International Journal of Air-Conditioning and Refrigeration Vol. 16, No. 4 (2008) 117~123

Flow and Heat Transfer Characteristics of CO_2 / Oil Mixtures in a Circular Tube

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(Received October 22, 2008; Revision received December 3, 2008; Accepted December 18, 2008)

Abstract

The present study is directed at flow and heat transfer of CO_2 and oil mixtures in a circular tube. PAG and POE oils are considered in this study. Flow characteristics of CO_2 and oil mixtures have been investigated by flow visualization. Pressure drop has been measured in the range of operating mass flow rate from 0.1 to 0.4 kg/min in a circular tube. Heat transfer characteristics of CO_2/oil mixtures have been investigated using a counterflow heat exchanger. In case of pure liquid CO_2 as well as CO_2 and POE mixtures, flow are seen to be uniform so that CO_2 and POE oil are still miscible even at flowing state. However, it is found that CO_2 and PAG are not miscible. Pressure drop of CO_2/PAG mixtures are much higher than that of CO_2/POE mixtures as well as pure CO_2 at a fixed mass flow rate. As the concentration of POE oil is increased from 0 to 5 wt%, pressure drop is increased. However, heat transfer rate and heat transfer coefficient of CO_2/POE mixtures are much higher than that of CO_2/PAG mixtures. The f-factor correlation and Nusselt number correlation for CO_2/POE oil mixtures are suggested in this paper.

Key words: CO_2/Oil mixtures, Flow visualization, Pressure drop, Heat transfer rate

Nomenclature

A_i : inner area of tube , [m^2]
A_o : outer area of tube, [m^2]
D_i : inner diameter of tube, [m]
f : friction factor
h : heat transfer coefficient, [W/m^2K]
K_L : loss factor
l : length, [m]
Nu : Nusselt number, $\frac{h_i D}{k_f}$
Pr : Prandtl number, $\frac{\nu}{\alpha}$
$\dot{Q}$: heat transfer rate, [W]
Re : Reynolds number, $\frac{4\dot{m}}{\pi D \mu}$
T_{lm} : log mean temperature difference (LMTD), [℃]
U : total heat transfer coefficient, [W/m^2K]
ΔP : pressure drop, [kPa]

Greek letters

ρ : density [kg/m^3]

1. Introduction

A modern industrial society is presently considering not only technical development, but also environmental protection to enhance the quality of life. Among the environmental problems, serious attention has been given to ozone depletion and global warming effect. Due to ozone depletion and global warming effects, the use of CFCs and HCFCs has been restricted by Montreal and Kyoto protocol. Recently, even HFCs above GWP(Global Warming Potential) 150 will be phased out for automobile systems in 2010.[1]

In order to solve this problem, environmentalists are requesting the use of natural material as refrigerant, such as NH_3, N_2, CO_2 and propane etc., in a refrigeration system. Among these natural refrigerants, carbon dioxide is a strong candidate as refrigerant for automobile air conditioning systems and heat pump

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systems with many remarkable advantages, such as environmental attractiveness, non-toxicity, non-flammability, light weight and large refrigeration capacity. It also has good refrigerant characteristics such as low surface tension and low liquid viscosity.

When an alternative refrigerant is applied to a refrigeration system, one of the most important problems is selection of suitable oil (or lubricant) on the alternative refrigerant. Oils are supplied to the contact region of the compressor and play a role of cooling, washing, sealing and protecting the corrosion. Although oils maintain mostly at the compressor, some of them circulate inside of the refrigeration system with the flow of the refrigerant and comes back to the compressor again. However, in case oils leaving the compressor cannot return to the compressor by bad solubility or miscibility of refrigerant-oil mixtures, the compressor can be damaged due to shortage of oil in the compressor. Pressure drop and heat transfer rate in the heat exchangers are also affected by the mixture of oil in the refrigerant.[2-4]

Therefore, much effort has been given to the characteristics of CO_2/oil mixtures to apply CO_2 to a refrigeration system as refrigerant. Among the recent investigations of CO_2 and oil mixtures, Choi et al.[5] presented the data for vapor pressure and miscibility of CO_2/oil mixtures. Both POE and PAG oils are considered. They found that POE is miscible but PAG is not miscible to CO_2 at liquid state. Coutinho et al.[6] found out the three-phase regions in CO_2/oil mixtures. Zingerli and Groll[7] studied heat transfer and pressure drop characteristics of CO_2/POE oil mixtures during evaporation process in supercritical region.

The present study is directed to the miscibility of CO_2/oil mixtures during flowing state by flow visualization. Pressure drop and heat transfer rate has been also investigated in the range of operating condition through a circular tube Both PAG and POE oils are considered in this study. The f-factor correlation and Nusselt number correlation for CO_2/POE oil mixtures are suggested.

2. Experimental system

Fig. 1 shows the schematic diagram of the experimental system. This is divided into three major loops, the CO_2 refrigerant loop, cooling loop of the ethylene glycol/water mixture and brine loop for a heat exchange with the refrigerant. The tube of CO_2 is connected by high-pressure fitting since the system operates at high pressure conditions.

CO_2 refrigerant is stored in a receiver maintained at the given temperature by the cooling loop. This liquid CO_2 is circulated by the magnetic gear pump and passes through the sight indicator for visualization. And then, refrigerant enters test section of 1 m length for measurement of pressure drop as seen in Fig. 2. Heat transfer rate measurement is achieved in the counterflow heat exchanger. The cross section of the counterflow heat exchanger is shown in Fig. 3. Pressure drop is measured by the differential pressure sensor(Druck corp., LPM9481) in the bypass line of test section. Also, brine for a counterflow heat exchanger is circulated by a pump inside brine cooler. The refrigerant from outlet of test section finally is gathered again to receiver. The sight indicator consists of Pyrex glass with 20 mm diameter and 10 mm thickness, established on front and back surface. Operating temperature is controlled at the constant temperature bath and operating pressure of refrigerant is measured using absolute pressure transducer (Sensotec co., THE) in this experiment. The oil injector is used for the oil mixing.

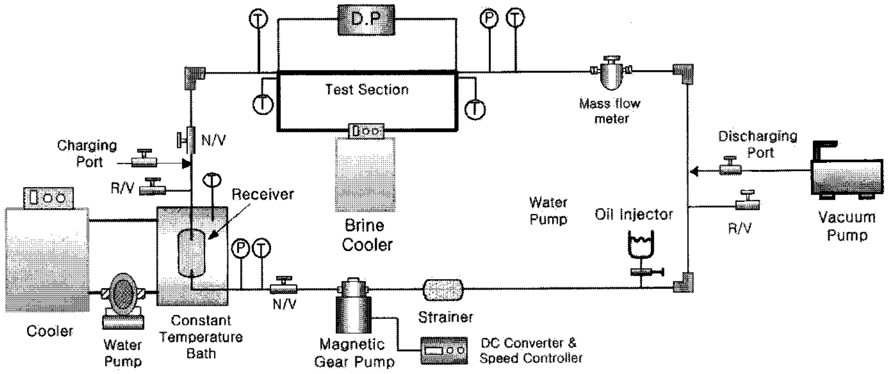


Fig. 1. Schematic diagram of experimental system.

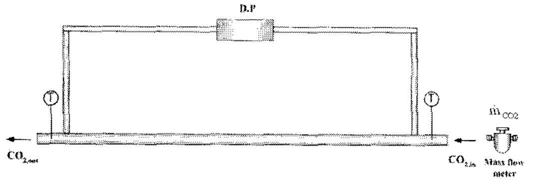


Fig. 2. Test section for pressure drop measurement.

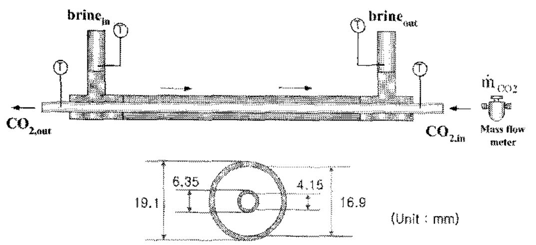


Fig. 3. Test section for heat transfer rate measurement.

Table 1. Test conditions for pressure drop measurement.

Parameter	Value	
Refrigerant	Pure CO_2	CO_2/Oil Mixtures
Inner diameter (mm)	1.74 (SUS-316)	
Inlet temperature of refrigerant (℃)	4	
Mass flow rate of refrigerant (kg/min)	0.1∼0.4 (0.05 intervals)	

Table 2. Test conditions for heat transfer rate measurement.

Parameter	Value	
Refrigerant	Pure CO_2 , CO_2/POE oil Mixtures	CO_2/PAG oil Mixtures
Inner diameter (mm)	4.15 (copper)	
Inlet temperature of refrigerant (℃)	4	
Inlet temperature of brine (℃)	-10	
Mass flow rate of refrigerant (kg/min)	0.1∼0.6 (0.1 intervals)	0.1∼0.2 (0.05 intervals)

Test conditions for pressure measurement is shown in Table 1. Tube inner diameter is 1.74 mm and inlet temperature are fixed at 4℃. Mass flow rate of refrigerant is varied from 0.1 to 0.4 kg/min. Table 2 shows test conditions for heat transfer rate measurement. Tube inner diameters of refrigerant and brine flow are 4.15 mm and 16.9 mm, respectively.

Table 3. Typical properties of the lubricant oils.

Items	Property	
Type	PAG oil	POE oil
Manufacturer	Laporte	MOBIL
Model	RFL 100-X	EAL Arctic 68
Specific gravity at 15℃	-	-0.971
Density at 15℃(kg/m^3)	-	971
Melting point (℃)	-43	-43
Boiling point (℃)	200	254
Viscosity at 40℃(cSt)	107.3	63
Viscosity at 100℃(cSt)	20.0	8.3

Inlet temperatures of refrigerant and brine flow are fixed at 4℃ and -10℃. Mass flow rate of refrigerant is varied 0.1 - 0.6 kg/min. Also, mass flow rate of brine is fixed at 5.6 kg/min, which is more than 10 times of mass flow rate of refrigerants. Therefore, the brine is assumed to be isothermal state at -10℃. The concentration of PAG oil is fixed at 1 wt% and that of POE oil is varied from 1 to 6 wt%. Total weight of refrigerant CO_2 is measured during charging into the experimental system. Weight of oil injected into the system is also measured. Thus, oil concentration can be estimated by weight ratio of oil and CO_2. Table 3 shows typical properties of oils employed in this study.

3. Result and discussion

3.1 Flow visualization

Fig. 4 shows flow visualization of the pure CO_2 refrigerant. Pure CO_2 is circulated as a compressed liquid state. The flow of the pure CO_2 looks to be packed in the sight indicator within operating temperature ranges. Fig. 5(a) indicates flowing state of the CO_2/PAG oil mixtures for 1 % oil concentration and Fig. 5(b) indicates that of the CO_2/POE oil mixtures for 3 % oil concentration. As seen in Fig. 5(a), the CO_2 refrigerant is not mixed with PAG oil. Thus, oil trap is formed in the sight indicator entrance. The PAG oil in the oil trap displays flow pattern that is spouted out as a type of liquid droplet by flow of mixtures. As a result, PAG oil remains behind inside refrigeration system and may result in lubrication failure in the compressor finally. However, in case of CO_2/POE oil mixtures, as seen in Fig. 3(b), the flow

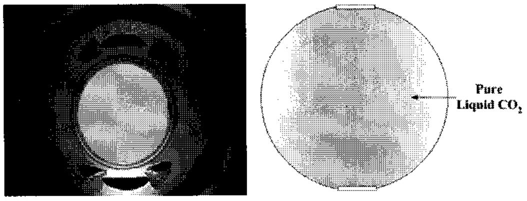
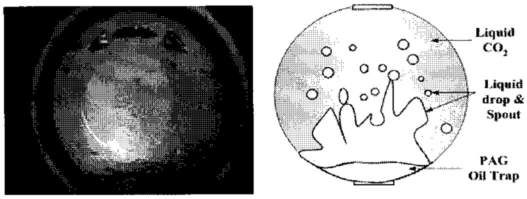
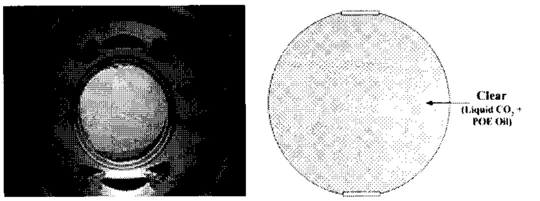


Fig. 4. Flow visualization of pure CO_2 refrigerant.



(a) CO_2 and PAG oil mixtures



(b) CO_2 and POE oil mixtures

Fig. 5. Flow visualization of CO_2 and oil mixtures.

is clear because POE oil is mixed well with CO_2. Merely, hazing flow pattern is observed by deflection of light according to density difference of the refrigerant and the oil.

3.2 Pressure drop

Fig. 6 shows the comparison of pressure drop for the pure CO_2 refrigerant between present experimental data and prediction value. Predicted pressure drop is calculated by Eq. (1) and (2)[8].

$$\Delta P = f\frac{l}{D}\frac{\rho V^2}{2} \tag{1}$$

$$f = \frac{0.316}{\mathrm{Re}^{1/4}} (\mathrm{Re} \le 10^5) \tag{2}$$

As the mass flow rate is increased, the pressure drop is gradually increased, as expected. Both experimental data and predicted values have a good agreement in the range of low mass flow rate and a little discrepancy is observed in the high mass flow rate range.

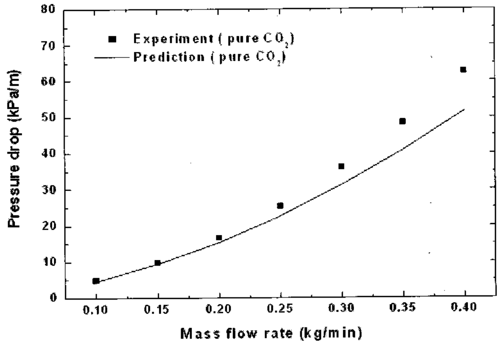


Fig. 6. Comparison of pressure drop for the pure CO_2 refrigerant on experiment and prediction value.

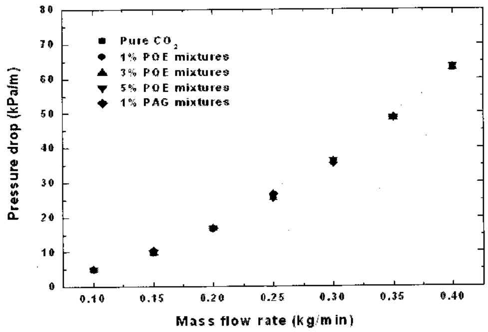


Fig. 7. Comparison of pressure drop for CO_2/oil mixtures.

Fig. 7 shows the effect of oil concentration on pressure drop for CO_2/oil mixtures. As the oil concentration is increased, the pressure drop is slightly increased for CO_2/POE oil mixtures. It is also found that pressure drop of CO_2/PAG oil mixtures is higher than that of CO_2/POE oil mixtures for 1% oil concentration. This is because dynamic viscosity of the PAG oil is higher than that POE oil. Friction factor is plotted as a function of Reynolds number for various oil concentrations using the pressure measurement, as shown in Fig. 8. Friction factor correlation is obtained for CO_2/POE oil mixtures, as expressed by eqn. (3). Here, C is oil concentration of POE (wt%) and correlation coefficient is 0.997.

$$f = 0.0086\mathrm{Re}^{-0.1131}(1 - 0.01\times \mathrm{C})^{0.05} \quad (10^4 \le \mathrm{Re} \le 6\times 10^4) \tag{3}$$

3.3 Heat transfer rate

The test section is a double-tube type counter-flow heat exchanger, which consists of a 4.15 mm ID inner tube mounted concentric to a 16.9 mm OD outer tube. Heat transfer rate can be calculated by

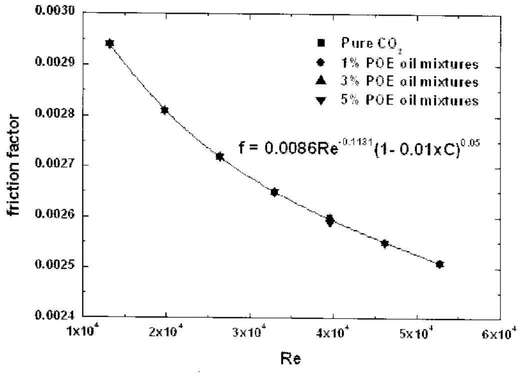


Fig. 8. f-factor correlation for CO_2/POE oil mixtures.

enthalpy difference between the inlet and the outlet of refrigerant loop in the test section. This heat transfer rate can be also estimated using overall heat transfer coefficient and LMTD (log mean temperature difference) by eqn. (4), Here, LMTD is expressed by eqn. (5). Preliminary experiment has been carried out to obtain heat transfer coefficient h_o using water inside the inner tube instead of CO_2/oil mixture and brine in outer tube. Dittus-Boelter correlation[9] is employed for hi as given in eqn. (6) in order to estimate ho for brine flow in the outer tube. This h_o is used to obtain hi inner tube for oil mixture experiments.

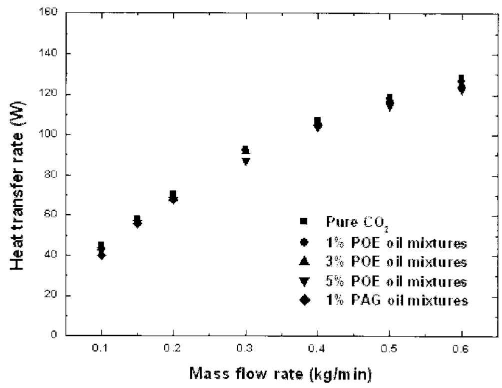
$$\dot{Q} = \dot{m}_{CO_2}(h_{out,CO_2} - h_{in,CO_2}) = UA\Delta T_{lm} \quad (4)$$

$$\Delta T_{lm} = \frac{(T_{in,CO_2} - T_{out,CO_2})}{\ln[(T_{in,CO_2} - T_{out,brine})/(T_{out,CO_2} - T_{in,brine})]} \quad (5)$$

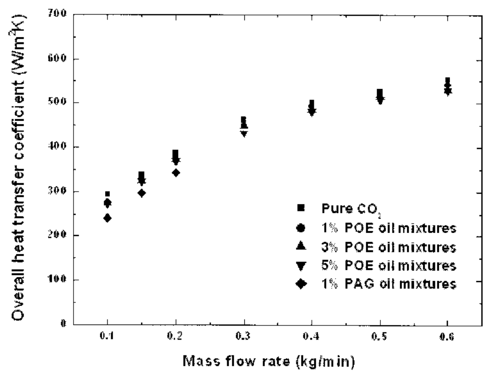
$$UA = \frac{1}{\frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_o A_o}} \quad (6)$$

$$\mathrm{Nu}_d = \frac{h_i D}{k_b} = 0.023(\mathrm{Re}_d)^{0.8}\mathrm{Pr}^{0.3} \quad (7)$$

Heat transfer rate and overall heat transfer coefficient are shown in Fig. 9 for various oil mixtures. As mass flow rate of the oil mixtures is increased, heat transfer rate and overall heat transfer coefficient are increased, as expected. However, heat transfer rate is decreased with an increase in the oil concentration for CO_2/POE oil mixtures at a given mass flow rate of mixtures. This is because the specific heat and thermal diffusion coefficient of CO_2/POE oil mixtures are



(a) heat transfer rate



(b) overall heat transfer coefficient

Fig. 9. Comparison of heat transfer rate and overall heat transfer coefficient for pure CO_2 and CO_2/oil mixtures. 3.3 Heat transfer rate.

decreased with an increase in the oil concentration. Similarly, the heat transfer rate of CO_2/PAG oil mixtures is seen to be lower than that of pure CO_2.

Fig. 10 shows heat transfer coefficient hi of CO_2/oil mixtures as a function of Reynolds number for various oil concentrations. Heat transfer coefficient is increased as Reynolds number increased. It is also found that heat transfer coefficient is decreased with an increase in the oil concentration for CO_2/POE oil mixtures at a given mass flow rate of mixtures. It is also interesting to note that heat transfer coefficient of CO_2/POE oil mixtures is higher than that of CO_2/PAG oil mixtures at a fixed oil concentration 1 wt%. Nusselt number is plotted as a function of Reynolds number for various oil concentrations using heat transfer coefficient, as shown in Fig. 11. Nusselt number correlation is obtained for CO_2/POE oil mixtures, as expressed by eqn. (8). Here, C is oil concentration of POE (wt%) and correlation coefficient is 0.998.

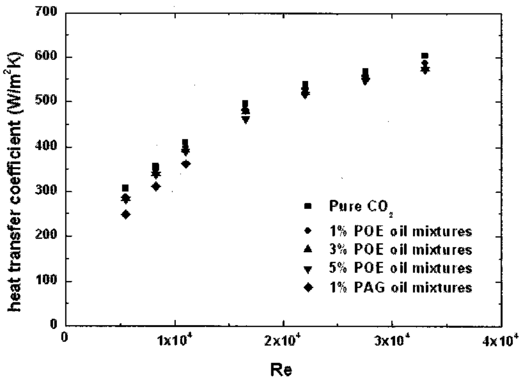


Fig. 10. Comparison of heat transfer coefficient for pure CO_2 and CO_2/oil mixtures.

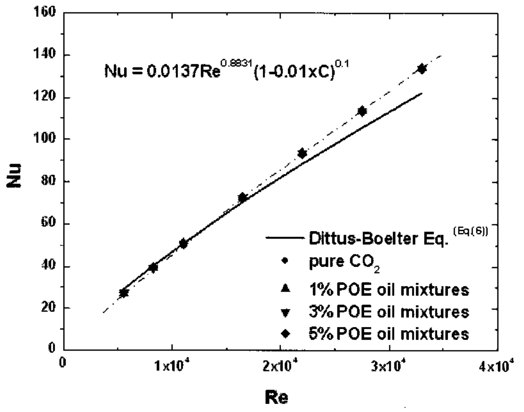


Fig. 11. Nusselt number correlation for CO_2/POE oil mixtures.

$$Nu = 0.0137\,\mathrm{Re}^{0.8831}(1-0.01\times \mathrm{C})^{0.1}$$
$$(5\times10^3 \le \mathrm{Re} \le 4\times10^4) \qquad (8)$$

4. Conclusions

An experimental study has been carried out to investigate the flow visualization, pressure drop and heat transfer characteristics of pure CO_2 and CO_2/oil mixtures.

In case of pure liquid CO_2 as well as CO_2/POE mixtures, flow are seen to be clear so that CO_2 and POE oil are still miscible even at flowing state. However, for the CO_2/PAG oil mixtures, CO_2 refrigerant is not mixed with PAG oil and oil trap is formed. Therefore, POE oil is more suitable to CO_2 refrigerant than PA oil from the miscibility point of view.

The pressure drop of CO_2/oil mixture is gradually increased as mass flow rate is increased. As the oil concentration is increased, the pressure drop is slightly increased for CO_2/POE oil mixtures. It is also found that pressure drop of CO_2/PAG oil mixtures is a little higher than that of CO_2/POE oil mixtures at given oil concentration.

Heat transfer coefficient is increased as Reynolds number is increased. It is also found that heat transfer coefficient is decreased with an increase in the oil concentration for CO_2/POE oil mixtures at a given mass flow rate of mixtures. It is also interesting to note that heat transfer coefficient of CO_2/POE oil mixtures is higher than that of CO_2/PAG oil mixtures at a fixed oil concentration.

Acknowledgment

This work was supported by Research Program 2008 of Kookmin University, Korea.

References

[1] Clodic, D., 2008, Latest HVAC&R Development Trends in European Response to Imminent HFC Regulations, Plenary Session, Proc. 12th International Refrigeration Conference at Purdue, West Lafayette, USA.

[2] Lee, J. H. and Park, Y. M., 2002, Vapor Pressure and Miscibility for R404A/POE Oil Mixtures, Korean Journal of Air-Conditioning and Refrigeration Engineering, Vol. 14, No. 4, pp.285-292.

[3] Lim, T. W. and Kim, J. H., 2003, A Study on Pressure Drop in Two-Phase Flow Boiling of Refrigerants in Horizontal Tube, Korean Journal of Air-Conditioning and Refrigeration Engineering, Vol. 15, No. 6, pp. 510-517.

[4] Mori, K., Onishi, J., Shimaoka, H., Nakanishi, S., and Kimoto H., 2002, Cooling Heat Transfer Characteristics of CO_2 and CO_2-Oil Mixtures at Supercritical Pressure Conditions, Proceeding of the Asian Conference on Refrigeration and Air Conditioning 2002, pp. 81-86.

[5] Choi, H. S., Kang, B. H., Park, K. K., and Kim, S. H., 2004, An Experimental Study on Miscibility and Vapor Pressure of R-744/Oil Mixtures, Korean Journal of Air- Conditioning and Refrigeration Engineering, Vol. 16, No. 2, pp. 150-157.

[6] Coutinho, J. A. P., Joergensen, M., and Stenby, E. H., 1995, Predictions of three-phase regions in CO_2-oil mixtures, Journal of Petroleum Science & Engineering, Vol. 12, No. 3, pp.201-208.

[7] Zingerli, A. and Groll, E. A., 2000, Influence of Refrigeration Oil on the Heat Transfer and Pressure

Drop of Supercritical CO_2 during In-Tube Cooling, IIF-IIR Commission B1,B2,E1 and E2, Purdue Univ., USA-2000, pp. 269-278.

[8] Munson, B. R., Young, D. F., and Okiishi, T. H., *Fundamentals of Fluid Mechanics*, 2nd Edition, Wiley, New York, 1997, pp. 552-575.

[9] William, S. J., *Engineering Heat Transfer*, 2nd Edition, CRC Press, New York, 2000, pp. 461-466.