

IMPROVEMENT OF A MINI ANNULUS-CHANNEL BY TWISTING TUBE WALL FOR SUPERCRITICAL CO₂ FLOW

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Abstract

A mini annulus-channel with a hydraulic diameter of 1.65 mm was improved by twisting the inner tube wall with the twisted ratio of 9.13 for enhancing heat transfer coefficients and reducing pressure losses of supercritical CO₂ flowing turbulently with the mass flux of 450 kg/m²-s. The experimented heat transfer coefficients of the supercritical CO₂ flow in both channels (before and after improvement) were figured out by a modified Wilson plot method in various conditions. It can be shown that heat transfer coefficients of the improved channel are higher evidently than that of the former channel at and near the pseudo-critical temperatures of supercritical CO₂ while heat transfer coefficients in both channels are not different when the CO₂ temperatures are far away from the pseudo-critical temperature. The apparent advantage of the improved flow channel is that it can reduce pressure losses due to the supercritical CO₂ flow by approximately 4 times of the original flow channel. In addition, a heat transfer correlation multiplied with a correction factor is suggested for this flow channel based on experimental data.

Keywords: Mini-channel, supercritical CO₂, pseudo-critical temperature

Introduction

Vapor compression heat pump (VCHP) is widely used for many applications such as air-conditioning, drying, water heating etc. Nowadays, this system is much concerned by environmental effects. Two major concerns are ozone depleting and global warming which one cause is from a release of an unfriendly environment refrigerant used in the system. Natural refrigerants are promising refrigerants due to low ozone depleting potential (ODP) and low global warming potential (GWP) such as hydrogen, carbon dioxide, propane, ammonia etc. According to criteria of refrigerants

based on environmental effect, safety and toxicity, carbon dioxide is seemed to be the best alternative refrigerant. However, based on heat source and heat sink temperatures, a VCHP used carbon dioxide as refrigerant must operate on the transcritical cycle because the critical pressure and temperature of carbon dioxide are 73.8 bars and 31°C respectively. Low pressure side will work on sub-critical pressure about 25 to 50 bars while high pressure side will work on super-critical pressure about 90 to 130 bars. Thus, this cycle will work very high pressure for both sides when compared with a sub-critical cycle

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(low pressure side is about 2 to 9 bars while high pressure side is about 10 to 40 bars for HFCs refrigerants). Fortunately, developments in materials and compressors used in this transcritical VCHP have continuously been for last decade and available in the market. An obvious advantage is, when the CO₂ transcritical VCHP is used for water heating, it can produce water temperature up to 90°C while the conventional VCHP can produce water temperature not more than 55°C (Neksa *et al.*, 1998).

The key design of the CO₂ transcritical VCHP is a heat exchanger design for cooling process in which the refrigerant condition is above the critical point. As in this condition, thermodynamic properties will be huge changed by a few degree of temperature change especially in near around the pseudo-critical temperature (The temperature, for a given pressure, at which the specific heat exhibits a maximum). Thus, design method for heat exchangers based on constant thermo-physical parameters for ordinary heat exchangers is not employed. Another design method such as a segmental design method is employed to accurately handle the considerable variation of thermodynamic properties. For a heat exchanger (gas cooler) in the CO₂ heat pump water heater, the tube-in-tube heat exchanger seems to be relevant with this design method while a plate heat exchanger is very complex to design for the gas cooler. Therefore, developments and researches have been focused on the tube-in-tube heat exchanger. To improve heat transfer, fin-tube, mini or micro tube can be used in this heat exchanger. One parameter which is the most important for the heat exchanger design is convective heat transfer coefficient. Conventional heat transfer coefficient correlations used for predicting single phase flow in tube cannot absolutely predict. Extensive review on heat transfer correlations developed for predicting heat transfer coefficients of supercritical CO₂ flow in smoothly conventional and mini channels was reviewed by Ehsan *et al.* (2018). Unfortunately, there is a lack of a universal correlation for

predicting heat transfer coefficients of CO₂ flow in each geometry. Those correlations have a limitation in use and they cannot predict whole path of the process and their predicted values will be high error when they predict at and near the pseudo-critical temperature. Therefore, the use of those correlations should be careful. They will be accepted if their predicted values are closed to experimental values. Most experimented heat transfer coefficients of the CO₂ flow in the above critical point were found in the smoothly horizontally tubes (Dang and Hihara, 2004; Cabeza *et al.*, 2017). A few experimented values were found in enhanced tubes such as in mini channel with a hydraulic diameter less than 3 mm and more than 200 μm (Liao and Zhao, 2002; Huai *et al.*, 2005; Guo *et al.*, 2020) or in tube with fin (Lee *et al.*, 2013). A mini tube can improve heat transfer coefficient of a fluid flow but pressure loss due to fluid flow will be increased by this tube. Thus, this paper proposes experimented heat transfer coefficients and measured pressure losses of CO₂ flow in two annulus channels: a smooth mini-channel and a rough mini-channel which was done by twisting the inner tube of the former channel. The heat transfer coefficients predicted by some correlations are compared to the experimented values for the smooth mini-channel. Discussion on heat transfer coefficients and pressure losses for both channels and a recommendation to use a heat transfer correlation for the improved channel will be offered.

Materials and Methods

In this research, the laboratory which was used for testing heat transfer coefficients and pressure drops had been built and installed in the Energy Unit Research, Mahasarakham University. The details are shown in Figure 1. Consider in the test section (Figure 1), the smooth tube-in-tube heat exchanger (Figure 3) was firstly installed in the test section as shown in Figure 2. The inner copper tube has

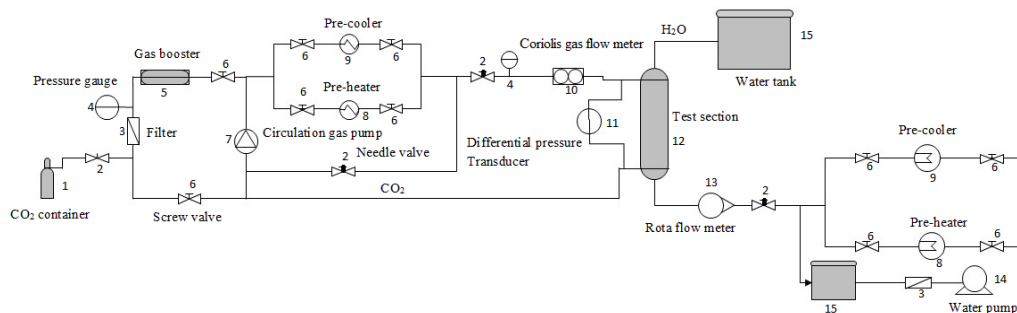


Figure 1. The test facility

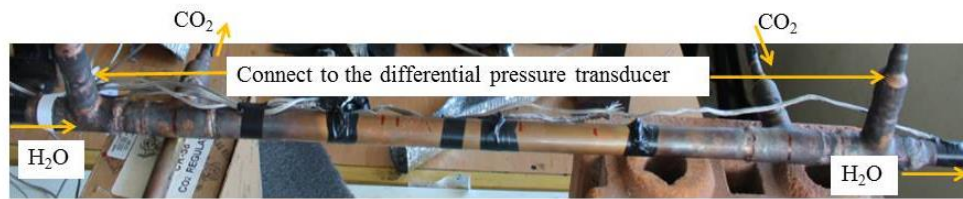


Figure 2. Heat exchanger installed in the test section before insulated

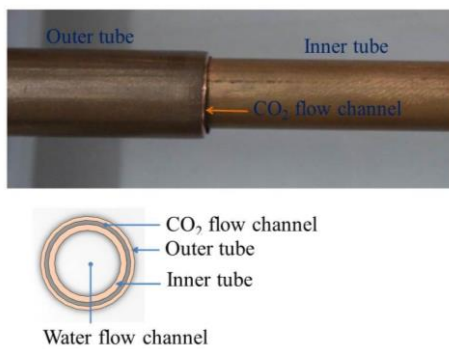


Figure 3. The smooth tube-in-tube heat exchanger

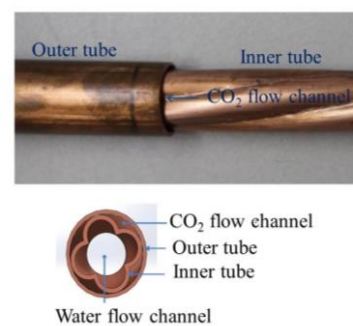


Figure 4. The twisting tube-in-tube heat exchanger

Table 1. Details of instruments

Measurement	Instrument	Model	Range	Uncertainty
Temperature	Thermocouple	Type K	-25 to +150°C	±0.05%
Pressure	Gauge pressure	Okura	0 to 200 bar	±0.10%
Gas flow rate	Coriolis mass flow meter & Transmitter	Emerson/F025 Series & IFT9701	0 to 445 kg/hr	±0.50%
Water flow rate	Rota flow meter	Dwyer RMC Series	0 to 2,000 LPH	±2.0%
Differential pressure	Differential pressure transducer	Endress+Hauser PMD75 Series	0.25 mbar to 40 bar	±0.05%

15.88 mm outside diameter and 13.84 mm inside diameter. The outer copper tube has 17.53 mm inside diameter. The length of this heat exchanger is 460 mm. The four positions of temperatures (the inlet and outlet of heat exchanger for both fluids) were measured by thermocouple type K and all temperatures measured were recorded by data logger of Agilent model 3970A. The coriolis mass flow meter connected with transmitter of Emerson/F025 series & IFT9701 and the Rota flow meter of Dwyer RMC series were installed in order to measure CO₂ mass flow rate and water flow rate respectively. The differential pressure transducer of Endress+Hauser PMD75 series and the gauge pressure of Okura were used to measure the differential pressure and gauge pressure of CO₂ flow in the test section respectively. The details of instruments are shown in Table 1. Next, the twisted tube-in-tube heat exchanger in which size and length are the same as the former heat exchanger excepted for the inner tube that is four-start spirally

twisted tube with the twisted ratio of 9.13 (Figure 4) were installed for experiments. In the experiments, both two flow channels were tested in the same conditions. Three absolute pressures: 75 bar, 80 bar and 90 bar of CO₂ flow in both channels with the constant mass flux of 450 kg/m²-s were used to be the test conditions and each pressure, CO₂ temperature was varied from 30°C to 60°C.

In each condition of CO₂, for an example, at the entrance of CO₂ flow, the pressure and the temperature were controlled by 75 bar and 30°C respectively. The eight experiments to find a CO₂ heat transfer coefficient in this condition had been done by adjusting the ten water flow rates from 300 LPM to 1000 LPM (100 LPM difference in each experiment). In addition, the inlet water temperature must be controlled by pre-heater or pre-cooler (Figure 1) to ensure that the log mean temperature difference (LMTD) in each experiment is the same value or small different. As the above reasons, the heat transfer rate will be only increased by

an increase of convective heat transfer coefficient of water in tube following an increase of water flow rate.

The methodology to find a heat transfer coefficient of CO₂ flow in each condition from experiments as mentioned above, a modified Wilson plot method (Munkejord, 1997) had been used in this research. The details of this method are described by the following.

The total thermal resistance of the heat exchanger is determined by:

$$R_{\text{tot}} = \frac{1}{\pi D_i L h_w} + \frac{\ln \frac{D_o}{D_i}}{2\pi k_{\text{wall}} L} + \frac{1}{\pi D_o L h_{\text{co}_2}} \quad (1)$$

where

- R_{tot} = Total thermal resistance, °C/kW
- h_w = Convective heat transfer coefficient of water, kW/m²°C
- h_{co_2} = Convective heat transfer coefficient of CO₂, kW/m²°C
- L = Length of heat exchange, m
- k_{wall} = Conductivity of the tube wall, kW/m°C
- D_i = Inside diameter of the inner tube, m
- D_o = Outside diameter of the inner tube, m

The convective heat transfer coefficient of the water is calculated by the Dittus-Boelter equation as:

$$h_w = C \cdot \text{Re}^m \cdot \text{Pr}^n \cdot \frac{k_w}{D_i} \quad (2)$$

where

- C, m, n = Constant values
- Re = Reynolds number
- Pr = Prandtl number
- k_w = Conductivity of water, kW/m°C

For the case of heating, substitute $n = 0.4$ into Equation (2) and place Equation (2) into Equation (1) we get:

$$\left(R_{\text{tot}} - \frac{\ln \frac{D_o}{D_i}}{2\pi k_{\text{wall}} L} \right) = \frac{1}{\pi D_i L C \cdot \text{Re}^m \cdot \text{Pr}^{0.4} \cdot \frac{k_w}{D_i}} + \frac{1}{\pi D_o L h_{\text{co}_2}} \quad (3)$$

Rewrite R_{tot} in the form of the overall heat transfer coefficient ($1/UA$) as presented in Equation (4)

$$\left(\frac{1}{U} - \frac{D_i \cdot \ln \frac{D_o}{D_i}}{2k_{\text{wall}}} \right) = \frac{1}{C \cdot \text{Re}^m \cdot \text{Pr}^{0.4} \cdot \frac{k_w}{D_i}} + \frac{D_i}{D_o h_{\text{co}_2}} \quad (4)$$

where U = The overall heat transfer coefficient, kW/m²°C

We can see that the Equation (4) is in the form of linear Equation as:

$$Y = A \cdot X + B \quad (5)$$

where

$$Y = \left(\frac{1}{U} - \frac{D_i \cdot \ln \frac{D_o}{D_i}}{2k_{\text{wall}}} \right) \quad (6)$$

$$A = \frac{1}{C} \quad (7)$$

$$X = \frac{1}{\text{Re}^m \cdot \text{Pr}^{0.4} \cdot \frac{k_w}{D_i}} \quad (8)$$

$$B = \frac{D_i}{D_o h_{\text{co}_2}} \quad (9)$$

The overall heat transfer coefficient (U) can be calculated by:

$$U = \frac{Q}{(\pi D_i L) \cdot \frac{(T_{\text{co}_2, \text{out}} - T_{w, \text{in}}) - (T_{\text{co}_2, \text{in}} - T_{w, \text{out}})}{\ln \frac{(T_{\text{co}_2, \text{out}} - T_{w, \text{in}})}{(T_{\text{co}_2, \text{in}} - T_{w, \text{out}})}}} \quad (10)$$

where

- $T_{\text{co}_2, \text{in}}$ = Inlet CO₂ temperature, °C
- $T_{\text{co}_2, \text{out}}$ = Outlet CO₂ temperature, °C
- $T_{w, \text{in}}$ = Inlet water temperature, °C
- $T_{w, \text{out}}$ = Outlet water temperature, °C

The heat transfer rate is determined by Equation (11).

$$Q = \dot{m}_w c_{p, w} (T_{w, \text{out}} - T_{w, \text{in}}) \quad (11)$$

The convective heat transfer coefficient of CO₂ in the above equations is assumed to be constant. Thus, position in y-axis and x-axis are calculated by the Equations 8 and 9 respectively; therefore, we can plot the first point in the graph. To find a convective heat transfer coefficient of CO₂, we plotted eight points in the graph depending on the water flowrate as shown in Figure 5. As a consequence, we can find convective heat transfer coefficient of CO₂ from y-intercept. However, we cannot solve the above linear equation because we have three unknown parameters (h_{co_2} , C and m). Therefore, we should guess the value of m in order to calculate the convective heat transfer coefficient of CO₂ (h_{co_2}).

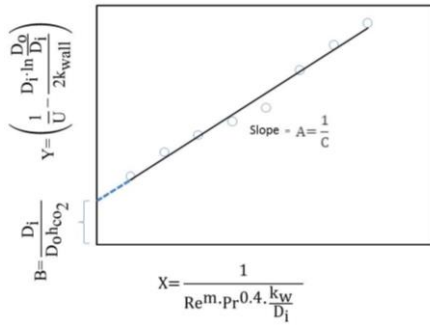


Figure 5. A Wilson Plot Method

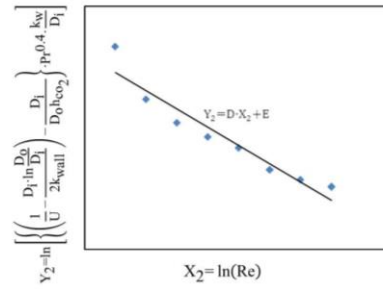


Figure 6. A logarithmic Wilson Plot

The first guess of m is resulted in plotting the second graph which is plotted by the following equations:

From Equation (4) can be rewritten by:

$$\left\{ \left(\frac{1}{U} - \frac{D_i \ln \frac{D_o}{D_i}}{2k_{wall}} \right) - \frac{D_i}{D_o h_{co2}} \right\} \cdot Pr^{0.4} \cdot \frac{k_w}{D_i} = \frac{1}{C \cdot Re^m} \quad (12)$$

where

$$Y_2 = \frac{1}{C \cdot Re^m} \quad (13)$$

Multiply $\ln$ in Equation (13) we have:

$$\ln Y_2 = -m \cdot \ln(Re) - \ln C \quad (14)$$

The Equation (14) can be written by the linear equation as:

$$Y_2 = D \cdot X_2 + E \quad (15)$$

where

$$Y_2 = \ln Y_2 \quad (16)$$

$$D = -m \quad (17)$$

$$X_2 = \ln(Re) \quad (18)$$

$$E = -\ln C \quad (19)$$

The dependent variable is determined by the Equation (20).

$$Y_2 = \ln \left[\left\{ \left(\frac{1}{U} - \frac{D_i \ln \frac{D_o}{D_i}}{2k_{wall}} \right) - \frac{D_i}{D_o h_{co2}} \right\} \cdot Pr^{0.4} \cdot \frac{k_w}{D_i} \right] \quad (20)$$

From the first guess of m we can get the y-intercept of the second graph (Figure 6). Finally, if the first guess of m is equal to the negative sign of the slope of the second graph ($-D$), the convective heat transfer coefficient of CO_2 in which we have from the y-intercept in Figure 4 is corrected. In contrast, if the first guess of m is not equal to the negative sign of the slope of the second graph ($-D$), the next guess of m is given until it is equal

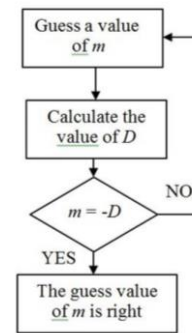


Figure 7. Flow chart diagram of the Wilson Plot algorithm (Munkejord, 1997)

the negative sign of the slope of the second graph ($-D$). The procedure is shown in Figure 7.

Results and Discussion

Heat transfer coefficients of CO_2 flow in the mini annulus-tube with mass flux of $450 \text{ kg/m}^2\text{-s}$ and hydraulic diameter of 1.65 mm predicted by correlations which were compared with experimented values in various temperatures are presented in Figure 8: at the absolute pressure (a) 75 bar, (b) 80 bar, and (c) 90 bar respectively. It was found that predicted values by correlations which are normally used for single phase flow in normal pressure (Dittus and Boelter, 1985; Gnielinski, 1976) are under experimented values for all temperatures. High difference between predicted values and experimented values were found in near and at the pseudo critical temperature for all pressures while these different values will be declined when the temperatures are away from the pseudo critical temperature. Among correlations used for predicting CO_2 -supercritical heat transfer coefficients namely Yoon *et al.* (2003); Pitla and Groll (2002); Soon and Park (2006) correlations which are normally used for predicting a smooth channel. Yoon *et al.*

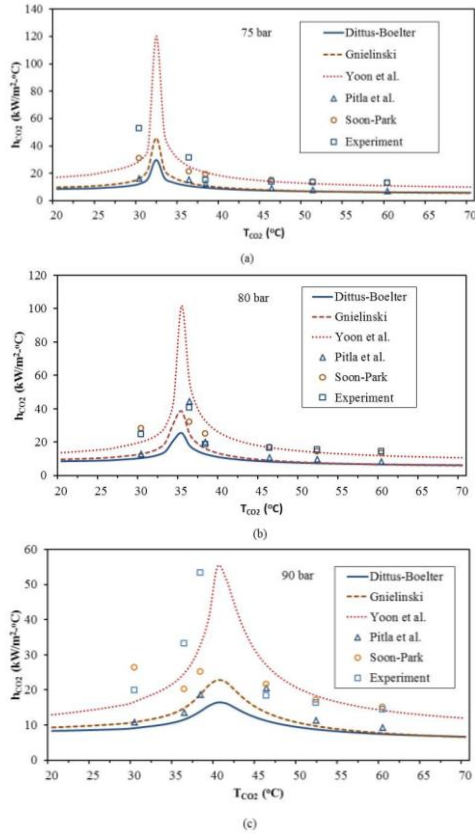


Figure 8. Heat transfer coefficients of CO₂ flow in the mini annulus-tube at different temperatures resulted by experiments and predictions: (a) 75 bar, (b) 80 bar, and (c) 90 bar

correlation gives the best predicted values with deviation of 2-8% when compared to experimented values for away from the pseudo critical temperature. However, at the near pseudo critical temperature, the predicted values are highly deviated from the experiment values. Thus, we suggested the correction factor (CF) to multiply predicted values based on the experiments for mini-smooth annulus tube as;

$$h_{CO_2,corrected} = CF \times h_{CO_2,predicted} \quad (21)$$

where

$$h_{CO_2,corrected} = \text{Corrected convective heat transfer coefficient of CO}_2, \text{ kW/m}^2\text{C}$$

$$h_{CO_2,predicted} = \text{Predicted convective heat transfer coefficient of CO}_2, \text{ kW/m}^2\text{C}$$

The correction factor is depending on each condition as the following;

For $75 \text{ bar} \leq P < 80$, at $30^\circ\text{C} \leq T < 38^\circ\text{C}$;

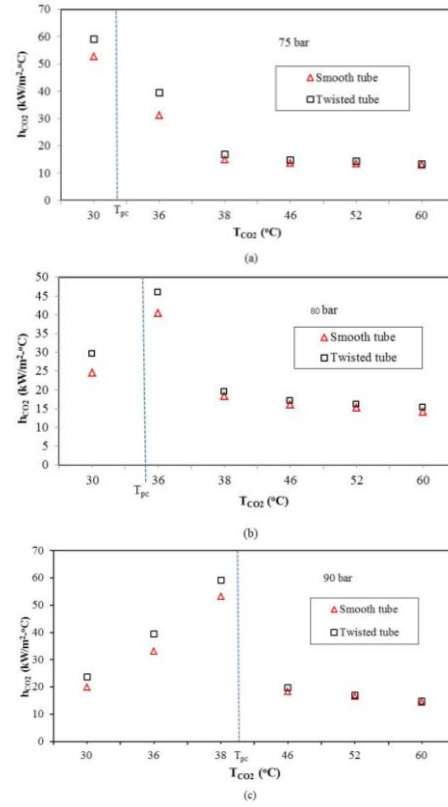


Figure 9. Comparison of experimented heat transfer coefficients of CO₂ flow in the mini annulus-channel between the smooth tube and the twisted tube: (a) 75 bar, (b) 80 bar, and (c) 90 bar

$$CF = 4.6253 - 0.1033T \quad (22)$$

For $80 \text{ bar} \leq P \leq 90$, at $30^\circ\text{C} \leq T < 46^\circ\text{C}$;

$$CF = 0.0076T^2 - 0.5927T + 12.187 \quad (23)$$

Consider Figure 9 presents heat transfer coefficients of CO₂ flow in the mini annulus-tube and flow in the mini annulus-tube with twisting tube wall at pressure of 75 bar, 80 bar and 90 bar respectively. For 75 bar and 80 bar, when the CO₂ temperature is between 30°C and 36°C, the heat transfer coefficient of CO₂ flow in the improved channel is obviously higher than that of CO₂ flow in the original channel. In addition to CO₂ temperatures mentioned above, the difference of heat transfer coefficient between the original channel and the improved channel is very small (Figures 9(a) and 9(b)). At 90 bar, the heat transfer coefficient of CO₂ flow in the improved channel is obviously higher than that of CO₂ flow in the original channel when the temperatures are in between 36°C-38°C. The results showed that the improved channel will enhance heat transfer due to

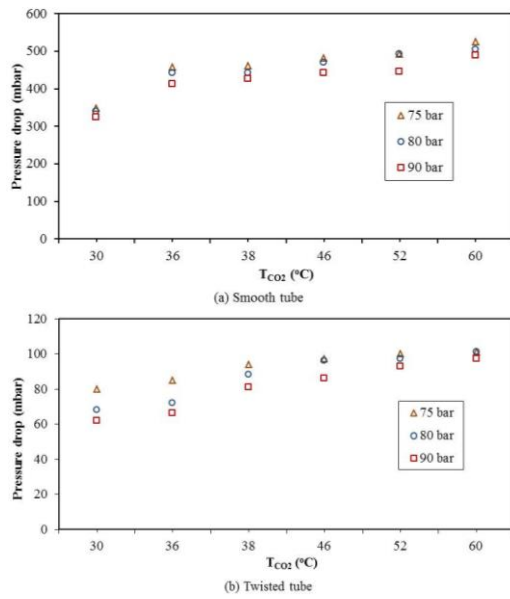


Figure 10. Pressure drops of CO₂ flow in the annulus channel with different temperatures: (a) the smooth tube and (b) the twisted tube

swirling flow in the annulus by twisting the inner tube. The effect of the swirling flow is predominant in heat transfer at the near pseudo critical temperature. It can be concluded that the magnitude differences of heat transfer coefficients will be occurred when the CO₂ temperatures are near and at the pseudo critical temperature for all pressures due to huge change thermodynamic properties with small temperature difference in this region. In the above results, we suggested heat transfer correlation for predicting the improved channel based on the experiments as;

$$\text{For } 75 \text{ bar} \leq P \leq 90, \text{ at } 30^\circ\text{C} \leq T \leq 38^\circ\text{C};$$

$$h_{CO_2} = 1.2h_{CO_2, \text{corrected}} \quad (24)$$

$$\text{For other temperatures;}$$

$$h_{CO_2} = h_{CO_2, \text{corrected}} \quad (25)$$

Note that the hydraulic diameter of the improved channel used for predicting in the above equations is the same as the original channel (before twisting).

Figure 10 presents: (a) the pressure drops of CO₂ flow in the mini annulus-tube and (b) the pressure drops of CO₂ flow in the mini annulus twisted tube. It can be seen that the pressure drops of CO₂ flow in the twisted tube are much lower than that of CO₂ flow in the smooth tube about 4 to 6 times. It can be concluded that the CO₂ flow in the mini annulus twisted tube can reduce pressure

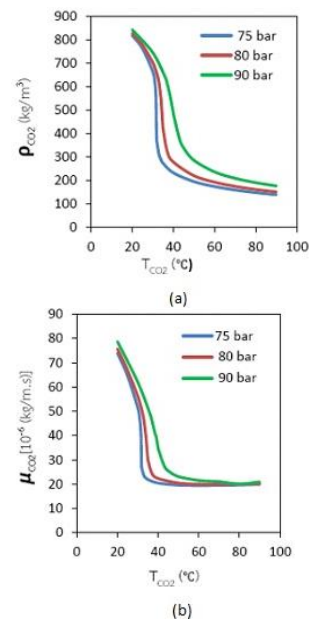


Figure 11. Thermodynamic properties of CO₂: (a) density and (b) viscosity

drops due to increasing the flow channel by twisting tube wall in the inner tube. In addition, the trends of pressure losses with respect to the pressure applied will be declined by an increase of the pressure. In the experiments, the CO₂ mass flux was controlled to be constant, average velocity of CO₂ flow in the channel will be decreased as density goes up (Figure 11(a)) or the Reynold numbers of the flow will be decreased as viscosity goes up (Figure 11(b)). For that reasons, the pressure loss will be decreased when the pressure is increased. This obvious decrease can be seen at the near pseudo critical temperature. In contrast, the trends of pressure loss with respect to the temperature applied will be increased by an increase of the CO₂ temperature.

Conclusions

A mini annulus-channel can be improved for enhancing heat transfer coefficients and reducing pressure losses of supercritical CO₂ flowing turbulently by twisting tube wall. The results from experiments showed that the improved channel can evidently increase supercritical CO₂ heat transfer coefficients when CO₂ temperature is near and at the pseudo critical temperature but an increase of supercritical CO₂ heat transfer coefficient is not occurred when CO₂ temperature is far away from the pseudo critical temperature. Fortunately, the

improved channel can reduce pressure losses approximately 4 times of that the original channel. Yoon *et al.* correlation is suitable for predicting the improve channel based on the originally hydraulic diameter when CO₂ temperature is away from the pseudo critical temperature. By contrast, a predicted value must be multiplied by a correction factor when CO₂ temperature is near and at the pseudo critical temperature.

Acknowledgements

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