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Experimental study on heat transfer characteristics of internal heat exchangers for CO_2 system under cooling condition †

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Abstract

This paper presents the heat transfer characteristics of the internal heat exchanger (IHX) for CO_2 heat pump system. The influence on the IHX length, the mass flow rate, the shape of IHX, the operating condition, and the oil concentration was investigated under a cooling condition. Four kinds of IHX with a coaxial type and a micro-channel type, a mass flow meter, a pump, and a measurement system. With increasing of the IHX length, the capacity, the effectiveness, and the pressure drop increased. For the mass flow rate, the capacity of micro-channel IHX are higher about 2 times than those of coaxial IHX. The pressure drop was larger at cold-side than at hot-side. In the transcritical CO_2 cycle, system performance is very sensitive to the IHX design. Design parameters are closely related with the capacity and the pressure drop of CO_2 heat pump system. Along the operating condition, the performance of CO_2 IHXs is different remarkably. For oil concentration 1, 3, 5%, the capacity decreases and the pressure drop increased, as compared with oil concentration 0%.

Keywords: CO_2; Internal heat exchanger(IHX); Transcritical cycle; Heat transfer; Pressure drop

1. Introduction

Due to environmental concerns, HFC refrigerants such as R410A have been considered as the replacements for R22 of air conditioning application. In recent years, CO_2 has been reintroduced because CO_2 naturally exists in the atmosphere and has a lower GWP than HFCs. It has better thermo-physical properties such as a low viscosity, a high specific heat, a high volumetric heat capacity and a high thermal conductivity, compared with the conventional refrigerants.

Since the first publication by Lorentzen and Pettersen [1], CO_2 system has been attracted attention both in industry and academia. But, the performance of CO_2 system has been not completely resolved. Successful application of CO_2 system depends on the development of efficient and compact components with low weight, good reliability, and suitable performance. Internal heat exchanger (IHX) is one of components among many possible technologies enhancing the performance of CO_2 system. Yin et al. [2] and McEnaney et al. [3] compared experimentally a conventional system with CO_2 system. They showed that CO_2 system had slightly lower capacity and COP at very high ambient temperature but much higher capacity and COP at lower ambient temperatures. Domanski et al. [4] studied experimentally the system performance and the thermodynamic benefits of IHX using CO_2 and R-134A. Park et al. [5] analyzed the thermal performance on IHX with a circular coil type.

Since CO_2 has thermodynamic disadvantage of a transcritical cycle, it requires the use of the improved

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refrigerant system. Due to its very high operating pressure, there are many difficulties to develop CO_2 system comparable to the existing systems in aspects of efficiency, reliability, safety and compactness. To improve the performance of CO_2 system, Boewe et al. [6] applied IHX to the automotive air-conditioning system. They showed that the capacity of a system with IHX could be increased up to about 25% for automotive air-conditioning system, compared with that of one without IHX. In the transcritical CO_2 cycle, system performance is very sensitive to the shape of IHX.

Thus, more comprehensive and systematic study for IHX is required to develop the performance of air-conditioning and heat pump system. Especially, to develop a compact and high performance IHX for CO_2, the study on heat transfer characteristics of IHX should be necessary.

In the present study, the heat transfer characteristics on four types of IHX have been investigated experimentally at the cooling condition and effects of an IHX length, a mass flow rate, an IHX shape, a system operation condition and an oil concentration are introduced. These results can be utilized in the design of a compact air-conditioning systems using CO_2.

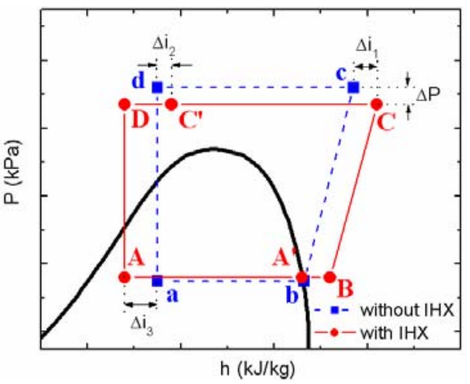


Fig. 1. Pressure-enthalpy diagram of CO_2 cycle.

2. Fundamental of IHX

CO_2 system has a transcritical cycle. So, the working pressure of CO_2 cycle is very higher than of that the existing refrigerant systems, because of the specific thermodynamic properties of CO_2. The critical pressure and temperature of CO_2 are 7.38 MPa and 31.3 ℃, respectively. Its vapor pressure is much higher than CFC, HCFC, HFC and HC refrigerants. High pressure and density can reduce the pipe and vessels size, and improve heat exchanger effectiveness. This leads to the need of new design of IHX for CO_2 system. IHX has been used to exchange energy between the high-temperature/high-pressure refrigerant leaving the gas cooler and the low-temperature/ low-pressure refrigerant leaving the evaporator. To fully bring out the potential IHX benefits, a system integrator has to reduce mass flow rate by superheated refrigerant vapor and higher evaporating temperature.

Fig. 1 shows pressure-enthalpy diagram of CO_2 cycle. The schematic diagram of the CO_2 system in the present study is shown in Fig. 2. Fig. 2(a) and (b) are the basic cycle without IHX(a-b-c-d) and CO_2 cycle with IHX(A-A`-B-C-C`-D), respectively. As shown in Fig. 2(b), the refrigerant from the gas cooler exit flows into the expansion valve with a low temperature as (C'-D). This increases the cooling effect ($\Delta i_3 = \Delta q$). The refrigerant from the evaporator exit is superheated as (A'-B), and then flows into the compressor with the specific volume increased by superheating. The refrigerant flow rate and the compressor discharge pressure (ΔP) decrease. Thus, the compressor work decreases. Theoretically, the system with IHX can prove higher COP and better cooling capacity ($q+\Delta q$).

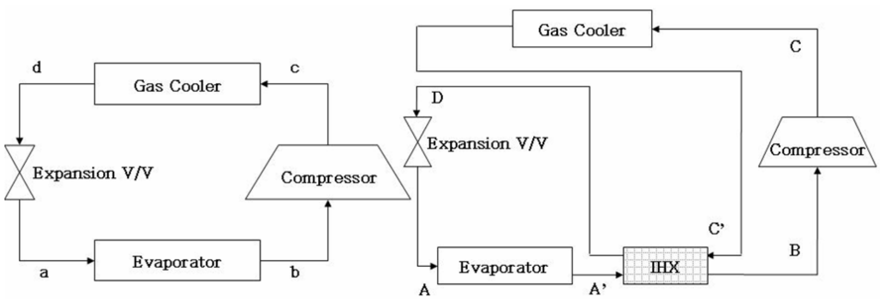


Fig. 2. Schematic diagram for CO_2 system used in the present study ((a) without IHX, (b) with IHX).

Table 1. Experimental conditions.

Parameters	Specifications
Mass flow rate (g/s)	40, 50, 60, 70, 80
$T_{hot,in}$ (℃)	35, 36, 37, 38, 39
$T_{cold,\ in}$ (℃)	Quality x at saturate temp. 0℃ : 0.94, 0.96, 0.98, 1.00
	degree of superheat : 0, 3, 5 ℃
$P_{hot,in}$ (kPa)	9,000
$P_{cold,\ in}$ (kPa)	3,485
Length (m)	0.5, 1.0, 1.5, 2.0, 2.5

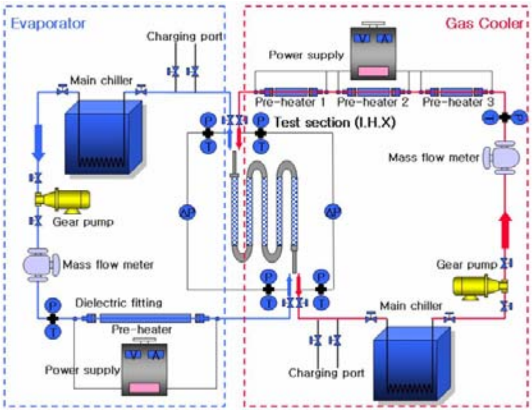


Fig. 3. Schematic diagram of CO_2 experimental apparatus.

From these considerations, the improvement of COP and capacity Q of CO_2 system with IHX can be expressed as follows:

$$COP_{IHX} = \frac{q+\Delta q}{w+\Delta w} \approx \frac{q}{w}\left(1+\frac{\Delta q}{q}-\frac{\Delta w}{w}\right)$$
$$= COP\left(1+\frac{\Delta q}{q}-\frac{\Delta w}{w}\right) > COP \tag{1}$$
$$\frac{COP_{IHX}}{COP} = \left(1+\frac{\Delta q}{q}-\frac{\Delta w}{w}\right) > 1 \tag{2}$$

The subscript "IHX" refers to CO_2 cycle with IHX. Since the mass flow rate m_{IHX} is less than m, one has to design and control the system such that Δq is sufficient ($Q_{IHX} \geq Q$) and w_{IHX} is less than w.

$$\frac{Q_{IHX}}{Q} = \frac{m_{IHX}(q+\Delta q)}{mq} > 1 \tag{3}$$

From these, we can see that CO_2 cycle with IHX has higher capacity, COP, better system performance with less refrigerant charge, lower compressor discharge pressure, better compressor protection (no wet compression) and better compact heat exchanger [7].

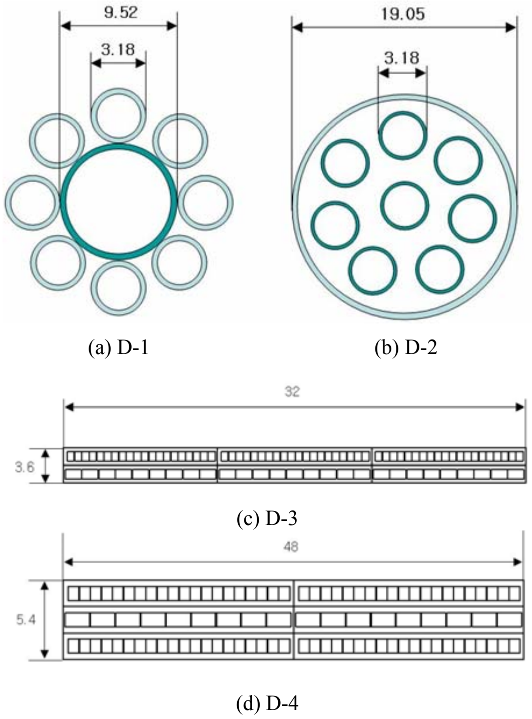


Fig. 4. Cross-section of IHXs used in the present study.

3. Experiment

3.1 Experimental apparatus and method

The schematic diagram of CO_2 experimental apparatus is shown in Fig. 3. It has two loops and consists of a test section, a magnetic gear pump, a chiller, a mass flow meter, a pre-heater and a power supply. The mass flow meter is installed before the pre-heaters to measure the flow rate of the liquid refrigerant. The subcooled refrigerant is heated by an electric current supplied by a DC power supply.

Experimental conditions are shown in Table 1. The temperatures were measured with RTD sensors and T-type thermocouples calibrated using a standard thermometer and a calibrator. Pressures were monitored using a pressure transducer. A differential pressure transducer was used to measure pressure drop along the test section. The test section was insulated by rubber foam with 40mm thickness.

All tests were run at steady state conditions. Data acquisition was done by DAQ system using GPIB communication. Steady state condition was decided by the deviation of measuring data. The data were obtained during 15minutes, when the deviation of temperature, pressure and mass flow rate were within ±0.1℃, ±0.5 kPa and 3%, respectively.

The classifications of IHXs are shown in Table 2. Fig. 4 shows the cross-section of IHXs used in the present study. Fig. 4(a) shows coaxial IHX with an inner large tube and outer small tubes (D-1). Fig. 4(b) shows coaxial IHX with an outer large tube and inner small tubes (D-2). Fig. 4(c) and (d) show the micro-channel IHX with two-pass (D-3) and three-pass (D-4), respectively.

3.2 Data reduction

Capacity of IHX was measured with a mass flow rate, and temperature. It can be calculated by

$$Q = \dot{m}\left(h_{cold,out} - h_{cold,in}\right) \tag{4}$$

Table 2. Classifications of IHXs.

<table>
<tr><th colspan="2">Type</th><th>Arrange-ment</th><th>Number of tubes or holes</th><th>Material</th></tr>
<tr><td>D-1</td><td>Coaxial (outer)</td><td rowspan="4">Counter flow</td><td>Hot : 1
Cold : 8
(tube)</td><td rowspan="2">Copper</td></tr>
<tr><td>D-2</td><td>Coaxial (inner)</td><td>Hot : 8
Cold : 1
(tube)</td></tr>
<tr><td>D-3</td><td>Micro-channel (2pass)</td><td>Hot : 60
Cold : 27
(holes)</td><td rowspan="2">Aluminum</td></tr>
<tr><td>D-4</td><td>Micro-channel (3pass)</td><td>Hot : 18
Cold : 80
(holes)</td></tr>
</table>

where $\dot{m}$ is a mass flow rate of the refrigerant, $h_{cold,out}$ is the enthalpy of a cold-side outlet and $h_{cold,in}$ is the enthalpy of a cold-side inlet.

The effectiveness of IHX is expressed as follows;

$$\varepsilon = \frac{Q_{\exp}}{Q_{\max}} = \frac{T_{cold,out} - T_{cold,in}}{T_{hot,in} - T_{cold,in}} \tag{5}$$

where $T_{cold,out}$ is the temperature of a cold-side outlet, $T_{cold,in}$ is the temperature of a cold-side inlet and $T_{hot,in}$ is the temperature of a hot-side inlet. The properties of CO_2 were calculated by using REPROP 6.01 [8].

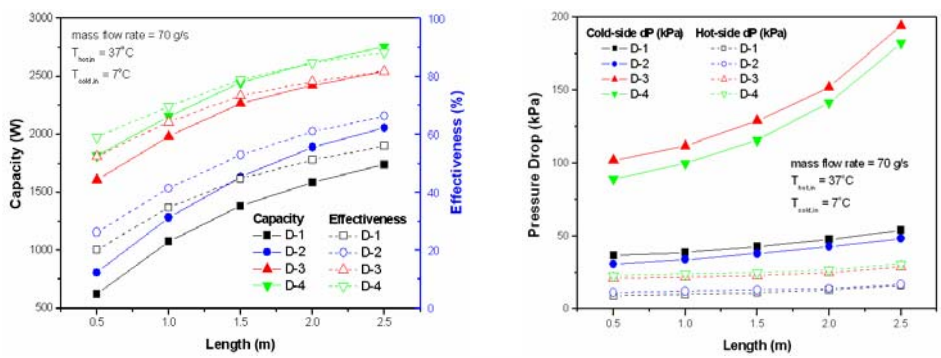
4. Results and discussions

4.1 IHX length

Fig. 5 shows the variation of the capacity and the effectiveness (a), and the pressure drop (b) along the IHX length. With increasing of the IHX length (from 0.5 m to 2.5 m), the capacity, the effectiveness and the pressure drop increase. When IHX length is 0.5m, the capacity and the effectiveness of micro-channel IHX are larger about 2.4 times them of coaxial IHX. When IHX length is 2.5m, the capacity and the effectiveness of micro-channel IHX are larger about 1.4 times than those of coaxial IHX.

Also, the increase ratios of the capacity and the effectiveness of coaxial IHX are larger than those of micro-channel IHX. When IHX length increases from 0.5m to 2.5m, the capacity increases about 2.4 times (D-1), 2.6 times (D-2), 1.6 times (D-3) and 1.5 times (D-4) also the effectiveness increases 2.8 times (D-1), 2.5 times (D-2), 1.6 times (D-3) and 1.5 times (D-4).

With increasing IHX length, the pressure drop of the hot-side increases 1~2 kPa for coaxial IHX, 2~4 kPa



(a) (b)

Fig. 5. Capacity and effectiveness(a), and pressure drop(b) on IHX length.

for micro-channel IHX, and that of cold-side increases 3~17 kPa for coaxial IHX, 10~100kPa for micro-channel IHX. That is, the pressure drop of hot-side increases smoothly, but that of cold-side increases remarkably.

Fig. 5 shows that the optimum length of IHX to improve the performance of IHX should be selected by considering both the capacity and the pressure drop of IHX within the operation range of CO_2 system.

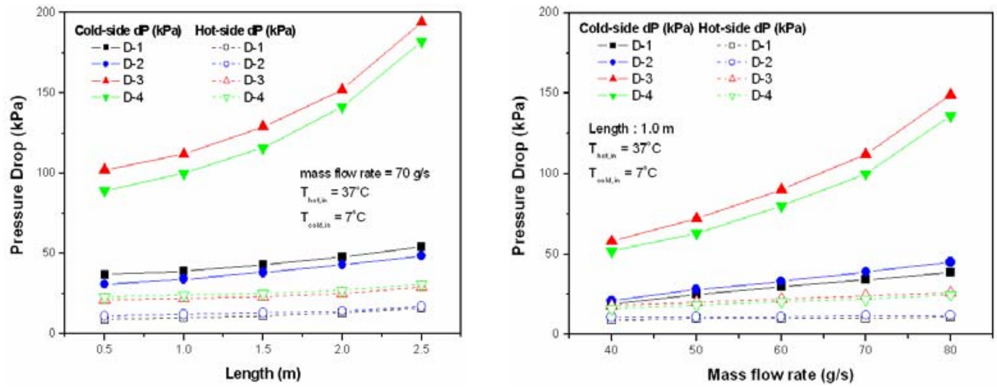
4.2 Mass flow rate

Fig. 6 shows the variation of the capacity, the effectiveness (a) and the pressure drop (b) along the mass flow rate. With increasing of the mass flow rate, the capacity increases about 25% per 10 g/s. The capacity of micro-channel IHX is higher about 2 times than that of coaxial IHX. However, the increase of the effectiveness is insignificant.

As the mass flow rate increases, the pressure drop of micro-channel IHX is larger than that of coaxial IHX. The pressure drop of the cold-side increases 1 kPa for coaxial IHX, 2 kPa for micro-channel IHX and that of cold-side increases 5~7 kPa for coaxial IHX, 10~40 kPa for micro-channel IHX. That is, the pressure drop of hot-side increases linearly for coaxial IHX and micro-channel IHX. But the pressure drop of cold-side increases remarkably for micro-channel IHX and increases linearly for coaxial IHX.

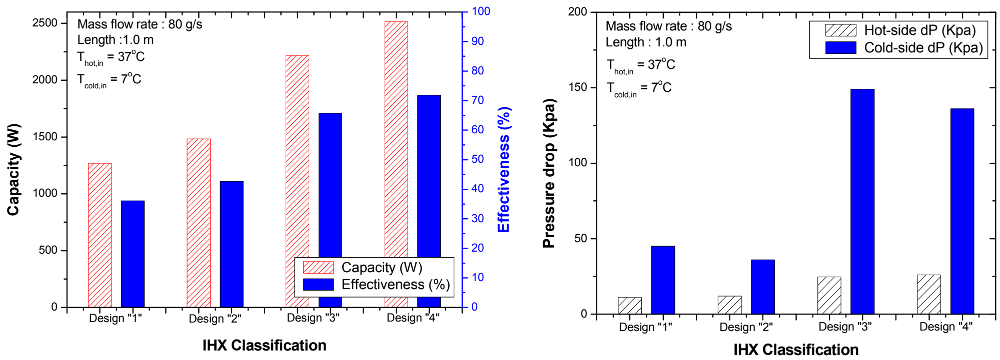
4.3 Design shape of IHX

Fig. 7 shows the capacity and the effectiveness (a), and the pressure drop (b) on D-1, D-2, D-3 and D-4 IHXs, Data were obtained under the conditions of the hot-side temperature 37℃, the cold-side temperature 7℃, the IHX length 1.0 m, the mass flow rate 80 g/s. As shown in Fig. 7(a), the capacity and the effectiveness of micro-channel IHXs (D-3 and D-4) are larger than those of coaxial IHXs (D-1 and D-2). The capac-



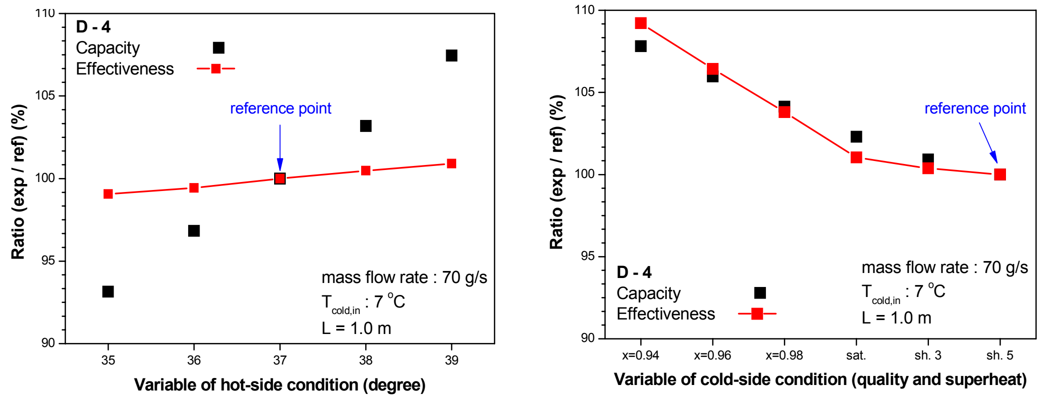
(a) (b)

Fig. 6. Capacity and effectiveness(a) and pressure drop(b) on mass flow rate.



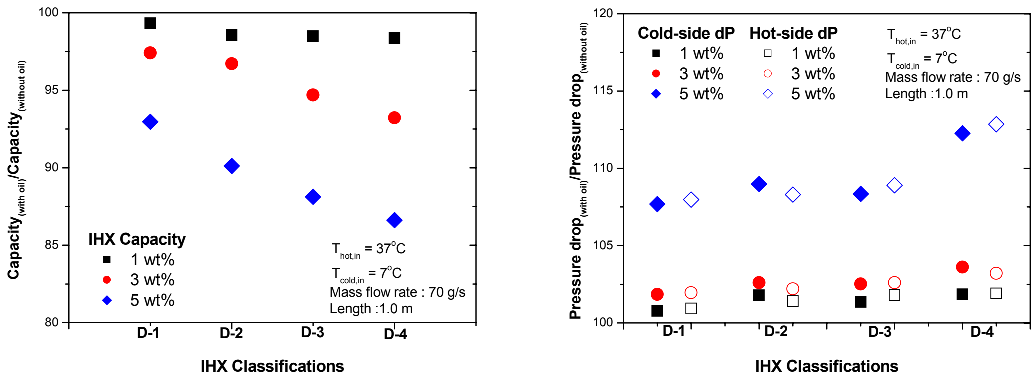
(a) (b)

Fig. 7. Capacity and effectiveness(a) and pressure drop(b) on IHX shape.



(a) (b)

Fig. 8. Capacity and effectiveness on operating condition at hot-side(a) and cold-side(b).



(a) (b)

Fig. 9. Capacity(a) and pressure drop(b) on oil concentration.

ity and the effectiveness increases about 1.2 times (D-2), 1.8 times (D-3), and 2 times (D-4), as compared with those of D-1 IHX. As shown in Fig. 7(b), the pressure drop of micro-channel IHXs (D-3 and D-4) are much larger than those of coaxial IHXs (D-1 and D-2). The pressure drop increases about 1.1 times (D-2), 2.3 times (D-3) and 2.4 times (D-4) for hot-side and about 0.8 times (D-2), 3.3 times (D-3) and 3 times (D-4) for cold-side, as compared with that of D-1 IHX.

From these, we can see that the system performance of the transcritical CO_2 cycle is sensitive to the design shape of IHX. To design IHX with better capacity and lower pressure drop. Thus, the systematic study on the matching of CO_2 cycle is needed.

4.4 Operating conditions

From experimental results on IHX length, mass flow rate and design shape of IHX, D-4 is selected as IHX for CO_2 cycle. Thus the performance on hot-side and cold-side operating conditions is investigated.

Fig. 8 shows that the capacity and the effectiveness on the inlet condition of hot-side and cold-side for micro-channel IHX (D-4). Hot-side of Fig. 8(a) and cold-side of Fig. 8(b) indicate the gas-cooler condition and the evaporator condition, respectively. Temperature of reference point is 37℃ (gas-cooler operating temperature) at hot-side condition and 5℃ (degree of superheat of evaporator operating temperature) at cold-side condition. Also, cold-side conditions are divided as the superheat region (degree of superheat 5, 3 and 0℃) and the two-phase region (quality 1, 0.98, 0.96 and 0.94 at 0℃).

As shown in Fig. 8(a), when the inlet temperature of hot-side is changed from 35℃ to 39℃, the capacity and the effectiveness increase about 3.6% and

0.5% per 1℃, respectively. As shown in Fig. 8(b), when the degree of superheat is changed from 5℃ to 3℃ and 0℃, the capacity increases about 1% and when quality is changed from 1 to 0.94, the capacity increases about 2%, per quality 0.02. The effectiveness increases about 0.4% at the superheat region and 2.6% at the two-phase region. Although the inlet temperature of the low-pressure side is constant, the latent enthalpy region increases due to the decrease in quality. Thus the total IHX capacity increases. This shows that the safety system design as well as the capacity enhancement and compactness is possible, if the evaporator exit condition is considered as not the superheat region but the two-phase region.

4.5 Oil concentration

Fig. 9 shows the decrease ratio of the capacity (a) and the increase ratio of the pressure drop (b) of IHXs when the concentrations of PAG oil are 1, 3 and 5%. With increasing the oil concentration, the capacity of IHXs decreases linearly. For the oil concentration 1, 3, 5%, the decrease in the capacity is 0.6, 2.2 and 6.4% at D-1, 0.7, 2.6 and 7.1% at D-2, 1.5, 3.3 and 10.9% at D-3, and 1.9, 7.1 and 16.7% at D-4, as compared with oil concentration 0%. This is due to the liquid film formed by oil at the inner surface of CO_2 IHX. This oil film decreases CO_2 heat transfer capacity.

With increasing the oil concentration, the pressure drop of IHXs increases. The pressure drop of micro-channel IHX (D-3 and D-4) is larger than that of coaxial IHXs (D-1 and D-2). For the oil concentration 1, 3, 5%, the increase in the pressure drop of hot-side is 1, 2 and 8% at D-1, 1.4, 2.2 and 8.3% at D-2, 1.8, 2.6 and 8.9% at D-3, and 1.9, 3.2 and 12.8% at D-4, and that of cold-side is 0.8, 1.8 and 7.7% at D-1, 1.8, 2.6 and 9% at D-2, 1.4, 2.5 and 8.4% at D-3, and 1.9, 3.6 and 11.3% at D-4, as compared with oil concentration 0%.

5. Conclusions

In this work, the heat transfer characteristics of IHX for CO_2 system was studied under a cooling condition. With increasing of the IHX length, the capacity, the effectiveness, and the pressure drop increase. The capacity and the effectiveness of micro-channel IHX are larger about 1.4～2.4 times than that of coaxial IHX. This shows that the optimum length of IHX should be selected as considering both the capacity and the pressure drop. For the mass flow rate, the capacity increases about 25% per 10 g/s. The capacity of micro-channel IHX is higher about 2 times than that of coaxial IHX. The pressure drop is larger at cold-side than at hot-side. The system performance of the transcritical CO_2 cycle is sensitive to IHX shape. It is closely related with the capacity and the pressure drop of CO_2 system. At hot-side operating condition, the capacity and the effectiveness increase about 3.6% and 0.5% per 1℃. At cold-side condition, the capacity increases about 1% with decreasing the degree of superheat and about 2% with decreasing quality. With increasing the oil concentration, the capacity decreases linearly and the pressure drop increases, as compared with oil concentration 0%. To develop IHX with better capacity and lower pressure drop, the systematic study on the matching between CO_2 IHX and the system cycle is needed.

Acknowledgment

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Nomenclature

COP	:	Coefficient of performance
h	:	Enthalpy [kJ/kg]
L	:	Length [m]
$\dot{m}$	:	Mass flow rate [kg/s]
Q	:	Capacity [W]
q	:	Capacity per mass flow rate [W/kg]
T	:	Temperature [℃]
W	:	Compressor work [W]
x	:	Quality
ε	:	Heat exchanger effectiveness

Subscript

$cold$	:	Cold side
hot	:	Hot side
in	:	IHX inlet
out	:	IHX outlet
max	:	Maximum data
exp	:	Experimental data

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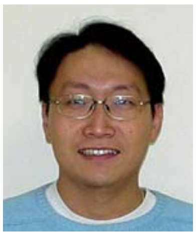

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