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EFFECT OF LUBRICATING OIL ON FLOW BOILING HEAT TRANSFER OF CARBON DIOXIDE

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ABSTRACT

The flow boiling heat transfer of carbon dioxide with PAG-type lubricating oil entrained from 0% to 5% in horizontal smooth tubes was examined. Experiments were conducted using test tubes with inner diameter of 2–6 mm at mass fluxes of 360–1440 kg m^{-2} s^{-1} and heat fluxes of 4.5–36 kW m^{-2}. The saturation temperature was 15°C.

At low oil concentrations of 0.5%–1%, the heat transfer coefficient decreased to less than half that under oil-free conditions. The heat transfer coefficient did not decrease further with increasing oil concentration up to 5%. The heat flux positively influenced the heat transfer coefficient in low vapor quality regions, not the high vapor quality regions. The presence of oil caused the mass flux to significantly influence the heat transfer coefficient at a low heat flux till dryout, while no significant influence of the mass flux at a high heat flux was observed. The dryout quality decreased at a large mass flux. The measured pressure drops increased monotonously because of the lubricating oil.

Keywords: Carbon dioxide; heat transfer coefficient; pressure drop; lubricating oil; flow boiling; dryout

NOMENCLATURE

d	inner diameter (m)
ΔP	pressure drop (kPa)
G	mass flux (kg m^{-2} s^{-1})
m	mass flow (kg s^{-1})
P	pressure (MPa)
q_w	heat flux (W m^{-2})
Q	heat added to test tube (W)
T_{sat}	saturation temperature (°C)
T_w	wall temperature (°C)
x	vapor quality
ω	oil concentration (wt%)
α	heat transfer coefficient (W m^{-2} K^{-1})
α_{lo}	single phase heat transfer coefficient with all flow as liquid (W m^{-2} K^{-1})

1. INTRODUCTION

Previous studies on the flow boiling heat transfer of CO_2 revealed that the characteristics of CO_2 refrigerant differed from those of other refrigerants due mainly to the former's high operation pressure and, consequently, unique thermophysical properties (e.g., Hihara and Tanaka, 2000; Pettersen et al., 2000; Sun and Groll, 2002a, 2002b, 2002c). Hihara and Dang (2007) systematically investigated the flow boiling heat transfer of CO_2 using tube diameters of 1–6 mm under various saturation temperature, mass flux, and heat flux conditions. Their experimental data revealed that nucleate boiling is predominant in CO_2 flow boiling due to its low liquid-to-vapor density ratio and low surface tension. Ducoulombier et al. (2008) established a heat transfer database for the flow boiling of pure CO_2 based on recent literature. They discussed the effects of the mass flow rate, heat flux, and tube inner diameter on the heat transfer of carbon dioxide by reviewing the results of different studies, including 216 test conditions, and concluded that some phenomena were not clarified, such as the decrease in the heat transfer coefficient with an increase in vapor quality. Schael and Kind (2005) and Dang et al. (2010b) also conducted experiments on the heat transfer coefficient of pure CO_2 flowing inside a micro-fin tube.

In an actual heat pump cycle, lubricating oil is essential to the compressor for cooling, sealing, and lubrication purposes. However, when the lubricant flows into other components, it usually causes some unexpected negative effects. Hwang et al. (2007) experimentally measured oil retention in the suction line, evaporator, and gas cooler at various refrigerant mass flow rates and oil concentrations to investigate the distribution of PAG-type oil in a CO_2 transcritical system. They found that about 50% of the total oil is retained in the heat exchangers and the suction line at an oil circulation ratio of 5 wt%. The retention of oil in the heat exchangers may result in several problems, such as an increase in the pressure drop, a decrease in the heat transfer performance, and lower system reliability. Shen and Groll (2008) reviewed the effect of lubricant on the heat transfer and pressure drop by referring to 142 papers published from 1987 to 2002; these papers examined more than ten types of refrigerants. Thome's study (1998) was discussed in this review, particularly his list of the nine effects of lubricant on refrigerant flow boiling. The five negative effects are oil viscosity effect, mass transfer resistance, oil holdup, transition from turbulent flow to laminar flow, and miscibility effect. The four positive effects are nucleate boiling, foaming, increase in the wetted surface, and flow pattern of the refrigerant–oil mixture.

Bandarra et al. (2009) addressed experimental studies on the flow boiling heat transfer and flow pattern of refrigerant/oil mixtures, including R134a/oil, ammonia/oil, R407C/oil, and CO_2/oil. They suggested that the physical mechanisms of the oil on the flow boiling of refrigerant/lubricant oil mixtures should be further investigated because there was no agreement on the influence of oil on the flow boiling heat transfer based on present results. Katsuda et al. (2003) measured the flow boiling heat transfer coefficient of CO_2 inside a 3 mm I.D. tube at heat fluxes of 5–15 kW m^{-2} and mass fluxes of 200–600 kg m^{-2} s^{-1}. They reported that the heat transfer coefficient decreased sharply at an oil concentration of 0.3%. Zhao and Bansal (2009) addressed recent studies on the heat transfer of mixtures of CO_2 and oil. They summarized the results of these papers (Hassan, 2004; Katsuta et al., 2002, 2003; Tanaka, 2001; Gao and Honda, 2006; Gao et al., 2007; Dang et al., 2006; Zhao et al., 2002) and tried to use the properties of CO_2 and oil mixtures to predict the heat transfer coefficient of CO_2 with oil through some available correlations.

Although only limited information is available, the aforementioned studies indicate that the presence of lubricating oil in subcritical CO_2 significantly influences the heat transfer performance. This implies that it is important to systematically investigate the effect of lubricating oil over a wide range of experimental conditions. This report aims to present a systematic study of flow boiling heat transfer over a wide range of experimental conditions. Experiments were conducted using horizontal tubes with inner diameters of 2, 4, and 6 mm at oil concentrations of 0.5%–5%, heat fluxes of 4.5–36 kW m^{-2}, and mass fluxes of 360–1440 kg m^{-2} s^{-1}. The heat transfer coefficient in both the pre- and post-dryout regions and the dryout quality were measured. This report presents only experimental results because it was difficult to correlate them over a wide range of experimental conditions; this correlation effort is currently in progress.

2. EXPERIMENTAL APPARATUS AND DATA REDUCTION

2.1 Experimental apparatus

Figure 1 shows a schematic diagram of the experimental apparatus used in this study. The test loop was comprised of a magnetic gear pump for working fluid circulation, mass flow meter (±0.5% accuracy), heater, oil sampling section, test section, oil separator, and cooler for cooling the refrigerant before it returns to the pump. The vapor quality at the inlet of the test section was controlled by the heater. The oil concentration was measured in the oil sampling section and controlled by the oil separator.

Figure 2 shows the detailed structure of the test section. The test section was made of stainless steel and was heated electrically using a DC power source. The electrical resistance of the test tube was approximately 0.6 Ω. The outer wall temperature of the test section was measured using T-type thermocouples with an uncertainty of ±0.1°C; these thermocouples were attached to the top, bottom, and sides of the test tube. To avoid the influence of the electric current, the tips of all thermocouples were electrically insulated from the test tube by a Teflon film of 0.05 mm thickness. The pressure of the working fluid at the inlet and outlet of the test section was monitored by using a precision pressure sensor with an uncertainty of ±0.1% F.S.

The specific enthalpy of the working fluid entering the test section was calculated from the measured temperature and pressure at the inlet of the preheater, and the power input to the heater. The thermophysical properties of CO_2 were determined using REFPROP 7.0 (2003). The inner wall temperature T_w was calculated from the measured outer wall temperature using an equation for one-dimensional heat conduction. The local saturation temperature T_{sat} was determined from the corresponding saturation pressure, which was interpolated from the measured pressures at the inlet and outlet. The local heat transfer coefficient α was estimated from the heat flux q_w, T_{sat}, and T_w.

$$\alpha = \frac{q_w}{T_w - T_{sat}} \tag{1}$$

It is essential to reduce the heat loss from the test tube to its surroundings in order to obtain correct measurements of q_w and T_w. The test tube was heavily insulated and placed inside a wind tunnel; the air temperature in the tunnel was maintained at a value close to the outer wall temperature of the test tube. The electrical input power was measured using a voltmeter and ammeter to confirm that the heat generated by direct electrification was transferred effectively well to the fluid. A preliminary test using water showed that the heat loss to surroundings was within 3% and the maximum deviation of measured heat transfer coefficient to the calculated results by Gnielinski equation was less than 10%.

The oil concentration ω was determined by the sampling method described in previous studies (Dang et al., 2007, 2010a). A sampling tube with an I.D. of 4.35 mm and length of 1.5 m was placed ahead of the test section. The overall weight of the sampled CO_2–oil mixture was measured; the CO_2 inside the sampling tube was then carefully released to obtain the weight of the retained oil. ω was measured thrice for each experimental condition, and the difference in the measured ω values was confirmed to be less than 0.1%; this was considered as the uncertainty of oil concentration.

2.2 Experimental conditions

The experimental conditions are summarized in Table 1. The inner diameters of the smooth tubes were 2, 4, and 6 mm. PAG-type lubricating oil, which is partially miscible with the liquid CO_2, was used at concentrations of 0.5–5.0 wt%; the kinematic viscosity of the oil was 105 $mm^2\,s^{-1}$ at 40°C and 20 $mm^2\,s^{-1}$ at 100°C. Experiments were conducted by varying the heat and mass fluxes in ranges of 4.5–36 kWm^{-2} and 360–1440 $kg\,m^{-2}s^{-1}$, respectively.

2.3 Experimental uncertainty

The uncertainty of the experimental results was calculated by means of the following equation:

$$\delta R = \left[\sum_{n=1}^{k} \left(\frac{\partial R}{\partial y_n} \delta y_n \right)^2 \right]^{1/2} \tag{5}$$

where δR is the total uncertainty associated with the dependent variable R, y is the independent variable which affects the dependent variable R, δy is the uncertainty of y. The relative uncertainty of α can be calculated as follows.

$$\frac{\delta\alpha}{\alpha} = \sqrt{\left(\frac{\delta q_w}{q_w}\right)^2 + \left(\frac{\delta T_w}{T_w - T_{sat}}\right)^2 + \left(\frac{\delta T_{sat}}{T_w - T_{sat}}\right)^2} \tag{6}$$

The uncertainty for q_w and T_w follows the same method by using the equation (5). The total measurement uncertainty varies with the operating conditions but mainly depends on the accuracy of the wall superheat, inner wall temperature and the saturation temperature. The uncertainty of inner wall temperature is determined from the outer wall temperature T_w, and is approximately equal to ±0.1^{o}C in spite of effects of q_w and Q. The uncertainties of q_w and Q is less than 3% due to the heat loss to environment. The uncertainty in the saturation temperature is related to the uncertainty in the equation of state as well as the accuracy of the local pressure. Measurement uncertainty of oil concentration is estimated to be 0.1wt%. Uncertainties of main measurement devices are shown in Table 3. According to uncertainty analysis, the maximum measurement uncertainty of the heat transfer coefficient was between 8.9% and 13%.

3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Effect of oil on heat transfer coefficient

The presence of lubricating oil may enhance or deteriorate the heat transfer coefficient according to the miscibility of oil under various refrigerant and flow conditions. For a miscible oil and refrigerant mixture, i.e., HFO1234yf + PAG oil (Saitoh, et al, 2012), usually the heat transfer coefficient is enhanced in the low vapor quality region, which is attributed to foaming and the increase in wetted surface due to increased surface tension, but the heat transfer coefficient in the high vapor quality region is found decreasing monotonically with oil concentration. However, when the oil is immiscible with a refrigerant, normally the heat transfer coefficient decreases with the increase of oil concentration at the whole vapor quality region. Figures 3 and 4 compare the measured heat transfer coefficients with and without the presence of oil at oil concentrations of 0%–5% inside the 2, 4 and 6 mm tubes, respectively. Since the partially miscible PAG-type oil is tested, in general, the presence of a small amount of lubricating oil resulted in a dramatic decrease in the heat transfer coefficient. Figure 3 compares the heat transfer coefficients of the 2 mm tube at oil concentrations of 0%, 0.5%, 1%, and 5% at a heat flux of 9 kW m^{-2} and mass flux of 360 kg m^{-2}s^{-1}. In comparison with the average value of 8–9 kW m^{-2}K^{-1} in the pre-dryout region under the oil-free condition, the heat transfer coefficient decreased to 3–5 kW m^{-2}K^{-1} at an oil concentration of 0.5%. However, there were only slight differences in the measured heat transfer values for oil concentrations of 0.5%–5%. It appears that the effect of oil on the heat transfer coefficient is saturated at a critical oil concentration, which was 0.5% for the 2 mm tube. Previous studies also confirmed similar saturation phenomena for microfin tubes with an average diameter of 2 mm (Dang et. al., 2010b). Figure 4 shows similar results for the 4 mm and 6 mm tubes at a heat flux of 18 kW m^{-2} and mass flux of 720 kg m^{-2} s^{-1}. As for the 4 mm tube, in the presence of lubricating oil, the heat transfer coefficient in the pre-dryout region decreased from 9–10 kW $m^{-2}K^{-1}$ to approximately 4 kW $m^{-2}K^{-1}$. The critical oil concentration at which the heat transfer coefficient significantly decreased was approximately 1%. Similar results can be found for 6 mm tube as shown in Fig.4 (b). In addition, it is seen from Figs. 3 and 4 that the dryout quality and post-dryout heat transfer were not influenced by the presence of oil up to concentrations of 5% for both the 2, 4 and 6 mm tubes.

3.2 Effect of heat flux on heat transfer coefficient

Figures 5 and 6 show the effect of the heat flux on the heat transfer coefficient for pure CO_2 and at an oil concentration of 1% inside the 2 mm tube at mass fluxes of 360 and 1440 kg m^{-2} s^{-1}, respectively. In general, for pure CO_2, the heat transfer coefficient is proportional to heat flux. With the presence of oil, the heat flux positively influences the heat transfer coefficient in the low-quality region at small mass fluxes. However, in the high-quality region or at a large mass flux, the heat flux was not observed to have an obvious influence on the heat transfer coefficient.

Because CO_2 has a low liquid-to-vapor density ratio, nucleate boiling is considered to be the dominant mechanism in CO_2 flow boiling. As shown in Fig.5 (a), the heat transfer coefficient remains constant with the increase of quality in the pre-dryout region and the heat transfer coefficient at a higher heat flux is much higher than that at a lower heat flux. However, the presence of lubricating oil in CO_2 results in the suppression to some extent of nucleate boiling depending on the mass flux. Figure 5 (b) shows the measured heat transfer coefficient at heat fluxes of 4.5–36 kW m^{-2} and a small mass flux of 360 kg $m^{-2}s^{-1}$. The heat transfer coefficient increased with the quality in the pre-dryout region at heat fluxes of 4.5 and 9 kW m^{-2}, indicating the influence of convective heat transfer. With the generation of vapor constantly, the velocity of liquid is accelerated. That is beneficial for the increment of convective heat transfer and cause suppression for nucleate boiling. When heat flux is not so strong, the convective heat transfer is main contribution for heat transfer comparing to nucleate boiling, then the tendency of heat transfer coefficient virus vapor quality is upwards gradually. In contrast, at heat fluxes of 18 and 36 kW m^{-2}, the pre-dryout heat transfer coefficient did not change with the quality. In the low vapor quality region, as the heat flux was increased from 4.5 kWm^{-2} to 36 kWm^{-2}, the heat transfer coefficient increased by approximately 100%. However, no obvious influence was observed in the high vapor quality region. This implies that nucleate boiling dominates in the low vapor quality region, while it is suppressed in the high-quality region. Moreover, both the dryout quality and post-dryout heat transfer coefficient did not change with the heat flux.

Figure 6 shows the effect of the heat flux at a large mass flux of 1440 kg m^{-2} s^{-1}. For pure CO_2, it is found that the heat transfer coefficient in the pre-dryout region is mainly dependent on the heat flux and independent of the mass flux. With the presence of oil, the heat transfer coefficient in both the pre- and post-dryout regions and the dryout quality were not influenced by the heat flux. The dryout quality for pure CO_2 and with the presence of oil took a small value of 0.4 when the mass flux was 1440 kg m^{-2} s^{-1}. In the post-dryout region, the heat transfer coefficient increased with the quality.

3.3 Effect of mass flux on heat transfer coefficient

Figure 7 shows the measured heat transfer coefficients of the 2 mm tube at mass fluxes of 360–1440 kg m^{-2} s^{-1} and a heat flux of 18 kWm^{-2} for the pure CO_2 and with the presence of oil. Similar results at a higher heat flux of 36 kW m^{-2} are shown in Fig. 8. Under the oil-free condition, as shown in Fig. 7(a) and Fig.8 (a), the heat transfer coefficient at a large mass flux was slightly less than that at a small mass flux, and dryout occurred at a low quality for a large mass flux. In the presence of lubricating oil, the heat transfer coefficient increased significantly with the mass flux in the pre-dryout region at a low heat flux. This increase was attributed to the influence of convective boiling because nucleate boiling was suppressed by the oil. However, at a higher heat flux of 36 kW m^{-2}, no obvious difference was observed in the heat transfer coefficients at different mass fluxes in the pre-dryout region, as shown in Fig. 8(b). Dryout occurred early at the high mass flux and was not influenced by the heat flux.

The tendency of the heat transfer coefficient to vary with the quality in the post-dryout region differed according to the mass flux. For both the pure CO_2 and with the presence of oil, at a large mass flux, the heat transfer coefficient increased with the quality, while at a low mass flux, the reverse was observed. As explained by Hihara et al. (2008), this decrease in the heat transfer coefficient at a low mass flux is related to a thermodynamic non-equilibrium phenomenon, in which the vapor becomes superheated in order to evaporate liquid droplets suspended in the bulk vapor phase. With an increase in mass flux, the vapor-to-droplet heat transfer coefficient increases, which leads to the alleviation of the thermodynamic non-equilibrium and to an increase in the heat transfer coefficient with the vapor quality.

3.4 Effect of tube diameter on heat transfer coefficient

Figure 9 shows the measured pre-dryout heat transfer coefficients for pure CO_2 and with the presence of oil inside tube inner diameters of 2–6 mm (1-6 mm for pure CO_2) at heat fluxes from 4.5 to 36 kW m^{-2} and mass flux of 360 kg m^{-2} s^{-1}. The influence of tube diameters for pure CO_2 and the CO_2-oil mixture are similar. For all tube diameters, the heat transfer coefficient increased with the heat flux. This tendency was more significant for small tubes. Due to the small viscosity of CO_2, the flow is turbulent flow even for 1 mm tube. Normally the two-phase heat transfer coefficient is a superposition of convective heat transfer and nucleate boiling. The single phase convective heat transfer coefficient α_{lo} is proportional to $d^{-0.2}$, with the decrease in tube diameter, the convective heat transfer is enhanced. Although the nucleate boiling heat transfer coefficient is not explicitly related to tube diameter, the ratio of amount of heat added to mass flow ($Q/m = q_w/G*4/d$) is inversely proportional to tube diameter when compared at the same q_w and G, which implies with the decrease in tube diameter, the superheat increase, thus promotes the nucleate boiling.

As shown in Fig. 9(a), for pure CO_2, the heat transfer coefficient increases with the decrease in tube diameter, and this tendency becomes more distinct at higher heat flux. The heat transfer coefficient for 6 mm tube does not change with heat flux because the flow pattern in 6 mm tube at a low mass flux of 360 kg $m^{-2}s^{-1}$ is stratified flow, the local heat transfer at the top of the tube is much lower than bottom side (Hihara and Dang, 2007). At a heat flux of 36 kW m^{-2}, the heat transfer coefficient takes value of 5.2, 12, 16 and 20 kW $m^{-2}K^{-1}$ for 6, 4, 2 and 1 mm tubes, respectively. With the presence of oil, as shown in Fig.9(b), due to the suppression of nucleate boiling by the oil, the increase tendency of heat transfer coefficient against tube diameter becomes moderate. At a heat flux of 36 kW m^{-2}, the heat transfer coefficient increased by 15% when the tube diameter was decreased from 4 mm to 2 mm. However, when the tube diameter was decreased from 6 mm to 2 mm, the heat transfer coefficient increased by 100%.

3.5 Effect of oil concentration on pressure drop

Figure 10 shows the effect of oil concentration on the measured pressure drop. The pressure drop presented is the total pressure drop which includes the fraction pressure drop and momentum pressure drop. The pressure drop was compared for tube diameters of 2–4 mm, a heat flux of 18 kW m^{-2}, and a mass flux of 720 kg m^{-2} s^{-1}. The pressure drop increased with the tube diameter because a longer tube is needed to evaporate the CO_2 from the saturation liquid state ($x = 0$) to the saturation vapor state ($x = 1$). With the same tube diameter, the pressure drop monotonously increased with oil concentration. At an oil concentration of 5%, the pressure drop increased by 80% of its value under the oil-free condition. This result was attributed to the formation of an oil layer along the inner wall of the tube and an increase in viscosity due to the entrainment of lubricating oil in CO_2.

4. CONCLUSIONS

The flow boiling heat transfer of CO_2 with a small amount of PAG-type lubricating oil was studied experimentally. The main conclusions are summarized as follows:

(1) The heat transfer coefficient decreases sharply at the critical oil concentration. The critical oil concentration for a 2 mm tube was approximately 0.5%, while it was approximately 1% for the 4 and 6 mm tubes. However, the heat transfer coefficient does not decrease further at higher oil concentrations.

(2) The heat flux positively influences the pre-dryout heat transfer coefficient at a small mass flux in the low vapor quality region due to the nucleate boiling effect. At a large mass flux or in the high vapor quality region, no obvious difference was observed when the heat flux was varied. Moreover, the heat flux does not influence the dryout quality or post-dryout heat transfer coefficient.

(3) At a low heat flux, the heat transfer coefficient in the pre-dryout region increases with the mass flux. However, at a large heat flux, the influence of the mass flux on the pre-dryout heat transfer coefficient is not significant. The dryout quality decreases at a large mass flux.

(4) As the tube diameter decreases, the increase in the heat transfer coefficient due to the heat flux in the pre-dryout region becomes significant.

(5) The pressure drop monotonously increases with oil concentration. This was attributed to an increase in the viscosity of the CO_2–oil mixture and the formation of an oil layer along the inner wall of the tube.

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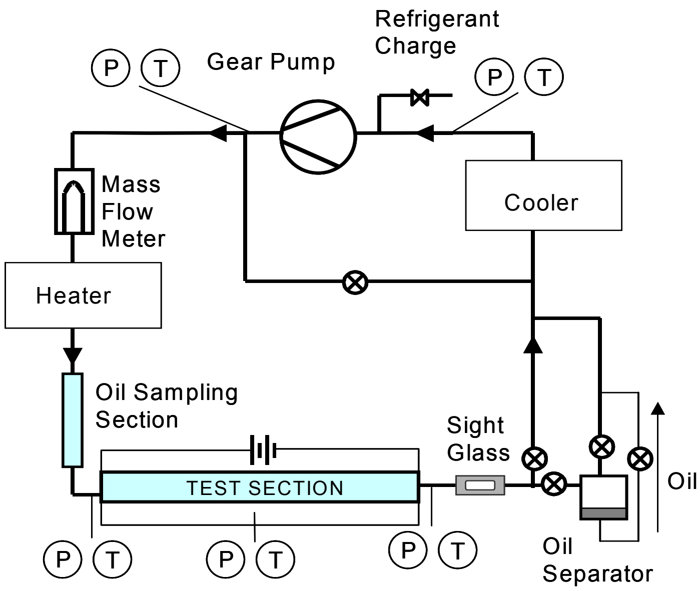


Figure 1. Schematic diagram of experimental apparatus

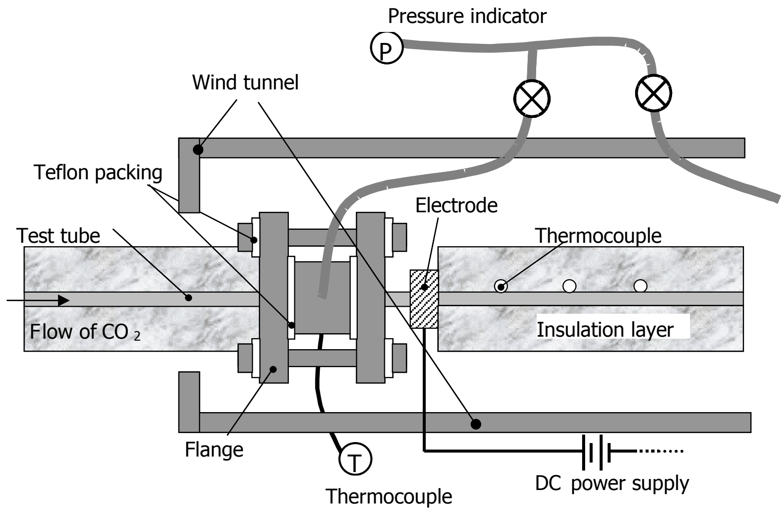


Figure 2. Detailed structure of test section

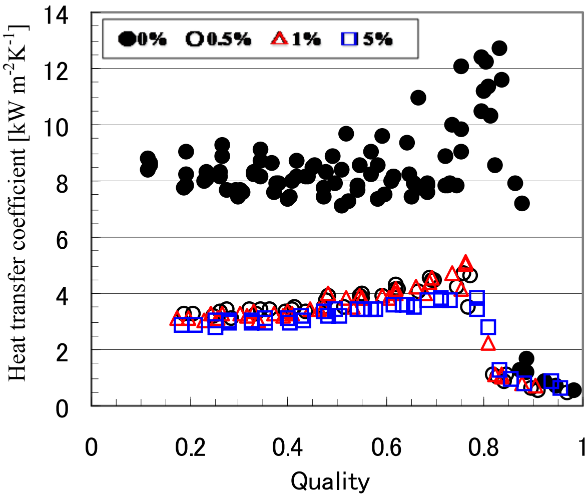
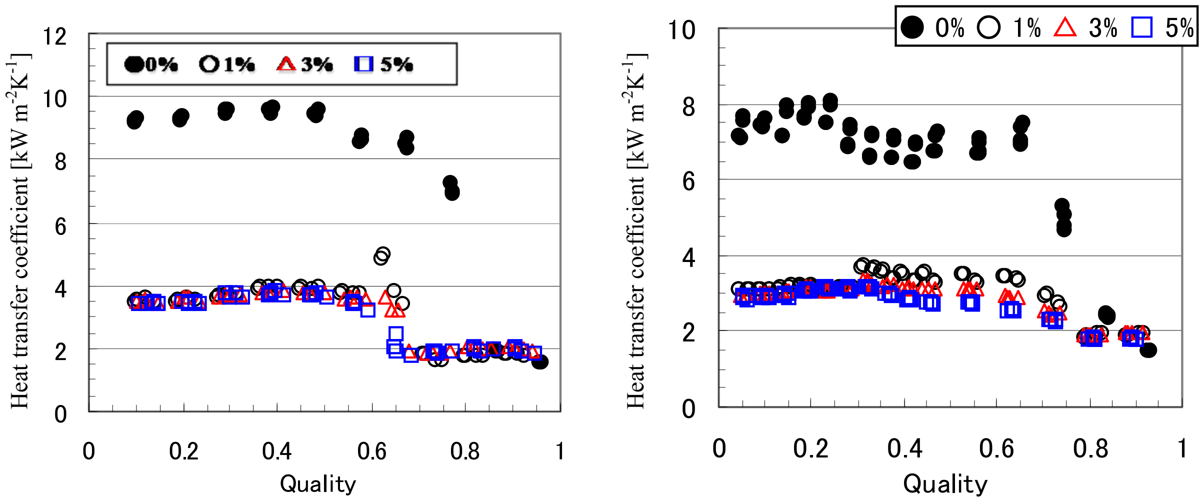
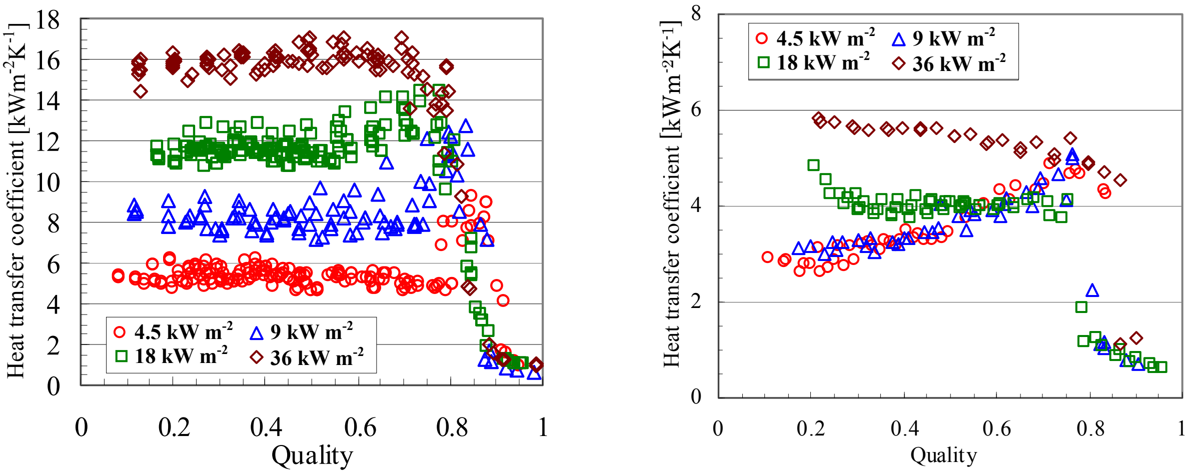


Figure 3. Comparison of heat transfer coefficient with and without the presence of lubricating oil. d = 2 mm, ω = 0%–5%. q_w = 9 kW m^{-2}, and G = 360 kg m^{-2} s^{-1}



(a) $d = 4$ mm (b) $d = 6$ mm

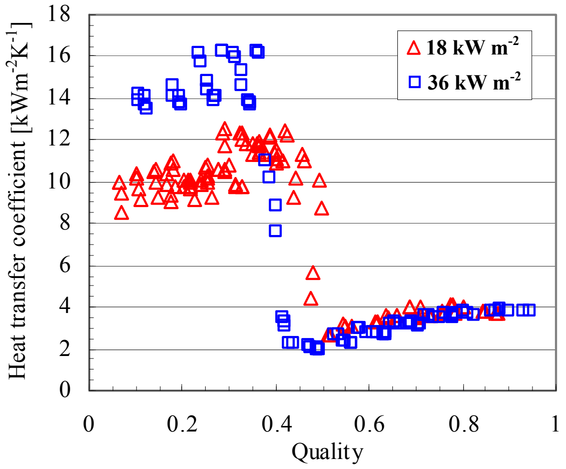
Figure 4. Comparison of heat transfer coefficient with and without the presence of lubricating oil. ω = 0%–5%, q_w = 18 kW m^{-2}, and G = 720 kg m^{-2} s^{-1}



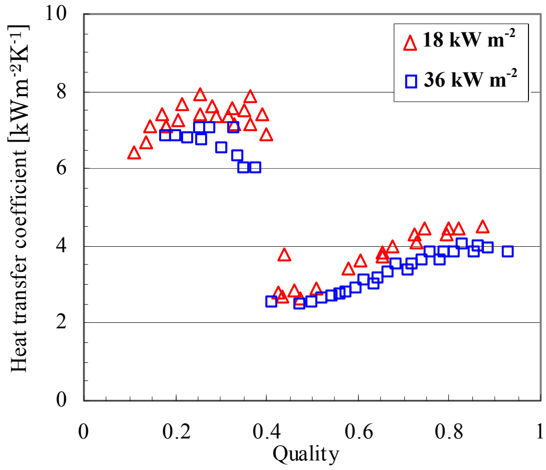
(a) Pure CO_2

(b) CO_2 with oil, ω = 1%

Figure 5. Effect of heat flux at a small mass flux of 360 kg m^{-2} s^{-1}, d = 2 mm

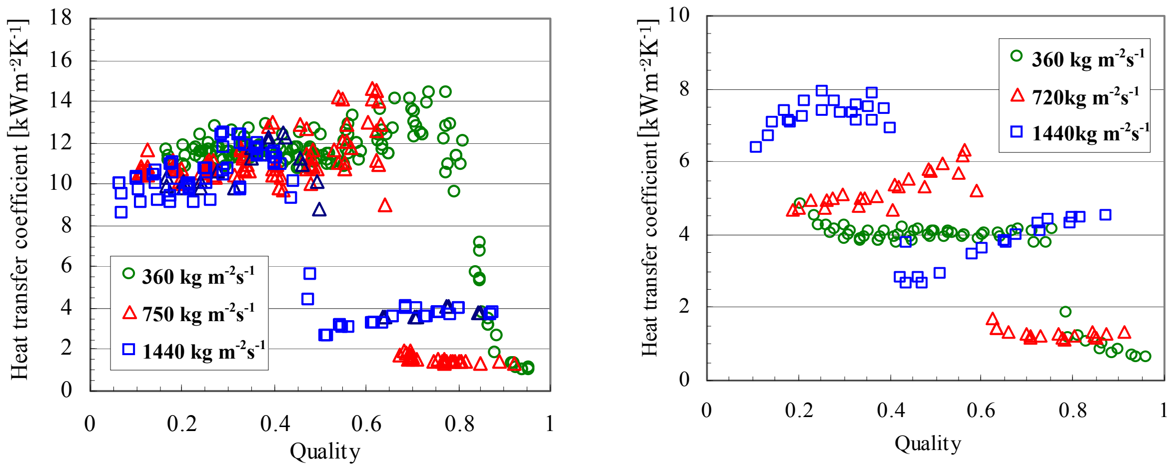


(a) Pure CO_2



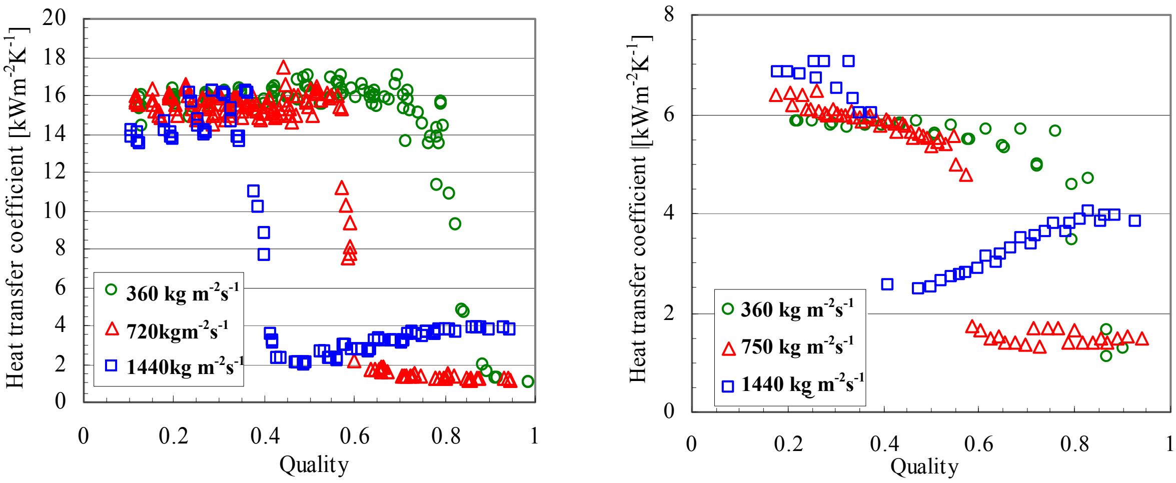
(b) CO_2 with oil, $\omega = 1\%$

Figure 6. Effect of heat flux at a large mass flux of 1440 kg m^{-2} s^{-1}. $d = 2$ mm



(a) Pure CO_2 (b) CO_2 with oil, ω = 1%

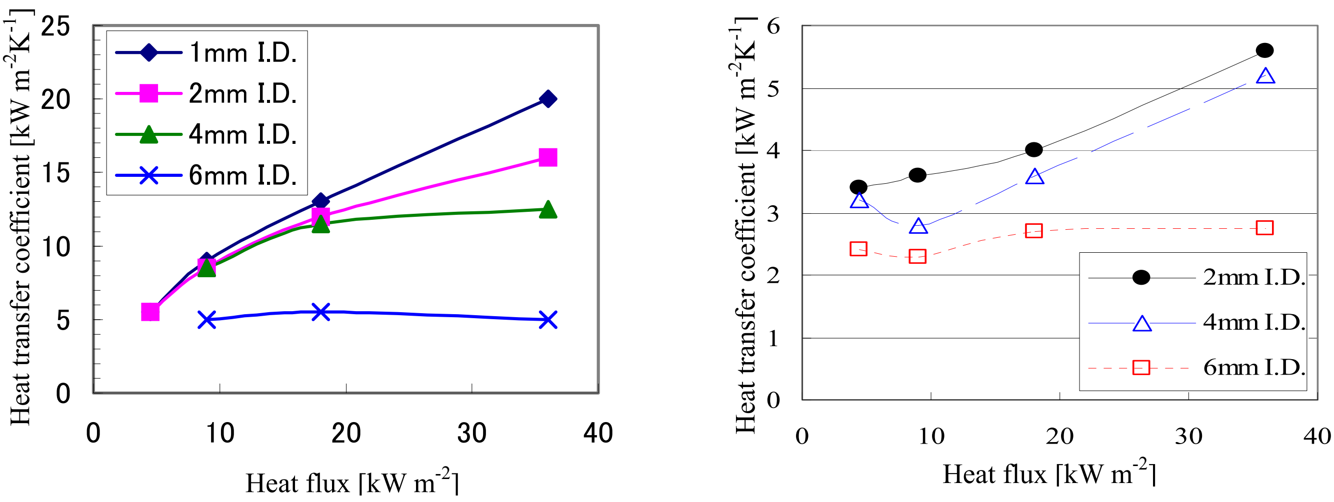
Figure 7. Effect of mass flux at a small heat flux of 18 kW m^{-2}. d = 2 mm



(a) Pure CO_2

(b) CO_2 with oil, $\omega = 1\%$

Figure 8. Effect of mass flux at a high heat flux of 36 kW m^{-2}, $d = 2$ mm



(a) Pure CO_2

(b) CO_2 with oil, ω = 1%

Figure 9. Measured heat transfer coefficients at different tube diameters and heat fluxes. G = 360 kgm^{-2} s^{-1}

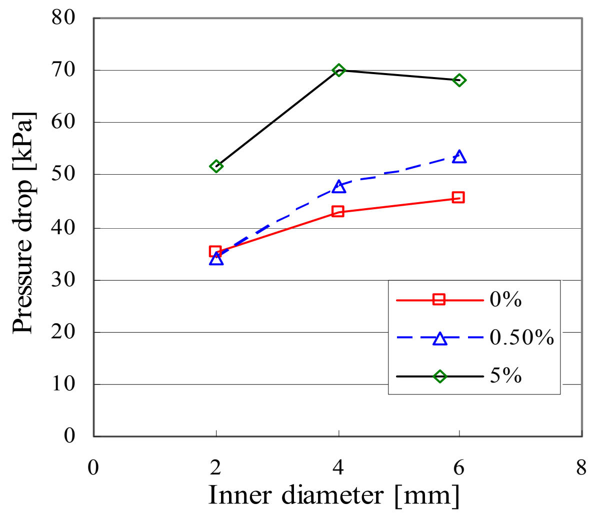


Figure 10. Comparison of measured pressure drop inside 2- 6 mm I.D. tube at different oil concentrations. q_w = 18 kW m^{-2}. and G = 720 kg m^{-2} s^{-1}

Table 1 Experimental conditions

Tube material	SUS316
Inner diameter of tube [mm]	2, 4, 6
Oil type	PAG100
Oil concentration [wt%]	0.5–5.0
Mass velocity [kg $m^{-2}\cdot s^{-1}$]	360, 720, 1440
Heat flux [kW m^{-2}]	4.5, 9, 18, 36
Evaporation temperature [°C]	15
Quality	0.1–1.0

Table 2 Uncertainty of measurement devices

Variable	Device	Range	Accuracy
Temperature of tube wall	Thermocouples	0–50 °C	±0.1 K
Temperature of refrigerant	Pt 100	-50–200 °C	±0.03 K
Mass flow rate	Coriolis type flow meter	0–150 g min^{-1}	±0.5% reading value
Pressure	pressure sensor	1–20 MPa	±0.1% full scale
Voltage	Voltmeter	0–64 V	±0.02% reading value
Inner diameter			±0.03 mm*
Length	Ruler	0–6 m	±0.001 m

* provided by the manufacture