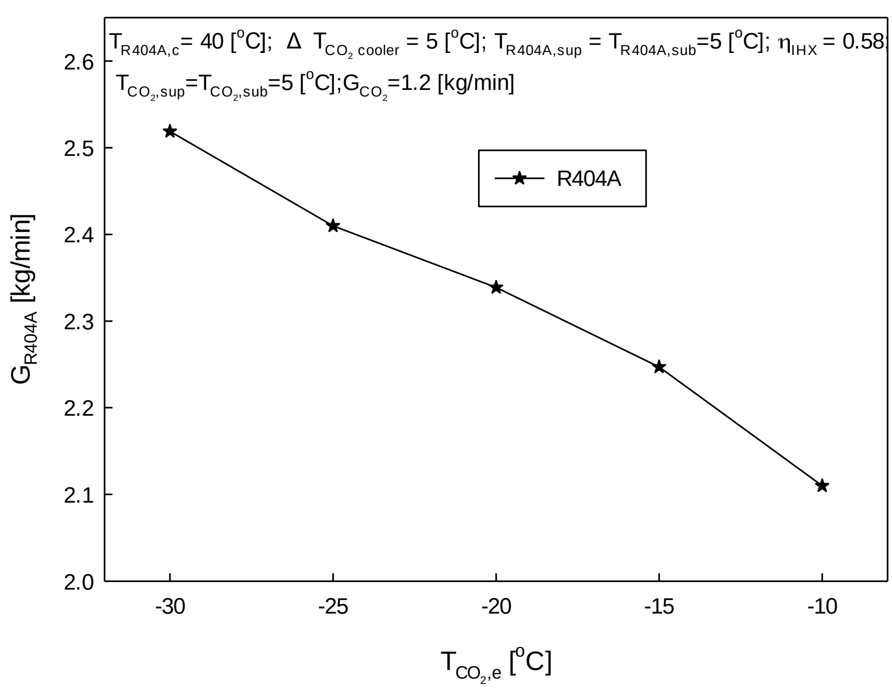

Fig. 10 Variation of the R404A mass flow rate with respect to the CO_2 evaporation temperature.

Effects of CO_2 Mass Flow Rate

Fig. 11 represents the COP of the R404A refrigeration system (COP_{R404A}), the COP of the indirect refrigeration system ($COP_{R404A+CO_2}$), the R404A evaporation heat capacity ($Q_{R404A,e}$), compressor power consumption (W_{com}), and pump power consumption (P_{pump}) according to the increasing R404A condensation temperature under the same conditions (R404A condensation temperature ($T_{R404A,c}$=40℃), CO_2 evaporation temperature ($T_{CO_2,e}$=-20℃), CO_2 cooler temperature difference ($\Delta T_{CO_2,cooler}$=5℃), subcooling degrees ($T_{R404A,sub}$, $T_{CO_2,sub}$=5℃), and superheating degree ($T_{R404A,sup}$, $T_{CO_2,sup}$=5℃) of the R404A refrigeration system and CO_2 secondary refrigeration system.

As can be seen in Fig. 11, as the CO_2 mass flow rate increased, the COPs of the R404A refrigeration system and indirect refrigeration system decreased very slightly. This is because the R404A evaporation heat capacity and compressor power consumption increased at almost constant rates of about 39.5 and 40.9%, respectively, when the CO_2 mass flow rate increased from 1.1 to 1.5 kg/min. The reason for the increase of the R404A evaporation heat capacity and compressor power consumption herein is that the R404A mass flow rate increases to maintain energy balance as the CO_2 mass flow rate increases. Furthermore, even though the increase in pump power consumption is smaller than the increases in the R404A evaporation heat capacity and compressor power consumption, it increased by about 36.1% when the CO_2 mass flow rate increased from 1.1 to 1.5 kg/min. The changes in the R404A mass flow rate according to the increasing CO_2 mass flow rate are shown in Fig. 12. It increased by about 35.7% when the CO_2 mass flow rate increased from 1.1 to 1.5 kg/min.

To summarize the above description, as the R404A condensation temperature and CO_2 cooler temperature difference increased, the COPs of the R404A refrigeration system and the indirect refrigeration system decreased, and the increase in the CO_2 mass flow rate had a very slight effect on the COPs of the R404A and indirect refrigeration systems. Furthermore, as can be seen from the changing rate of the pump power consumption in each experiment, the CO_2 mass flow rate has a greater effect on the pump power consumption than the coefficient of viscosity of CO_2 does

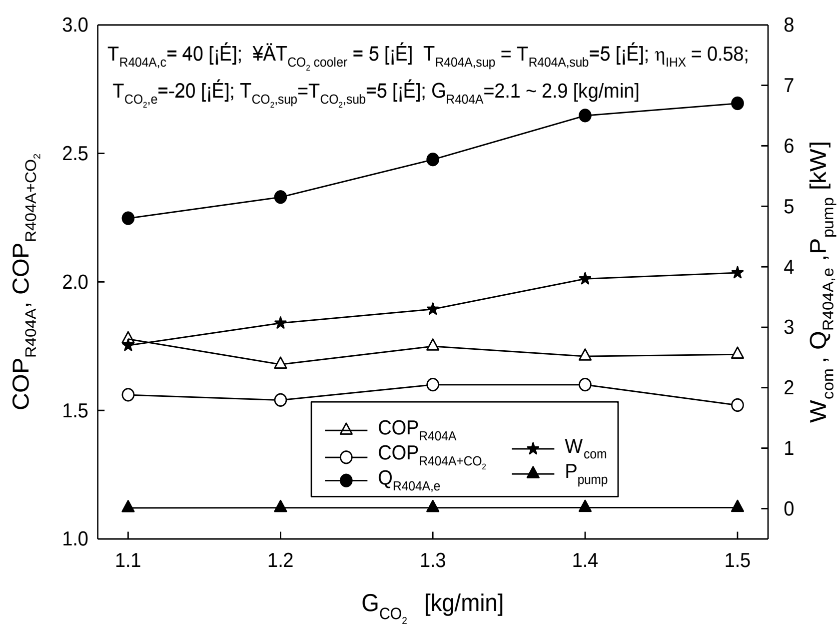

Fig. 11 COP, Q, and W of the indirect refrigeration system with respect to the variation of the mass flow rate of CO_2.

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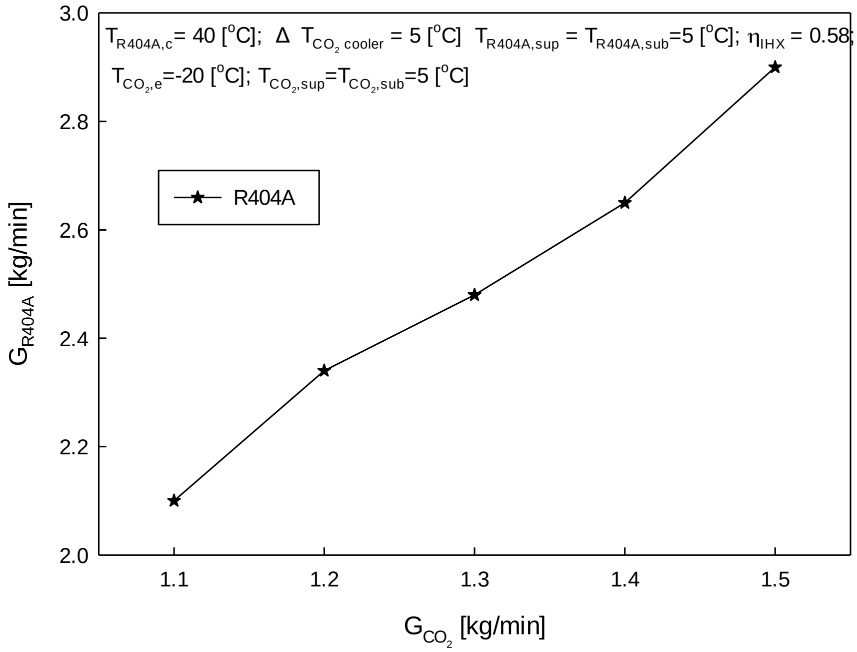

Fig. 12 Variation of the R404A mass flow rate with respect to the CO_2 mass flow rate.

Conclusions

In this study, the effects of such factors as the R404A condensation temperature, the efficiency of the R404A internal heat exchanger, and the difference in CO_2 cooler temperature, CO_2 evaporation temperature, and CO_2 mass flow rate of the indirect refrigeration system using CO_2 as a secondary refrigerant on the system performance were investigated. The major findings of this study can be summarized as follows.

Under certain experimental conditions, the lower the R404A condensation temperature and the CO_2 cooler temperature difference were, the greater the COP of the total indirect refrigeration system, including the R404A and CO_2 refrigerant circulation systems, became. Furthermore, the higher the internal heat exchanger efficiency for R404A and the CO_2 evaporation temperature were, the greater the COPs of the R404A refrigeration system and the CO_2 refrigerant circulation system became. In addition, the lower the CO_2 mass flow rate was, the greater the COPs of the R404A refrigeration system and the CO_2 refrigerant circulation system became, but its effect was very small. Lastly, the pump power consumption was influenced more by the CO_2 mass flow rate than by the CO_2 evaporation temperature.

The above findings show that the COP of the R404A indirect refrigeration system using CO_2 as a secondary refrigerant and the COP of the total refrigeration system are influenced by such parameters as the R404A condensation temperature, R404A internal heat exchanger efficiency, CO_2 cooler temperature difference, CO_2 evaporation temperature, and CO_2 mass flow rate. Therefore, the optimum indirect refrigeration system can be constructed by designing an R404A indirect refrigeration system using CO_2 as a secondary refrigerant by considering these parameters

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