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Seongho Kim
skim1010@illinois.edu

Predrag S. Hrnjak

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Effect of Oil on Flow Boiling Heat Transfer and Flow Patterns of CO$_2$
in 11.2 mm Horizontal Smooth and Enhanced Tube

Seongho KIM$^1*$, Pega HRNJAK$^{1,2}$

$^1$Air Conditioning and Refrigeration Center, Department of Mechanical Science and Engineering,
University of Illinois at Urbana-Champaign, Urbana, IL 61801, USA
(217 244 0043, 217 333 1942, skim1010@illinois.edu)

$^2$CTS - Creative Thermal Solutions,
Urbana, IL 61802, USA

* Corresponding Author

ABSTRACT

Flow boiling heat transfer characteristics of CO$_2$ with and without oil were investigated experimentally in horizontal smooth and enhanced tubes with an inner diameter of 11.2 mm. The visualization of flow pattern was also performed to provide the detailed attributes of the nucleate and the convective boiling heat transfer. In order to investigate the effect of the miscible oil on the heat transfer of CO$_2$, POE (polyolester) RENSIO C85E oil is added to give an oil circulation rate (OCR) between 0.5 % and 2 %. Results are compared with those of pure CO$_2$. The experimental conditions include evaporation temperatures of -15 °C, mass fluxes from 40 to 200 kg/m$^2$ s, heat fluxes from 0.5 to 10 kW/m$^2$, and vapor qualities from 0.1 to 0.8. Oil generally deteriorates the heat transfer coefficient of pure CO$_2$. The reduction in heat transfer coefficient is most apparent at low vapor qualities, 0.1 to 0.4, and at low mass fluxes, 100 and 200 kg/m$^2$. It is caused by the suppression of nucleate boiling due to increased surface tension. At conditions where the convective boiling contribution is dominant, vapor qualities above 0.5, oil increases heat transfer coefficients. Through visualization, it is shown that the wetted area on the perimeter of inner tube is enhanced due to formation of foaming in the smooth tube. However, such enhancement of heat transfer due to forming is negligible in the enhanced tube, because the enhanced factor due to micro-finned structures is dominant.

1. INTRODUCTION

Due to growing concerns of the ozone depletion and global warming, CO$_2$ has drawn great attention as an alternate environmentally-friendly refrigerant. Refrigeration systems using CO$_2$ as a working fluid are already commercialized on the market. Recent literature reviews by Thome and Ribatski (2005) and Zhao and Bansal (2007a) (2009a) reveal that many researchers experimentally investigated the flow boiling heat transfer of pure CO$_2$. Especially at low evaporation temperatures under -10 °C, several experiments had been conducted by Bredesen et al. (1997), Knudsen & Jensen (1997), Hassan (2004), Park & Hrnjak (2005) (2007), Zhao & Bansal (2007b) (2009b), and Kim et al. (2010). Comparing to pure CO$_2$, however, experimental studies on oil/CO$_2$ mixtures are somewhat limited. In typical refrigeration or air-conditioning systems, a small amount of the lubricating oil migrates from the compressor and through the system, which considerably affect the heat transfer performance of the components. In order to understand the effect of oil on the heat transfer characteristics, it is necessary to measure the flow boiling heat transfer coefficient of both pure CO$_2$ and oil/CO$_2$ mixtures. At low evaporation temperature of -30 ~ -10 °C, it is seen Hassan (2004) that the heat transfer coefficient decreases continuously as oil concentration increases, although positive effects of oil addition have been observed in some other studies of other types of mixtures (Bandarra et al., 2009). Furthermore, there is no information about the flow boiling heat transfer of pure CO$_2$ and oil/CO$_2$ mixtures in the enhanced tube at low evaporation temperature under -15 °C. In addition to measurement of heat transfer coefficient, it is essential to observe two-phase flow patterns through visualization in order to
understand comprehensively two-phase flow boiling heat transfer characteristics. To our best knowledge, however, there has been no study on visualization of oil/CO\textsubscript{2} mixtures, even though flow patterns have been observed for other refrigerants and oil mixtures (Bandarra et al., 2009). In this paper, we study the flow boiling heat transfer and the corresponding flow patterns of CO\textsubscript{2} with and without oil in the smooth and enhanced tubes with large diameter.

2. EXPERIMENTAL FACILITY AND TEST CONDITIONS

Figure 1 shows a schematic of the experiment facility. The test facility consists of a refrigerant loop for testing, and a secondary coolant loop with HFE 7100 for controlling the test conditions. In the test loop, liquid refrigerant is pumped by the gear pump to the calorimeter, which heats it to the quality desired at the inlet of the test section. And visualization section is located after the test section. The control heater adds heat to maintain a desired saturation temperature in the test section. The refrigerant leaves the test section and condenses in a plate heat exchanger connected to the R404A cooling unit. The single-phase, secondary HFE 7100 loop subcools the refrigerant in the subcooler and then adds heat in the test section, as shown in Fig. 1.

Polyolester (POE) oil, RENISO C85E (density at 15 °C is 1004 kg/m\textsuperscript{3} and viscosity at 40 °C is 80 mm\textsuperscript{2}/s), is used as the lubricating oil. It is completely miscible with CO\textsubscript{2} under operating evaporation temperatures and is widely used in industry. The mass of liquid CO\textsubscript{2} in the system is calculated by each test, and enough oil is added to achieve the desired oil circulation rate (OCR). The OCR is held constant during each experiment, and oil separator keeps oil from entering the fade-out vessel.

The test section consists of the test tube, brass jacket, and tube circuit for HFE as presented at Fig 1. The inside and outside diameters of the smooth tube are 11.2 and 15.8 mm, respectively. The enhanced tube, of which inner melting diameter is 11.17 mm, has 68 micro-fins along the internal periphery with helix angle of 30 \degree. The copper tube has a heated length of 150 mm which is surrounded by a two-piece, cylindrical brass jacket. The secondary fluid, HFE, flows around this jacket, in order to provide a uniform temperature. The brass jacket unifies normally low temperature glide of the secondary fluid, HFE, used here for heating. All gaps between the two brass pieces and the test tube are filled with high thermal conductivity paste and they are held tight with metal clamps to reduce the contact thermal resistance. Thermocouples are placed at the top, bottom, and sides at 3 locations along test section. The thermocouples were attached with solder in grooves carved on the tube surface. The remaining portion of the groves is filled with high thermal conductivity paste. The thermocouples are equally spaced along the axis of the test section at 50 mm intervals starting 25 mm from the inlet of the heated section. As a result, the temperatures at 12 points on the tube surface are measured and their average values are used to calculate the heat transfer coefficient.

T-type thermocouples with a calibrated accuracy of ±0.10 °C are used to measure the refrigerant temperature and wall temperature of the test tube. The absolute pressure of CO\textsubscript{2} is determined by a pressure transducer with an uncertainty of ±3.4 kPa and pressure drops are evaluated by differential pressure transducers with the accuracy of
±0.086 kPa. The refrigerant mass flow rate is measured by a mass flow meter with an accuracy of ±0.1 % of the reading. Electrical power inputs to the calorimeter and HFE heater are measured with watt transducers which have 0.2 % reading accuracy.

The visualization section, followed by the test section, consists of a glass tube and fittings covered with a polyester block. The inner and outer diameters of glass tube are 11.7 mm and 14 mm, respectively and the length is 80 mm. The glass tube is enclosed by the thick polyester block, the size of 130 × 80 × 80 mm, in order to prevent the generation of frost caused by low temperatures and endure operating pressure.

Flow boiling heat transfer coefficients for CO$_2$ with and without oil are measured at a saturation temperature of -15 °C, mass flux of 40, 100, and 200 kg/m$^2$, heat flux of 0.5, 1, 2, 5, and 10 kW/m$^2$ and qualities between 0.1 and 0.8.

### 3. DATA REDUCTION

In order to obtain an average heat transfer coefficient, the heat transfer rate to the refrigerant which is CO$_2$ in this study, $\dot{Q}_{\text{ref}}$, is determined as shown in the following equation.

$$ \dot{Q}_{\text{ref}} = (nC_p)_{HFE} \cdot (T_{HFE,i} - T_{HFE,o}) + \dot{Q}_{\text{Amb}} - \dot{Q}_{\text{Cond}} $$

(1)

$$ h = \frac{\dot{Q}_{\text{ref}}}{A_{\text{surf}} \cdot (T_r - T_{\text{CO}_2})} $$

(2)

From Eq. (1) the heat transfer rate from secondary fluid was determined from the HFE specific heat, mass flow rate, and temperature difference between the inlet and outlet of the test section. The heat exchange rate with the environment, $\dot{Q}_{\text{Amb}}$, was obtained in a calibration experiment where an electrical heater was inserted in the test section while the power was carefully measured and presented as a function of the overall heat transfer coefficient of the test section and the log mean temperature difference between the HFE and the ambient air. The axially transferred conduction heat loss through the pipe, $\dot{Q}_{\text{Cond}}$, is estimated by a finite element code and experimentally checked. As presented in Eq. (2), the average heat transfer coefficient, $h$, is determined from the calculated heat transfer rate to refrigerant, $\dot{Q}_{\text{ref}}$, the measured average tube wall temperature, the test tube geometry, and the refrigerant saturation temperature calculated from the measured saturation pressure. Data regression and determination of refrigerant properties are performed using Engineering Equation Solver (2005).

The uncertainty of the heat transfer coefficient occurs due to the uncertainties of the independent measured parameters: temperature, pressure, mass flow, and electrical power input as presented earlier. The uncertainty propagation of heat transfer coefficient is evaluated based on Moffat (1988). The uncertainty is within the range of 8 ~ 20% of the measured heat transfer coefficients, and is shown as a vertical error bar in the figures.

### 4. RESULTS AND DISCUSSION

#### 4.1 Effect of Oil Addition in the Smooth Tube

4.1.1 Flow Pattern of Pure CO$_2$ and Oil/CO$_2$ mixtures: In Fig. 2, we compare the flow patterns of pure CO$_2$ and oil/CO$_2$ mixture of 2 % OCR obtained through visualization in a smooth tube, vapor quality from 0.1 to 0.8, and mass flux from 40 to 200 kg/m$^2$. For comparison, the Wojtan et al. (2005) flow pattern map of pure CO$_2$ is presented in both Figs. 2a-b. For pure CO$_2$ results shown in Fig. 2a, stratified and stratified-wavy flows are observed throughout all mass fluxes and vapor qualities and neither slug flow nor annular flow does appear unlike predicted by the Wojtan et al. (2005) flow pattern map. It results from the relatively small density ratio of vapor to liquid of CO$_2$ comparing to other refrigerants. Therefore the velocity difference at the vapor-liquid interface is not significant and, thus, the shear force is not enough to generate slug flow or annular flow. Park and Hrnjak (2005) also showed that the region of slug + stratified wavy flow shrank as the transition to stratified flow occurred at much lower vapor quality than predicted by Wojtan et al. (2005). It will be shown later that such stratified-type flow patterns cause decrease in the heat transfer coefficient as increase of quality, as the dry-out region on the top of the tube grows.
For the observed flow patterns of oil/CO₂ mixture of 2 % OCR, shown in Fig. 2b, more various types of flow patterns are obtained. Slug flow (low vapor qualities of 0.1 and 0.25), annular-wavy, and semi-annular flows (vapor qualities higher than 0.5) appear at mass fluxes over 100 kg/m²s, where the flow pattern had been occupied by stratified and stratified-wavy flow for pure CO₂. Our visualization results reveal that the formation of foaming, developed throughout all experimental ranges in the oil/CO₂ mixture, plays a more critical role to alter its flow pattern with oil addition. To our best knowledge, this is the first observation of foaming in the oil/CO₂ mixture, even though it is presented in the previous studies of other refrigerant and oil mixtures, e.g., naphthenic oil/R-12 by Manwell & Bergles (1994) and PAG oil/ R-134a by Wongwises et al. (2002).

The visualization results of pure CO₂ and oil/CO₂ mixture (2 % OCR) are presented in Fig. 3 at high vapor qualities of 0.7 or 0.8 for the mass fluxes of 40, 100, and 200 kg/m²s. At lowest mass flux of 40 kg/m²s, the developed foam

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**Figure 2:** The visualization results of two-phase flow patterns of (a) pure CO₂ and (b) oil/CO₂ mixture of 2 % OCR in the smooth tube at the vapor quality from 0.1 to 0.8, and the mass fluxes of 40, 100, and 200 kg/m²s, which are compared to the Wojtan et al. (2005) two-phase flow patterns map.

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**Figure 3:** Two-phase flow patterns of pure CO₂ and oil/CO₂ mixture of 2 % OCR in the smooth tube at vapor quality from of 0.7/0.8, the mass fluxes of 40 and 200 kg/m²s, and the heat flux of 10 kW/m².

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**Figure 4:** Two-phase flow patterns of pure CO₂ and oil/CO₂ mixture of 2 % OCR in the smooth tube at vapor quality from 0.1, the mass fluxes of 200 kg/m²s, and the heat flux of 10 kW/m².
layer generates waves at the liquid-vapor interface due to the increased shear force, which results in the change of flow pattern from stratified-smooth to stratified-wavy flow. As increasing the mass flux over 100 kg/m², the froth, the cluster of foaming diminished in size, tends to attach along the sidewall because of its increased surface tension and sweep the wall surface owing to the amplified vapor velocity with increase in the vapor quality. Such a mechanism of froth advances the annular-like flow pattern to appear at lower mass fluxes and qualities in the presence of oil. Besides, the sweeping motion of froaming on the sidewalls is strongly related with the flow boiling heat transfer with oil addition. At low vapor qualities for the mass flux of 200 kg/m², the slug flow pattern is observed as shown in Fig. 4a. Slug flow patterns develop when the wave size grows rapidly to touch the upper wall of the tube due to the Kelvin-Helmholtz instability, which occurs in the presence of sufficient velocity difference across the interface between two fluids. When foaming occurs with oil addition, the process of stress dispersion through revolving motions of bubbles causes the reduction of velocity of foam layer in the flowing direction and, accordingly, the velocity difference across vapor-foam interface increases to generate the slug flow. To verify this, the velocities of vapor, foam near the vapor-foam interface, and liquid layer are estimated by tracking a sampled particle in each layer, shown in Fig. 4b, to be 0.416, 0.054, and 0.251 m/s, respectively.

4.1.2 Flow Boiling Heat Transfer of Pure CO₂ and Oil/CO₂ mixtures: Fig. 5 represent the flow boiling heat transfer coefficients of pure CO₂ and oil/CO₂ mixtures with respect to the varying heat and mass fluxes at the given evaporation temperature respectively and the effects of heat flux, mass flux, and quality are studied. As the oil is added, shown in Fig. 5c-d, the flow boiling heat transfer coefficient decreases as OCR increases from 0.5 to 2 %. For higher qualities over 0.6, however, the decrease in the flow boiling heat transfer coefficient becomes almost negligible or even enhanced. It can be explained by the development of foaming as mentioned in section 4.1.1. For pure CO₂, the flow boiling heat transfer coefficient of pure CO₂ decreases as quality increases because the dry-out region on the top of the tube grows in the stratified-types flow as explained by Kim et al. (2010) through analyzing local heat transfer coefficients. When the oil is added, however, the development of foaming tends to increase the wetted fraction of the tube wall and, thus, provides positive effect of oil addition within the partial ranges.

Figure 5a-c depicts the flow boiling heat transfer coefficients of oil/CO₂ mixtures over the heat flux ranged from 0.5 to 10 kW/m² for a given mass flux of 100 kg/m²/s. The deterioration of the flow boiling heat transfer coefficient of oil/CO₂ mixtures increases as the heat flux increases, especially at low vapor qualities. It is because the nucleate boiling suppression is not obvious at low heat flux ranges. Such reduction in heat transfer coefficient is consequential with a small amount of oil addition of 0.5 % OCR, and then becomes nearly independent on OCR.

4.2 Effect of Oil Addition in the Enhanced Tube

4.2.1 Flow Pattern of Pure CO₂ and Oil/CO₂ mixtures: The visualized flow patterns of pure CO₂ and the oil/CO₂ mixture of 2 % OCR in a micro-fin tube with 30° helix angle are presented in Fig. 6 with superimposed Wojtan et al.
The predominant flow pattern of pure CO$_2$ in the enhanced tube, shown in Fig. 6a, is the semi annular flow, resulting from the enhanced momentum of the fluid by the micro-fin structure. Comparing the results oil/CO$_2$ mixtures to those of pure CO$_2$, the flow patterns remain mostly unchanged with oil addition unlike the results in the smooth tube. Figure 7 shows the visualized flow patterns of pure CO$_2$ and oil/CO$_2$ mixture of 2 % OCR at the vapor quality of 0.7 for the mass fluxes of 200 kg/m$^2$s. Even though, foaming is observed in the visualization of oil/CO$_2$ mixture, both pure CO$_2$ and oil/CO$_2$ mixture display typical annular flow. Dissimilar to the effect of foaming in the smooth tube (i.e., developing semi annular flow from otherwise stratified flow by enhancing the wetted area on the wall), the development of foaming due to oil addition does not affect the flow pattern of annular flow in the enhanced tube. It shows different tendency from the results by Manwell and Bergles (1994) that foaming disappears in the enhanced tube.

Figure 6: The visualization results of two-phase flow patterns of (a) pure CO$_2$ and (b) oil/CO$_2$ mixture of 2 % OCR in the enhanced tube at the vapor quality from 0.1 to 0.8, and the mass fluxes of 40, 60, 80, 100, and 200 kg/m$^2$s. The Wojtan et al. (2005) flow pattern map of pure CO$_2$ in the smooth tube is superimposed as a reference.

Figure 7: The visualized flow patterns of (a) pure CO$_2$ and (b) oil/CO$_2$ mixtures of 2 % OCR in the enhanced tube at vapor quality of 0.7, the mass flux of 200 kg/m$^2$s, and the heat flux of 10 kW/m$^2$.

Figure 8: Heat transfer coefficients of oil/CO$_2$ mixtures in the enhanced tube at the heat flux of 10 kW/m$^2$ and the mass fluxes of (a) 100 and (b) 200 kg/m$^2$s.
4.2.2 Flow Boiling Heat Transfer on Pure CO₂ and Oil/CO₂ mixtures: Figure 8 shows the heat transfer coefficients of pure CO₂ and oil/CO₂ mixture (2 % OCR) in the micro-fin tubes as a function of vapor quality at the mass fluxes of 100 and 200 kg/m²/s. It can be seen from Fig. 8a that in the enhanced tube the flow boiling heat transfer coefficients of pure CO₂ increase as the vapor quality increases. For the annular flow developed earlier due to the micro-fin structure effect, as shown through visualization results, the increase of vapor velocity at higher vapor quality enhances the convective heat transfer. When oil is added, the deterioration of flow boiling heat transfer coefficient in the enhanced tube is limited to the vapor qualities above 0.8, in contrast with the results in smooth tube showing significant reduction of heat transfer at low qualities. Ha & Bergles (1993) and Zurcher et al. (1997) (1999) also observed the similar trend, local deterioration of the heat transfer of the oil/CO₂ mixtures at high vapor qualities over 0.7. It can be explained by the dominant contribution of the convective boiling owing to the early-developed annular flow, which makes the effect of nucleate boiling inconsequential. At high vapor qualities, however, the oil is accumulated in the partial space between micro fins, causing increase of the thermal resistance, and, thus, the heat transfer is reduced, as suggested by Ha and Bergles (1993).

4.3 Pressure Drop on Pure CO₂ and Oil/CO₂ mixtures

Figure 9 presents the measured pressure drop, measured per 1 m long horizontal smooth tube with an accuracy of ±0.086 kPa, of pure CO₂ and oil/CO₂ mixtures from 0.5 to 2 % OCR in the smooth and micro-fin tubes. The overall magnitudes of pressure drop obtained in the smooth tube are not considerable, less than 0.5 kPa/m, due to the relatively large diameter of the tube, even though they become more significant in the enhanced tube. The pressure drop increases as the vapor quality and mass flux increase since the higher fluid velocity induces more friction between fluid and wall. The effect of oil on the pressure drop is not critical because the density of the oil is close to the density of CO₂ at the evaporation temperature of -15 °C.

5. SUMMARY AND CONCLUSIONS

The flow boiling heat transfer coefficients and visualized flow patterns are experimentally investigated in the horizontal smooth and enhanced tube of 11.2 mm inner diameter for CO₂ and oil/CO₂ mixtures at evaporation temperature of -15 °C, mass fluxes from 40 to 200 kg/m²/s, and heat fluxes from 0.5 to 10 kW/m², and for vapor qualities from 0.1 to 0.8. The effects of heat flux, mass flux, and quality on the flow boiling of heat transfer and flow pattern are presented and the changes in the heat transfer performance with oil addition are analyzed. The flow boiling heat transfer trend of pure CO₂ agrees with previous studies. The results in the smooth tube show that the strong dependence of nucleate boiling and the effect of mass flux are important to determine the flow pattern and corresponding heat transfer performance. The effect of lubricating oil is also investigated with an OCR of 0.5 to 2 %. When the nucleate boiling is dominant, oil addition degrades the heat transfer of two phase flow, due to the

Figure 9: Measured pressure drops of pure CO₂ and oil/CO₂ mixtures at the mass fluxes of 100 kg/m²/s and 200 kg/m²/s in (a) smooth and (b) enhanced tubes.

<table>
<thead>
<tr>
<th>Vapor Quality</th>
<th>Smooth Tube Pressure Drop (kPa/m)</th>
<th>Enhanced Tube Pressure Drop (kPa/m)</th>
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<tr>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.2</td>
<td>0.4</td>
<td>0.2</td>
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<tr>
<td>0.4</td>
<td>0.6</td>
<td>0.4</td>
</tr>
<tr>
<td>0.6</td>
<td>0.8</td>
<td>0.6</td>
</tr>
<tr>
<td>0.8</td>
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<td>0.8</td>
</tr>
<tr>
<td>1.0</td>
<td>1.2</td>
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suppression of nucleate boiling. However, the development of foaming resulted from the oil is shown to change flow pattern to annular-like flow and so increase heat transfer coefficient in certain cases, by reducing wall dry-out. On the other hands, in the enhanced tube, the contribution of convective boiling heat transfer is dominant due to the semi annular flow patterns developed earlier: the flow boiling heat transfer coefficient increases as the vapor quality and mass flux increase, and the effect of oil addition is much less significant, comparing to the results in the smooth tube. The pressure drop is not significant in such tubes with relative large diameter, even though it increases with the fluid velocity and micro-fin structure.

**NOMENCLATURE**

<table>
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<tr>
<th>Symbol</th>
<th>Description</th>
<th>Subscripts</th>
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<tr>
<td>(A)</td>
<td>area ((m^2))</td>
<td>(\text{Amb}) ambient</td>
</tr>
<tr>
<td>(C_p)</td>
<td>specific heat ((J/kg\ K))</td>
<td>(\text{Cond}) conduction</td>
</tr>
<tr>
<td>(G)</td>
<td>mass flux ((kg/m^2\ s))</td>
<td>(\text{HFE}) secondary fluid, HFE</td>
</tr>
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<td>(h)</td>
<td>heat transfer coefficient ((W/m^2\ K))</td>
<td>(i) inlet</td>
</tr>
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<td>(m)</td>
<td>mass flow rate ((kg/s))</td>
<td>(l) liquid</td>
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<tr>
<td>(P_{\text{red}})</td>
<td>reduced pressure</td>
<td>(o) outlet</td>
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<tr>
<td>(q)</td>
<td>heat flux ((W/m^2))</td>
<td>(\text{ref}) refrigerant</td>
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<tr>
<td>(\dot{Q})</td>
<td>heat transfer rate ((W))</td>
<td>(\text{sat}) saturation</td>
</tr>
<tr>
<td>(T)</td>
<td>temperature ({°C or K})</td>
<td>(W) wall</td>
</tr>
<tr>
<td>(x)</td>
<td>quality</td>
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Kim, S., Pehlivanoglu, N., Hrnjak, P.S., 2010, R744 Flow boiling heat transfer with and without oil at low temperatures in 11.2 mm horizontal smooth tube, *International Refrigeration and Air Conditioning Conference at Purdue*, Purdue University, West Lafayette, IN, USA.


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