GAS COOLING HEAT TRANSFER AND PRESSURE DROP CHARATERICTICS OF CO_2/OIL MIXTURE IN A MICROCHANNEL

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ABSTRACT

The effects of oil on the gas cooling heat transfer and pressure drop characteristics of CO_2 in microchannel are investigated in this study. The hydraulic diameter of the microchannel used in the experiment is 1.0 mm. The oil used is PAG. The mass flux was varied from 200 to 400 kg/m^2s, the heat flux was 20 and 25 kW/m^2, and the test section inlet pressures of CO_2 were changed from 8.4 to 10.4 MPa. The test section inlet temperatures were controlled from 40 to 80°C. The oil mass fraction (OMF) of the test was 0, 1, 2, 3, and 4 wt.%. The water cooling loop was installed to remove the heat from the test section. The heat transfer coefficients of CO_2 were calculated by the heat transfer rate between water and CO_2. The water-side heat transfer coefficient that obtained by the Wilson plot method. The test results show that a 9.6% degradation of average gas cooling heat transfer coefficients for 2 wt.% OMF, and a 20.4% of degradation for 4 wt.% OMF compared with that of pure CO_2. The average pressure drop increases by 2.9 times for 2 wt.% OMF, and by 4.8 times for 4 wt.% OMF when they were compared with that of pure CO_2. As demonstrated, the influence of the oil concentration on the gas cooling heat transfer coefficient and pressure drop of the CO_2/oil mixture is significant. Therefore, it is recommended to minimize the oil concentration less than 1 wt.%.

1. INTRODUCTION

CO_2 has been reintroduced as a working fluid for various refrigeration and air conditioning systems such as automobile air-conditioners, heat pump water heaters, and small capacity refrigeration systems due to environmental concerns. The CO_2 system can favourably utilizes microchannel heat exchangers as its gas cooler and evaporator due to its high operating pressures. During the operation of vapor compression cycle, the oil circulates through all components of the system although the oil is only necessary for the compressor. The lubricating oil in heat exchangers deteriorates the heat transfer performance and increases pressure drop, especially its impact is greater in microchannel than in conventional fin-and-tube heat exchanger. Therefore, understanding the effects of oil on the heat transfer coefficient and pressure drop is very essential in developing microchannel heat exchangers appropriate for the transcritical CO_2 cycle. Some researchers investigated these issues as summarized next. Pettersen et al. (2000) studied gas cooling heat transfer and pressure drop characteristics of pure CO_2 in microchannel with a diameter of 0.79 mm. The test section inlet pressure of CO_2 was varied from 8.1 to 10.1 MPa, and the temperature ranged from 20 to 60°C. Mass flux was changed from 600 to 1200 kg/m^2s. They showed that the Gnielinski heat transfer model and the Colebrook and White pressure drop model well predicted their test results with 4% and 2% of mean deviation, respectively. Kuang et al. (2004a) compared their test results of gas cooling heat transfer coefficients of CO_2 in microchannel with the Gnielinski model (1976). The model predicted their experimental results within 10% of mean deviation in a low mass flux range. However, in high mass flux region, the mean deviations between the test results and the predicted data were as much as 60%. Kuang et al. (2004b) also investigated the effects of different types of oil on the gas cooling heat transfer coefficients of CO_2 in microchannel with hydraulic diameter

0.79 mm. The oils used in their study were PAG, POE, and AN/PAG. They found that the heat transfer coefficients were reduced by up to 57% and the pressure drop increased 44% for 5 wt.% OMF. However, their test conditions were very limited in the operating pressure and the mass flux ranges. Their tests were conducted under fixed CO_2 pressure and mass flux conditions of 9 MPa and 844 kg/m^{2}s, respectively. The average temperature of CO_2 across a test section was also limited from 30 to 50°C. The objective of this study is to investigate the effects of PAG oil on the gas cooling heat transfer coefficient and pressure drop of CO_2 in wider operation conditions, which are ranged with the actual system operation in mind.

2. EXPERIMENTAL SET-UP

Figure 1 shows the schematic of the test set-up. Basically, it has two loops. One is the vapor compression loop and the other is the test section loop. The vapor compression loop was utilized to attain the target test conditions of CO_2, such as mass flux, temperature and pressure. The CO_2 flow divided from the gas cooler flows into the test section loop. The mass flow rate of CO_2 at the test section was controlled by adjusting the opening of the needle valve. The cooling water circulated through the test section by a pump and removed the heat from the CO_2 so that it undergoes the gas cooling process. The cooling water supply temperature was set by the constant temperature water bath. The syringe pump was used to inject oil to the test section. Three oil separators were installed to ensure the oil from the compressor was separated and prevented from flowing into the test section so that only the oil from the syringe pump flows into the test section. The injected oil to the test section was sent back to the compressor. The flow visualization section was installed between the test section and the syringe pump to check the oil flow to the test section. The test microchannel has 10 holes whose hydraulic diameter is 1.0 mm. The width, height, and length are 16 mm, 2 mm, and 600 mm, respectively. Water channel, which was built around the microchannel, was made by polycarbonate sheets having milled groove. The pressures of CO_2 at the inlet and outlet of the test section were measured by pressure transducers whose ranges are from 0 to 21.4 MPa. It has ± 0.11% accuracy of the full scale. The CO_2 and water temperatures at the inlet and outlet of the test section were measured by RTDs that have an error of ±0.02°C. The mass flow rate of CO_2/oil mixture through the test section was measured by using a Coriolis effect flow meter with an uncertainty of ± 0.5%. The volumetric flow rate of the injected oil was set by the syringe pump with an uncertainty of ± 1%. The volume flow rate of water circulating the test section was measured by a turbine flow meter that has a range of 0 – 40 g/s. The uncertainty of it is ± 0.25% of reading value. A differential pressure transducer was installed across the test section to measure differential pressures between the inlet and outlet of the test section. The range of it is between 0 and 207 kPa. The accuracy of the differential pressure transducer is ± 0.2% of the full scale. The measurements were made when the desired test conditions were reached and operating conditions were reached in steady state conditions. In the tests, the mass flux was varied from 200 to 400 kg/m^{2}s, the heat flux was 20 and 25 kW/m^2, and the test section inlet pressures were changed from 8.4 to 10.4 MPa. The test section inlet temperatures were controlled from 40 to 80°C. The oil used in the tests was PAG with a viscosity of 43 cSt at 40ºC and 9.2 cSt at 100ºC. The OMF of the test section was varied in 0, 1, 2, 3, and 4 wt.%. The OMF is defined as the ratio between the oil mass flow rate and the refrigerant/oil mixture mass flow rate. The test section inlet pressure of CO_2 was controlled by adjusting the charge amount of CO_2 in the system and by adjusting the opening of the expansion valve. The test section inlet temperature of CO_2 was controlled by adjusting the opening of two valves at inlet and outlet of the gas cooler. The heat transfer rate between water and CO_2 was set by regulating the test section inlet water temperature. For each test, the heat balance between a water-side and a CO_2-side heat transfer rate was checked and it was within 5% difference. The average uncertainty of heat transfer coefficients was 6.3%.

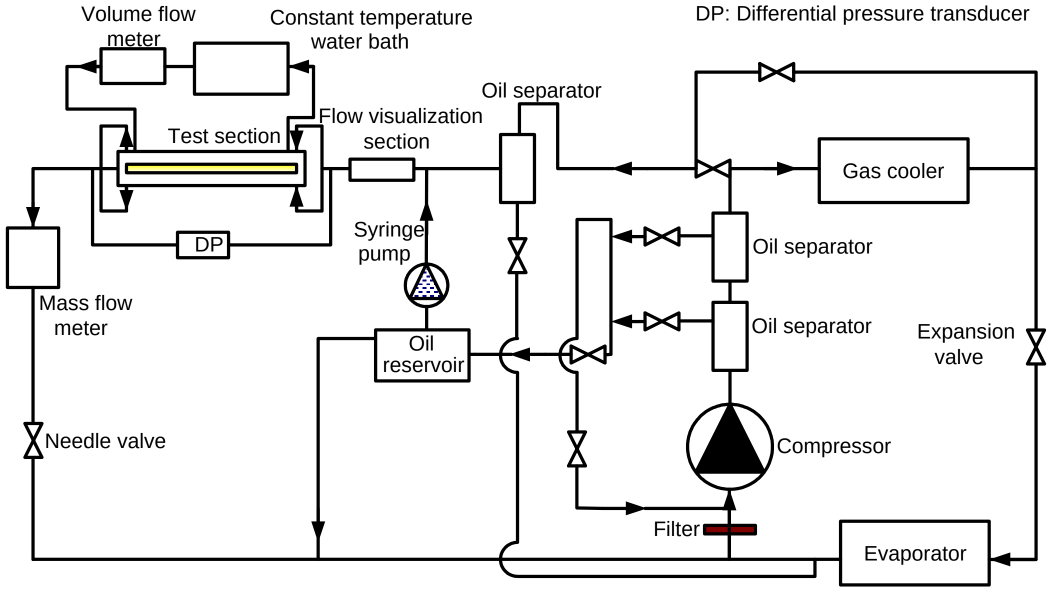

Figure 1. Schematic of Test Set-Up.

The water side heat transfer rate was calculated by using the eq. (2). The CO_2/oil side heat transfer rate was calculated by eq. (3), which includes the heat transfer rate of oil.

$$OMF = \frac{\dot{m}_{oil}}{\dot{m}_{co_2} + \dot{m}_{oil}} \times 100 \tag{1}$$

$$\dot{q}_{water} = \dot{m}_{water} c_{p,water} (T_{outlet} - T_{inlet})_{water} \tag{2}$$

$$\dot{q}_{co_2/oil} = \dot{m}_{co2} (h_{inlet} - h_{outlet}) + \dot{m}_{oil} c_{p,oil} (T_{inlet} - T_{outlet}) \tag{3}$$

The CO_2-side heat transfer coefficient was calculated by eq. (4) after the water-side heat transfer coefficient was calibrated by using the modified Wilson plot method (Briggs and Young, 1969). During the modified Wilson Plot evaluation, CO_2 was maintained at supercritical state with a pressure of 9.6 MPa and an inlet temperature of 54°C. Since the accuracy of this calibration can be improved with a higher heat transfer coefficient on the CO_2 side, a higher mass flux of 980 kg/m^{2}s was chosen. The calibration was made with 16 different water mass flow rates from 4.5 to 30 g/s.

$$\frac{1}{(hA)_{co_2}} = \frac{1}{(hA)_{overall}} - \frac{1}{(hA)_{water}} \tag{4}$$

3. RESULTS AND DISCUSSION

3.1. Heat Transfer Characteristics

Figure 2 shows the effects of oil on the gas cooling heat transfer coefficient. The pseudo-critical temperature is defined as the temperature at which the specific heat shows its highest value at the given pressure. When the temperature approaches the pseudo-critical temperature the specific heat increases. As a result, the temperature variation of CO_2 between the inlet and outlet of the test section decreases. When the pressure is 9.5 MPa its corresponding pseudo-critical temperature is 43°C. As can be seen from Figure 2, the highest heat transfer coefficient occurs around the pseudo-critical temperature not only the pure CO_2 case but also for 2 and 4 wt.% OMF cases. Moreover, the highest heat transfer coefficient decreases as the OMF increases. When the OMF is 2 wt.%, the average heat transfer coefficient decreases by 9.6% as compared with that of pure CO_2. For 4 wt.%

OMF, the degradation is 20.4% when it is compared with that of pure CO_2. It is also observed that the degradation of heat transfer coefficient is much significant around the pseudo-critical region than far region from that. This large degradation of heat transfer coefficient around the pseudo-critical region was also observed in Kuang et al.'s results (2004b).

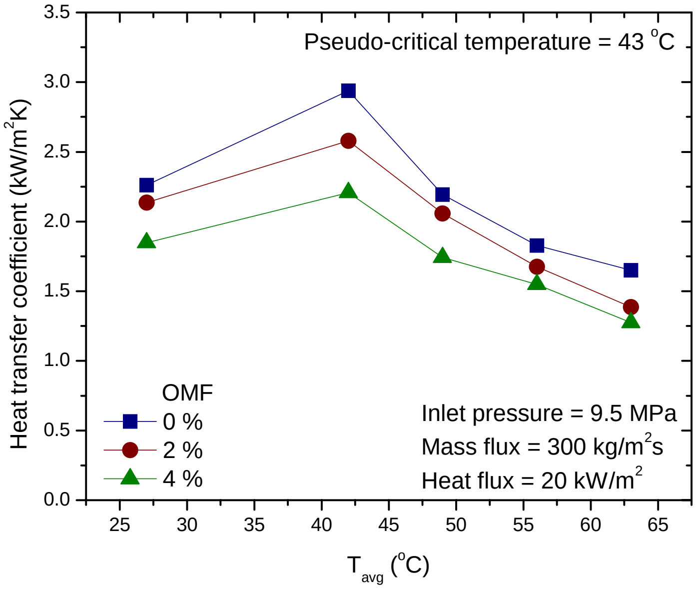

Figure 2. Effects of OMF on Gas Cooling Heat Transfer Coefficient.

Figure 3 shows the effects of the mass flux on the heat transfer coefficient at the same average temperature of CO_2 between inlet and outlet of the test section. It is crucial to set the same average temperature of CO_2 at each test in order to investigate the effects of mass flux on the heat transfer coefficient, because the thermophysical properties of CO_2 are critically dependent upon the temperature variation. Figure 3 clearly shows the trend that the heat transfer coefficient increases at higher mass flux condition under the same average temperature condition. On the other hand, it is also observed that the degradation ratio of the heat transfer coefficient with the OMF increases under the higher mass flux condition.

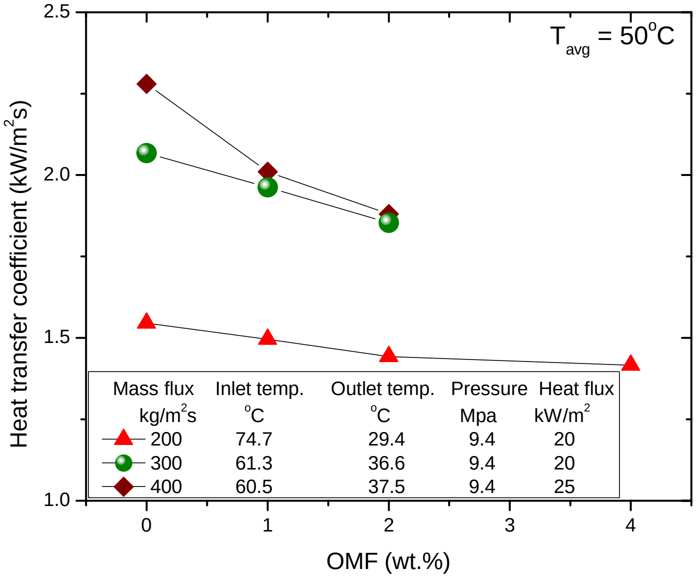

Figure 3. Effects of Mass Flux on Gas Cooling Heat Transfer Coefficient.

3.2. Pressure Drop Characteristics

Figure 4 shows the effects of the OMF on the pressure drop across the test section with the variation of average CO_2 temperature. It is shown that the pressure drop increases with the increase of average CO_2 temperature. The density of CO_2 decreases as the average temperature increases. This decrease of the CO_2 density results in the increased velocity of CO_2 inside the microchannel. Then, increase of CO_2 velocity results in the higher pressure drop. When the effects of the OMF on the pressure drop are considered, the pressure drop significantly increases with the OMF. The average pressure drop at 2 wt.% OMF is 2.9 times higher than that of pure CO_2 case. When the OMF is 4 wt.%, the average pressure drop increases by 4.8 times. Figure 5 shows the effects of the mass flux on the pressure drop with the OMF variation. When the mass flux increases from 300 to 400 kg/m^2s, the average pressure drop increases by 2.9 times.

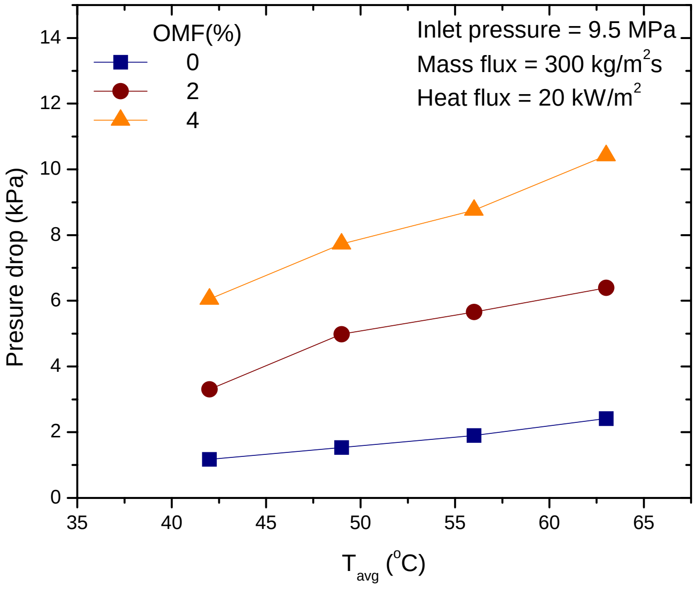

Figure 4. Effects of OMF on Gas Cooling Pressure Drop.

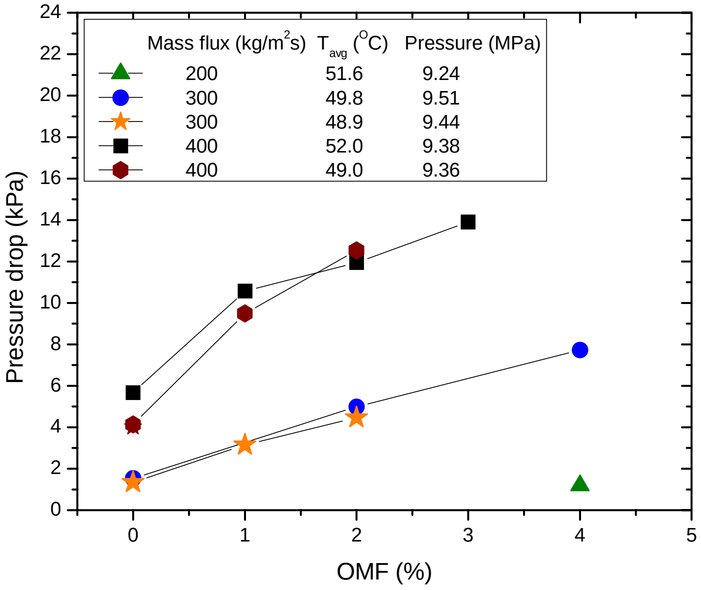

Figure 5. Effects of OMF on Gas Cooling Pressure Drop.

4. CONCLUSIONS

In this study, the effects of PAG oil on the gas cooling heat transfer coefficient and pressure drop in microchannel were experimentally investigated. Significant degradation of average gas cooling heat transfer coefficients were observed by up to 20.4% when the OMF is increased from 0 wt.% to 4 wt.%. The degradation ratio of the heat transfer coefficient increases with increase of mass flux of CO_2 at the same OMF. The average pressure drop increases by 2.9 times and by 4.8 times when the OMF increases from 0 wt.% to 2 wt.% and 4 wt.%, respectively. Since the influence of the oil concentration on the gas cooling heat transfer coefficient and pressure drop of the CO_2/oil mixture is significant, it is recommended to minimize the oil concentration less than 1 wt.%.

NOMENCLATURE

A	area, m^2
AN	Alkyl Naphthalene
c_p	specific heat, $Jkg^{-1}K^{-1}$
h	heat transfer coefficient, $Wm^{-2}K^{-1}$
$\dot{m}$	mass flow rate, kg/s
OMF	oil mass fraction
PAG	Polyalkylenglycol
POE	Polyolester
$\dot{q}$	heat transfer rate, W
T	temperature, K

Subscripts

avg	average

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