EFFECT OF LUBRICATING OIL ON COOLING HEAT TRANSFER OF SUPERCRITICAL CARBON DIOXIDE

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ABSTRACT

In this research, the cooling heat transfer coefficient and pressure drop of supercritical CO₂ with PAG type lubricating oil entrained were experimentally investigated. The inner diameter of the test tubes ranged from 1 to 6 mm. The experiments were conducted at lubricating oil concentrations from 0% to 5%, pressures from 8 to 10 MPa, mass fluxes from 200 to 1200 kg/m²s, and heat fluxes from 12 to 24 kW/m².

In comparison to the oil-free condition, when lubricating oil entrainment occurred, the heat transfer coefficient decreased and the pressure drop increased. From visual observation it was confirmed that this deterioration in the heat transfer was due to the formation of an oil-rich layer along the inner walls of the test tubes. The maximum reduction in the heat transfer coefficients—about 75%—occurred in the vicinity of the pseudocritical temperature. When a test tube larger than 4 mm I.D. was used with a mass flux lower than 400 kg/m²s, no significant difference was observed in the heat transfer coefficient until the oil concentration reached 1%. The influence of oil was significant for a small tube diameter and a large oil concentration. However, when the oil concentration exceeded 3%, no further deterioration in the heat transfer coefficient could be confirmed; this implies that the oil flowing along with CO₂ in the bulk region does not influence the heat transfer coefficient and the pressure drop significantly.

Keyword: Carbon dioxide; heat transfer coefficient; pressure drop; cooling heat transfer, lubricating oil

1. INTRODUCTION

Conventional refrigerants have a significant ozone-depleting effect and global-warming potential. Therefore, the use of natural materials as refrigerants has attracted considerable attention. Among several possible substitute refrigerant candidates such as water, air, hydrocarbons, and ammonia, carbon dioxide is promising because it is environmentally benign, safe, and has some attractive thermodynamic characteristics such as low viscosity and high heat capacity. When carbon dioxide is used in hot water heaters and automobile air conditioners, its performance is comparable to that of HCFC refrigerants (Lorentzen & Pettersen, 1993).

Due to the low critical temperature of carbon dioxide (31.1°C), the high pressure side of the CO₂ heat pump cycle usually operates under the supercritical condition. Here, the supercritical condition is defined as the condition with a pressure higher than the critical pressure and a temperature around the critical temperature. This definition is slightly different from that used in chemical engineering, where both the pressure and temperature exceed the critical value. In this research, we particularly focus on the heat transfer characteristics of supercritical CO₂ at pressures from 8 to 12 MPa and temperatures from 20 to 80°C. A supercritical fluid exists in an intermediate condition between the liquid and vapor states. Its thermophysical properties change continuously from a liquid-like state to a vapor-like state if it is heated at constant pressure. Figure 1 shows an example of the dependence of the thermophysical properties on the temperature at a pressure of 8 MPa. All the thermophysical properties were calculated using REFPROP ver. 7.0. Since both the properties in Figure 1, specific heat \( c_p \), thermal conductivity \( k \), density \( \rho \) and viscosity \( \mu \), directly influence the heat transfer characteristics, the heat transfer characteristics of supercritical fluids differ from those of constant-property-fluids and have generated a considerable amount of research interest. Dang & Hihara investigated the cooling heat transfer characteristics of supercritical CO₂ both experimentally (2004a) and numerically (2004b). They discussed the experimental results of the heat transfer coefficient and pressure drop of supercritical CO₂ under different mass flux, heat flux, and tube diameter conditions and proposed a modification of the Gnielinski correlation (1976) to...
correlate their experimental results. By comparing the numerical calculation with the experimental result for cooling (Dang & Hihara, 2004a) and heating conditions (Tanaka et al. 1967), they claimed that the flow and heat transfer can be predicted by numerical simulation if a suitable turbulent model is used; they recommended the JL model for this purpose (Jones & Launder, 1972). However, with regard to the heat transfer in the near-critical region, the compressibility of the supercritical fluid, which is referred to as the “piston effect,” must be taken into account, and this has been discussed by Onuki et al. (1990).

In an actual heat pump cycle, lubricating oil is commonly used in the compressor for cooling and sealing purpose. Small amounts of lubricating oil may be discharged by the compressor along with refrigerant; this oil flows through heat exchangers and expansion device and finally returns to the compressor. Therefore, in effect, instead of pure refrigerant, a mixture of the refrigerant and a small amount of oil flows inside the heat exchanger and exchanges heat with the environment. It is important to understand the heat transfer performance of supercritical CO₂ with entrained lubricating oil, which has not been studied in detail thus far.

Zingerli & Groll (2000) measured the heat transfer coefficient and pressure drop of a CO₂-oil mixture in a 2.85 mm I.D. tube. POE oil, which is miscible with CO₂, was used at three oil concentrations—0%, 2%, and 5.0%. The experimental results showed a large decrease in the maximum heat transfer coefficient in the near-pseudocritical temperature region. The study also reported that heat transfer was promoted on the high temperature side.

In Japan, PAG oil, which has excellent lubricity and stability under supercritical CO₂ environment, is used in the CO₂ heat pump cycle. However, since the PAG oil is partially miscible with CO₂ at the supercritical condition, the heat transfer performance may be significantly influenced by the entrainment of lubricating oil in CO₂. Gao & Honda (2002) investigated the heat transfer of supercritical CO₂ with an oil concentration of 1% and reported a maximum decrease of 40% in its heat transfer performance when compared to pure CO₂. Moli et al. (2002) observed the flow pattern of a CO₂-oil mixture inside the gas cooler. Although the oil concentration is unknown, it was found that the oil separated from CO₂ and formed an oil layer near the inner wall. Subsequent measurements of the heat transfer coefficient revealed a decrease in heat transfer, which is believed to have occurred due to the heat resistance of the oil layer. The experimental conditions were as follows: 9.5 MPa pressure and 6 mm tube diameter.

Although only limited information is available, the above-mentioned researches indicated a significant influence of the introduction of lubricating oil into supercritical CO₂ on the heat transfer performance, which in turn implies the necessity of a systematic investigation of the influence of lubricating oil under a wide range of experimental conditions. This report shows some experimental results of the cooling heat transfer coefficient and pressure drop of supercritical CO₂ with the entrainment of a small amount of PAG type lubricating oil. The inner diameter of the test tubes ranged from 1 to 6 mm. The experiments were conducted at oil concentrations from 0% to 5%, pressures from 8 to 10 MPa, mass fluxes from 200 to 1200 kg/m²s, and heat fluxes from 12 to 24 kW/m².

Figure 1. Thermophysical properties of supercritical CO₂ near the pseudocritical point at 8 MPa

2. EXPERIMENTAL APPARATUS AND DATA REDUCTION

2.1 Experimental apparatus

In Figure 2, the schematics of the test loop and test section are shown. The test loop comprised three main components—a main loop, a test section, and an oil sampling section. The main loop was a conventional CO₂ heat pump cycle for controlling the temperature and pressure at the inlet of
the test section. An oil separator was installed to control the oil concentration.

The test section (Figure 2b) was a 0.5 m long horizontal tube-in-tube counterflow heat exchanger, with CO₂ flowing inside the inner tube and cooling water inside the annular passage. The inner and outer tubes were made of smooth copper and acrylic resin, respectively. The heat transfer coefficient and pressure drop were measured for four different inner tubes with inner diameters \( d \) of 1, 2, 4, and 6 mm. The test section was covered with 25 mm thick polystyrene insulation to minimize heat loss to the environment.

The temperatures of CO₂ at the inlet and outlet of the test section (\( T_1 \) and \( T_2 \), respectively) were measured using T-type thermocouples. The temperature of the cooling water at the inlet and outlet were measured using Pt100 sensors to determine the heat transfer rate in the test section. The wall temperature was measured at ten locations spaced equally along the tube using T-type fine thermocouples soldered to the middle portion of the outer wall. The inner wall temperatures at the ten locations were calculated from the corresponding outer wall temperatures by solving one-dimensional heat conduction equations. The thermocouples and the Pt100 sensors were calibrated to an accuracy of ±0.1°C and ±0.03°C, respectively. The pressures at the inlet and outlet of the test section (\( P_1 \) and \( P_2 \), respectively) were measured using a pressure transducer with a measurement uncertainty of ±0.001MPa. The mass flow rate of the refrigerant was measured using a Coriolis-type mass flow meter with an accuracy of 0.1% F.S. The accuracy of the water side mass flow rate was 0.5% F.S.

The oil concentration was measured at the oil sampling section. The refrigerant-oil mixture was sampled, and the concentration of the lubricating oil in the refrigerant was weighed by carefully releasing the refrigerant through a needle valve.

![Schematic of the test loop and the test section](image)

**Figure 2. Schematic of the test loop and the test section**

### 2.2 Experimental conditions and data reduction

The experimental parameters were mass flux (\( G \)), pressure (\( P \)), heat flux (\( q_w \)), tube diameter (\( d \)), and oil concentration (\( x \)), as shown in Table 1. Under each experimental condition, \( T_1 \) was varied from 30°C to 70°C. PAG type lubricating oil was used with the oil concentration changed from 0.5 to 5.0 wt%. The kinematic viscosity of PAG oil is 105 mm²/s and 20 mm²/s at 40°C and 100°C, respectively.

<table>
<thead>
<tr>
<th>Tube material</th>
<th>Copper</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube I.D. [mm]</td>
<td>1.0, 2.0, 4.0, 6.0</td>
</tr>
<tr>
<td>Oil type</td>
<td>PAG100</td>
</tr>
<tr>
<td>Oil concentration [wt%]</td>
<td>0.5~5.0</td>
</tr>
<tr>
<td>Mass velocity [kg/(m²·s)]</td>
<td>200~1200</td>
</tr>
<tr>
<td>Heat flux [kW/m²]</td>
<td>12, 24</td>
</tr>
<tr>
<td>Pressure [MPa]</td>
<td>8.0, 9.0, 10.0</td>
</tr>
</tbody>
</table>

The heat transfer coefficient (\( \alpha \)) was calculated from \( q_w \) and the average temperature difference between the refrigerant and the wall (\( \Delta T \)), while the pressure drop (\( \Delta P \)) was calculated from the pressures measured at the inlet and outlet of the test section.

\[
\alpha = \frac{q_w}{\Delta T}
\]
\[ \Delta P = P_1 - P_2 \]  

\[ q_w \text{ was calculated from the waterside heat exchange rate:} \]

\[ q_w = G_{\text{water}} c_{\text{water}} (T_{\text{water, out}} - T_{\text{water, in}}) / A \]

where \( A \) is the heat transfer area; \( c_{\text{water}} \), specific heat of water; and \( T_{\text{water, in}} \) and \( T_{\text{water, out}} \), temperatures of water at the inlet and outlet of the test section, respectively.

The heat balance was examined by comparing the heat exchange rate measured at the waterside with that measured at the CO\(_2\) side. When \( T_1 \) was not near the pseudocritical temperature \( T_m \), the imbalance was less than 10%. Near \( T_m \), the imbalance increased to a relatively high level, about 15%. This high imbalance was caused partly due to the drastic variation in \( c_p \) as \( T_1 \) approached \( T_m \) and partly due to the large uncertainty in the temperature change from the inlet to the outlet of CO\(_2\).

At supercritical pressure, since both \( c_p \) and \( \alpha \) of CO\(_2\) change significantly along the flow direction, the definition of the mean temperature difference \( \Delta T \) should be prioritized. In this study, a logarithmic mean temperature difference \( \text{LMTD}_w \) was defined to calculate the average temperature difference between CO\(_2\) and the wall as follows:

\[ \text{LMTD}_w = \frac{(T_1 - T_{w1}) - (T_2 - T_{w2})}{\ln\left(\frac{T_1 - T_{w1}}{T_2 - T_{w2}}\right)} \]

### 3 EXPERIMENTAL RESULTS AND DISCUSSIONS

#### 3.1 Flow visualization

Since PAG oil is partially miscible with CO\(_2\) at supercritical pressure, the flow pattern of the CO\(_2\)-oil mixture may vary with the oil concentration, mass flux, temperature, and pressure conditions. To understand the mechanism of heat transfer inside the gas cooler, it is important to understand the flow pattern inside it. Figure 3 shows some flow visualization results for a 2 mm I.D. tube at different temperature and oil concentration conditions. For a detailed description of the flow visualization and additional results at other conditions please refer to Dang et al. (2006).

Figure 3(a) illustrates the flow pattern in a 2 mm I.D. tube at a low temperature of 25°C and a low oil concentration of 1%. Since supercritical CO\(_2\) is transparent, the white particles show the movement of the oil droplet. It is visible that the oil separates from the CO\(_2\) and forms oil droplets with diameters of 50 to 100 µm. The droplets flow along with the bulk CO\(_2\) at a slip ratio of about 0.5 in the flow direction; simultaneously, they move up and down along the radial direction due to the disturbing velocity of the bulk CO\(_2\) flow. Under this condition, no oil film flow along the inner wall is observed.

When the bulk temperature of CO\(_2\) increases to 50°C, the solubility of CO\(_2\) decreases, leading to an increase in the viscosity of oil. As a result, the separated oil seems to adhere to the inner wall and forms an oil film, which is visible as the stripes in Figure 3(b). Since the layer is much thicker than that at a low temperature, the possible movement of the oil droplets in the bulk region cannot be observed. The layer moves at a very low speed and may lead to a decrease in the heat transfer coefficient due to its heat resistance; simultaneously, it increases the pressure drop.

At a higher oil concentration such as 5%, there is an increase in both the amount of oil droplets flowing in the bulk region and the oil layer thickness. As shown in Figure 3(c)–(d), even at a low temperature of 25°C, the oil layer can be observed clearly. These results imply a decrease in the heat transfer coefficient at both low and high temperatures for high oil concentration.

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Figure 3. Flow visualization of CO\(_2\) with small amount of entrained lubricating oil. \( d = 2 \text{ mm}, \ P = 10 \text{ MPa}, \ G = 800 \text{ kg/m}^2\text{s} \)
3.2 Heat transfer coefficient $\alpha$

The heat transfer coefficients for the four test tubes with inner diameters of 1, 2, 4, and 6 mm were measured at various oil concentrations. In comparison to the oil-free condition, on the whole, the heat transfer coefficient decreases with oil entrainment. The smaller the tube diameter, the more significant is the effect of the lubricating oil. In Table 2, the ratio of the peak value of the heat transfer coefficient for oil entrainment to that at the oil-free condition is summarized.

Table 2 Decrease ratio of heat transfer coefficient compared with oil-free condition

<table>
<thead>
<tr>
<th>ID (mm)</th>
<th>Mass Flux (kg/m²s)</th>
<th>Heat Flux (kW/m²)</th>
<th>Pressure (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>8</td>
<td>9</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>1%</td>
<td>~3%</td>
<td>1%</td>
</tr>
<tr>
<td>6</td>
<td>12</td>
<td>24</td>
<td></td>
</tr>
<tr>
<td></td>
<td>200</td>
<td>94%</td>
<td>84%</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>96%</td>
<td>89%</td>
</tr>
<tr>
<td>4</td>
<td>12</td>
<td>110%</td>
<td>80%</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>52%</td>
<td>32%</td>
</tr>
<tr>
<td>2</td>
<td>12</td>
<td>86%</td>
<td>30%</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>113%</td>
<td>40%</td>
</tr>
<tr>
<td>1</td>
<td>12</td>
<td>94%</td>
<td>35%</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>113%</td>
<td>52%</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1200</td>
<td>12</td>
<td>113%</td>
<td>52%</td>
</tr>
</tbody>
</table>

Figure 4 shows the variation in the heat transfer coefficient with the bulk temperature in the 6 mm I.D. tube. The predicted values for the oil-free condition are indicated by the solid line. The heat transfer coefficients measured at different oil concentrations are plotted, and the horizontal error bar represents the temperature change from the inlet to the outlet of the test section. In comparison to the oil-free condition, it is observed that the effect of oil concentration on the heat transfer coefficient is not linear. For the 6 mm I.D. tube, the heat transfer coefficient does not decrease considerably until the oil concentration reaches 1%. When the oil concentration exceeds 1%, a sharp decrease in the heat transfer coefficient, around 30%, is observed. This decrease in the heat transfer coefficient is due to the heat resistance of the oil layer flowing along the tube inner wall. However, an increase in the oil concentration from 3% to 13% leads to a slight decrease in the heat transfer coefficient, As shown in Figure 4, an identical tendency is observed at both 8 and 10 MPa. This tendency implies a saturation phenomenon of the heat transfer deterioration, most probably because the oil flowing along with CO$_2$ in the bulk region does not significantly influence the heat transfer coefficient.

In Figure 5, the heat transfer coefficients measured for the 4 mm I.D. tube are shown. Similar to the case of the 6 mm I.D. tube, the effect of lubricating oil entrainment on the heat transfer coefficient is quite weak at an oil concentration of 1%. The heat transfer coefficient drops significantly at an oil concentration of approximately between 1% and 3%. However, a further increase in the oil concentration from 3% to 5% does not significantly influence the heat transfer coefficient. In comparison to the oil-free condition, the heat transfer coefficient decreases to as much as 50% for the 4 mm tube, which is greater than that for the 6 mm tube, thus indicating the effect of the tube diameter. It appears that with a decrease in the tube diameter, the oil that subsequently forms the film layer along the inner wall decreases the heat transfer more significantly.

In Figure 6, the heat transfer coefficients measured in the 1 and 2 mm I.D. tubes at different experimental conditions are shown. For a small-sized tube, since the oil layer easily forms near the wall, the addition of a small amount of lubricating oil leads to a significant reduction in the heat transfer coefficient. At an oil concentration of 1%, the heat transfer coefficient decreases to about 60% and 50% of the values at the oil-free condition in the 2 and 1 mm tubes, respectively. For the 2 mm I.D. tube, a further increase in the oil concentration from 1% to 3% leads to a decrease in the heat transfer coefficient of about 40%; however, the heat transfer coefficient at 5% is just slightly lower than that at 3%. On the contrary, for the 1 mm I.D. tube, the heat transfer coefficient decreases linearly with an increase in the oil concentration from 1% to 5%.

3.3 Pressure drop $\Delta P$

The pressure drops measured for 1 and 2 mm I.D. tubes at different oil concentrations are shown in Figure 7. The experimental data is plotted at the average temperature of each run, and the solid line indicates the pressure drop at the oil-free condition, which, according to Dang & Hihara (2004a), can be predicted by Filonenko’s equation as follows:

$\text{压力下降} (\Delta P) = \text{阻力系数} \times \text{流量} \times \text{温度差}$
Figure 4. Heat transfer coefficient measured in the 6 mm I.D. test tube at different oil concentrations

Figure 5. Heat transfer coefficient measured in the 4 mm I.D. test tube at different oil concentrations

Figure 6. Heat transfer coefficient measured in the small sized tube at different oil concentrations

\[ f = \left[ 1.82 \log_{10}(\text{Re}) - 1.64 \right]^2, \quad \Delta P = f \frac{L}{d} \frac{1}{2} \rho u^2 \]

where subscript \( b \) denotes the thermophysical properties at the bulk temperature.

The effect of lubricating oil on the pressure drop is not linear, similar to its effect on the heat transfer. In comparison to the oil-free condition, the pressure drop does not increase considerably until the oil concentration reaches 1%. Subsequently, the pressure drop increases sharply with an increase in the oil concentration. For the 2 mm I.D. tube, there is no difference between the pressure drops for the oil concentrations of 5% and 3%, indicating that the oil droplet flowing in the bulk region does not significantly influence the pressure drop. However, for the 1 mm I.D. tube, the pressure drop increases monotonously with an increase in the oil concentration. In addition, the increase in the pressure drop on the high temperature side is much more significant than that on the low temperature side, which is due to the formation of an oil film on the high temperature side, as
confirmed by the visualization observation.

The friction factor is compared in Figure 8 in order to show the effect of oil entrainment at different mass fluxes. For the oil-free condition, the friction factor decreases with an increase in temperature; on the other hand, with the entrainment of lubricating oil, the friction factor does not change with temperature, thus indicating the influence of the oil layer. In addition, the friction factor at a higher mass flux of 1200 kg/m²s is much higher than that at a lower mass flux.

![Figure 7](image1.png)

Figure 7. Measured pressure drop variation with $T_{\text{bulk}}$ at different oil concentrations

![Figure 8](image2.png)

(a) $G = 800$ kg/m²s          (b) $G = 1200$ kg/m²s

Figure 8. Measured friction factor variation with $T_{\text{bulk}}$ at different oil concentrations. $d = 1$ mm, $P = 10$ MPa, $q_w = 12$ kW/m²

4. CONCLUSIONS

The heat transfer characteristics of supercritical CO₂ cooled in horizontally placed circular tubes with a small amount of entrained PAG type lubricating oil were investigated experimentally. The oil concentrations were varied from 1~5 wt%, along with variations in the mass flux, pressure, heat flux, and tube diameter, and their effect on the heat transfer coefficient and pressure drop were analyzed. The main conclusions of this study are summarized as follows:

(1) From flow observations, it was found that at a low temperature and a low oil concentration, the oil flows in the bulk region with a diameter from 50 to 100 µm. With an increase in the oil concentration or temperature, an oil layer in the near-wall region is observed.

(2) The heat transfer coefficients decrease with the entrainment of lubricating oil in supercritical CO₂, which is due to the formation of an oil layer in the near-wall region.

(3) For a large-sized tube, instead of that with a 4 mm I.D., with a mass flux lower than 400 kg/m²s, no significant difference in the heat transfer coefficient was observed until the oil concentration reached 1%. A sharp decrease occurs at an oil concentration approximately between 1% and 3%. A further increase in the oil concentration above 3% does not decrease the heat transfer coefficient significantly.

(4) For small-sized tubes, the heat transfer coefficient drops at a rather low oil concentration. At an oil concentration of 1%, the heat transfer coefficient is 60% and 50% of that at the oil-free
condition for the 2 and 1 mm I.D tubes, respectively.

(5) In comparison to the pseudocritical temperature, the solubility of CO\textsubscript{2} in the oil is greater on the low temperature side, thereby reducing the significance of the degradation.

(6) The pressure drops for the 2 and 1 mm I.D. tubes do not increase significantly at an oil concentration of 1\%. For the 2 mm I.D. tube, there is no difference between the pressure drops for the oil concentrations of 5\% and 3\%. However, for the 1 mm I.D. tube, the pressure drop increases monotonously with an increase in the oil concentration.

**NOMENCLATURE**

\[ A \quad \text{heat transfer area, \text{[m}^2\text{]}} \]
\[ D \quad \text{diameter, \text{[m]}} \]
\[ k \quad \text{thermal conductivity, \text{[w/mK]}} \]
\[ \Delta P \quad \text{pressure drop, \text{[MPa]}} \]
\[ \Delta T \quad \text{temperature difference, \text{[C]}} \]
\[ f \quad \text{friction factor} \]
\[ G \quad \text{mass flux, \text{[kg/m^2s]}} \]
\[ \log_{\text{MDT}} \quad \text{logarithmic mean temperature difference between refrigerant and wall, \text{[C]}} \]
\[ Nu \quad \text{Nusselt number, (\alpha d/k)} \]
\[ P \quad \text{pressure, \text{[MPa]}} \]
\[ q_w \quad \text{heat flux, \text{[W/m^2]}} \]
\[ T \quad \text{temperature, \text{[C]}} \]

Greek symbols

\[ x \quad \text{oil concentration, \text{[wt\%]}} \]
\[ \mu \quad \text{dynamic viscosity, \text{[kg/ms]}} \]
\[ \rho \quad \text{density, \text{[kg/m^3]}} \]
\[ \alpha \quad \text{heat transfer coefficient, \text{[W/m^2K]}} \]

Subscripts

\[ 1 \quad \text{inlet} \]
\[ 2 \quad \text{outlet} \]
\[ b \quad \text{bulk} \]
\[ m \quad \text{pseudocritical} \]
\[ w \quad \text{wall temperature} \]
\[ \text{water waterside temperature} \]

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