The Effect of Inner Grooved Tubes on the Heat Transfer Performace of Air-Cooled Heat Exchangers of Co2 Heat Pump System

Ryuhei Kaji
ryuuhei.kaji@daikin.co.jp

Shun Yoshioka

Hirokazu Fujino

The Effect of Inner Grooved Tubes on the Heat Transfer Performace of Air-Cooled Heat Exchangers for CO_2 Heat Pump System

Ryuhei KAJI[1]*, Shun YOSHIOKA[1], Hirokazu FUJINO[1]

[1]Daikin Industries LTD., Kanaoka-cho, Kita-ku, Sakai-shi,
Osaka, 591-8511, JAPAN
Contact Information (E-mail : ryuuhei.kaji@daikin.co.jp)

ABSTRACT

In the CO_2 heat pump system, Poly alkyl glycol (PAG) oil is commonly used for lubrication of the compressor, while it is reported that the PAG oil has influence on the heat transfer performance due to its immiscibility against CO_2 within its working condition. Experimental work has been carried out to investigate the heat transfer performance of fin and tube heat exchangers using CO_2 as a refrigerant with taking PAG oil into account. From previous study it was found that the heat transfer performance decreased significantly in both evaporator and gas-cooler conditions when oil was mixed. The present work presents experimental results on three types of air-cooled heat exchangers with smooth and inner grooved tubes. It was found that the deterioration ratio of heat transfer performance with oil was different depending on the inner surface geometry of the grooved tubes. To understand this, flow visualization inside these tubes has also been carried out through transparent section made of glass, which can withstand high pressure. It was confirmed that the oil behavior inside tubes was related to heat transfer performance. Heat transfer performance can be improved by using inner grooved tubes with the optimal pattern to remove oil away from inner surface. These tubes are effective to develop high performance heat exchangers for the CO_2 heat pump system.

1. INTRODUCTION

Global warming is one of the great issues from an environmental point of view and to eliminate its cause is imperative. Since hydro fluorocarbon (HFC) refrigerants, widely used in air-conditioning and refrigeration industries, has a high global warming potential (GWP), natural refrigerants has been paid an attention. One of the most promising refrigerants is Carbon dioxide (CO_2, R744).

Due to the thermo physical properties of CO_2, the operating pressure becomes high. The high-pressure side works above supercritical; therefore, the temperature gliding occurs in its cooling process of air conditioner. Taking advantage of its temperature gliding in cooling process, this cycle has been successfully commercialized, especially in Japan, for hot water supply heat pump system. For using air-conditioning system, however, a few concerns are remaining: the operating pressure, immiscibility of lubricant oil and so on. In the former, the operating pressure of CO_2 requires three times higher than that of R410A. This causes larger loss of expansion process and thickened pipe wall. In the latter, since the CO_2 cycle commonly uses Poly alkyl glycol (PAG) oil for lubricant of the compressor, which is immiscible under the wide range of the operating condition, the performance of heat exchangers becomes worse when oil enters. Several researches have been reported for this pitfall (Hihara and Dang (2007), Katsuta et al. (2007), Higashiue et al. (2007), Gao and Honda (2006)). Yoshioka et al. (2008a, 2008b) have investigated the effect of PAG oil on CO_2 heat transfer performance and confirmed it by the refrigerant flow visualization that oil flow is related to the performance.

The objective of this study is to investigate the effect of inner grooved tubes on the heat transfer performance of air-cooled heat exchanger with refrigerant and oil flow. Two inner grooved tubes having different geometries were examined in this study. This paper describes experimental investigation to clarify the effect of these grooved tubes on heat transfer performance and evaluation of effectiveness of these tubes in applying to air-cooled heat exchangers.

2. EXPERIMENTS

2.1 Experimental apparatus

Figure 1 shows the schematic diagram of the experimental apparatus. Cycle can complete in both gas cooler and evaporation condition to control valves. The case of evaporation test is shown here. The refrigerant supply system consists of a test section, a compressor, a coriolis mass flow meter. In addition, a separated oil circulation line is located to enable providing PAG oil within refrigerant flow. The system was designed enabling controlled CO_2 refrigerant conditions such as pressure, temperature, quality, mass flow rate, and oil flow rate flowing into the test section. The oil circulation rate (OCR) was measured by capturing the refrigerant into the sampling tank, which was set in the circuit, corresponding to ASHRAE standard (1996). Oil circulation rate, *C*, is defined as follows:

$$C = m_{oil} / \left(m_{ref} + m_{oil} \right) \tag{1}$$

Two types of test sections were prepared for this study. One is a double pipe heat exchanger and the other is a fin-and-tube heat exchanger. The double pipe heat exchanger, shown in Figure 2, was used for measuring the heat transfer performance of tubes, whilst the fin-and-tube heat exchanger was for confirming the performance as an assembly, Figure 3.

The double pipe heat exchanger, made of stainless steel, has 12.88mm internal diameter and is completed by setting a copper-made heat transfer tube of 7mm internal diameter through it. Refrigerant flows inside the tube and water flows the outside annulus path concurrently. The test section has 1m in effective heat transfer length. The whole test section was set horizontally and thermally insulated during the experiment.

The fin-and-tube heat exchanger composed of copper tubes of 7mm outer diameter and aluminum fins of a 0.1mm thickness. This heat exchanger was set in front of the air chamber producing a certain air flow within the room where the temperature and humidity was controlled.

Platinum resistance thermometers were used to measure the refrigerant and water or air temperature. Pressure transducer and differential pressure cell were used to measure the pressure of refrigerant-side. For the experiment using double pipe heat exchangers, average heat transfer coefficient in tube length was obtained by subtracting thermal resistances of outer tube and wall conduction from total thermal resistance. Thermal resistance of outer tube was obtained by using Wilson plot method.

2.2 Specification of the test section

Table 1 shows the specifications of the test tube. The test tube, made of copper, has 7mm outer diameter. Smooth and two types of grooved tubes, spiral and herringbone, were used in a series of experiment. The conditions at which data were taken are listed in Table 2. Air-side conditions were set to meet the actual operation of an inside unit of an air conditioner.

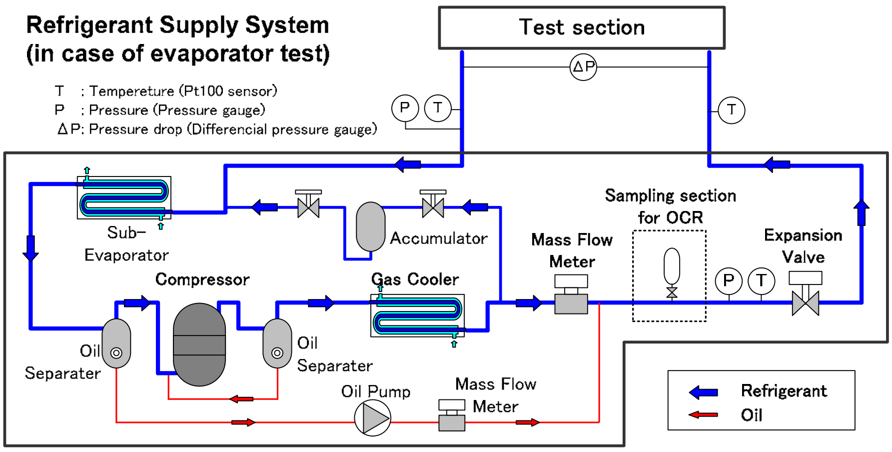
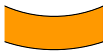
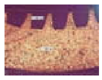
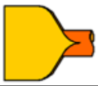
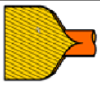
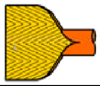
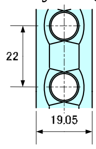
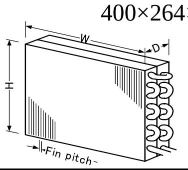


Figure 1: Schematic diagram of the experimental apparatus.

Table 1: Geometries of test tubes.

Tube		**Smooth**	**Spiral**	**Herringbone**
(Before tube expansion)	Cross-sectional view			
	Inner surface pattern			
	Outside diameter [mm]	7.00	7.00	7.00
	Bottom wall thickness [mm]	0.96	0.89	0.59
	Fin height [mm]	-	0.25	0.25
	Apex angle [deg]	-	15	20
	Helix angle [deg]	-	10	15
	Number of fin [-]	-	40	60
	Fin bottom width [mm]	-	0.26	0.14
	Averaged tube thickness [mm]	0.96	0.96	0.69
	Surface area expansion ratio [-]	1.00	1.80	2.05
Tube for heat exchanger (After tube expansion)	Bottom wall thickness [mm]	0.89	0.82	0.56
	Fin height [mm]	-	0.20	0.15
	Groove bottom width [mm]	-	0.16	0.13
	Averaged tube thickness [mm]	0.89	0.89	0.57
	Surface area expansion ratio [-]	1.00	1.66	1.59
Fin	Type	Wavy fin (hydrophilic coated)  fin thickenss t = 0.1		
	Fin pitch [mm]	1.4		
Size	W×H×D [mm]	400×264×38.1 		
	Step×Row -	12×2		

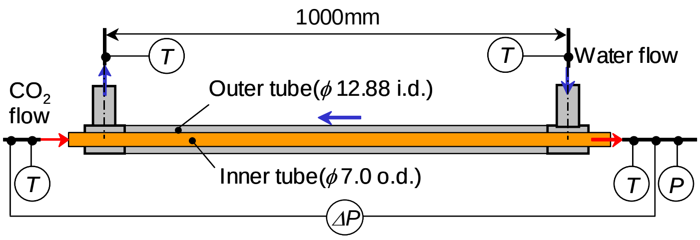


Figure 2: Schematic diagram of the test section (Experiment using double pipe heat exchangers).

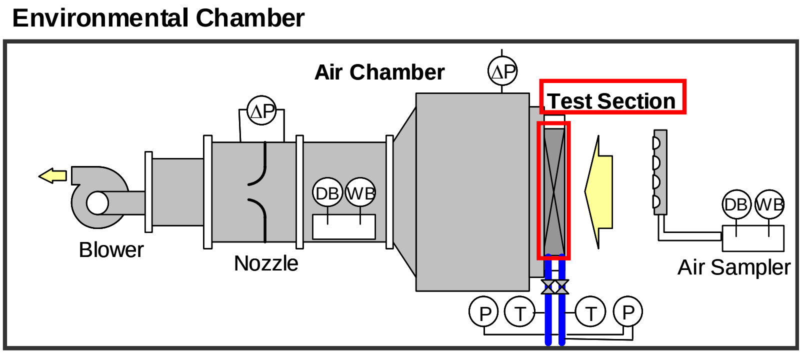
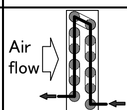
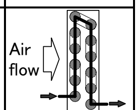


Figure 3: Schematic diagram of the test section (Experiments using heat exchangers).

Table 2: Experimental conditions.

				Gas cooler	Evaporator
Air	Dry bulb/Wet bulb temp.		[℃]	20 / 15	27 / 19
	Frontal velocity	V_a	[m/s]	1.0	
Refrigerant (R744)	Pressure	P	[MPa]	10.00	4.18
	Inlet temperature	T_i	[℃]	90.0	-
	Outlet temperature	T_o	[℃]	30.0	-
	Inlet quality	x_i	[-]	-	0.2
	Superheat temp.	T_{sh}	[K]	-	3.0
	Flow pattern			 Air flow Counter flow	 Air flow Parallel flow
Oil(PAG)	Oil circulation rate	C	[wt%]	0, 0.5	

3. RESULTS AND DISCUSSION

3.1 Performance in gas-cooler

Figures 4 and 5 show the heat transfer coefficient and pressure drop against refrigerant temperature for smooth and two grooved tubes at refrigerant pressure is 10.0MPa, mass flux is 600kg/m^{2}s and heat flux is 10kW/m^2. Filled and open plots indicate the results without and with oil. As seen in Figure 4, the heat transfer coefficient has maxima around 45°C which is the pseudo-critical temperature of 10.0MPa. Results without oil shows that the heat transfer coefficient of smooth tubes fit well with the broken line indicative of the correlation by Gnielinski(1976). As for the spiral and herringbone tubes the experimental data are proportional to the correlation by Ginelinski and 1.8 and 2.1 times higher than smooth tubes. These are the same as the inner surface area expansion ratio; therefore, it seems that the effect of the grooved geometry has not seen in gas cooler condition. When OCR is 0.5wt%, it was found that the results of herringbone tubes had the highest heat transfer rate and the least effect of oil, while the results of spiral tubes drop remarkably and are similar to smooth tubes.

As for the pressure drop, the data increases with refrigerant temperature, Figure 5. This is due to the density decrease with increasing temperature. The results also show that the increment of pressure drop is steeper near the pseudo-critical temperature. When no oil exists, the data of smooth tubes fit well with the correlation by Blasius (1913). For the spiral and herringbone tubes the experimental data are proportional to the correlation by Blasius and 2.4 and 2.7 times higher than smooth tubes. When OCR is 0.5wt%, the effect of oil is higher for smooth tubes than spiral and herringbone tubes. Similar to the results of the heat transfer coefficient, it was found that the results of herringbone tubes had the highest pressure drop and the least effect of oil.

It can also be seen that as increasing the refrigerant temperature, the effect of oil on pressure drop becomes larger. This can be estimated that the difference of density between the refrigerant and oil becomes bigger with temperature; therefore, the larger amount of oil having higher density flow along the inside tube wall and actual cross sectional area refrigerant flows shrinks.

3.2 Performance in evaporator

Figures 6 and 7 show the heat transfer coefficient and pressure drop against quality for each tube at refrigerant pressure is 4.18MPa, mass flux is 600kg/m^{2}s and heat flux is 10kW/m^2. Filled and open plots indicate without and with oil. A broken line in Figure 6 indicates the correlation by Kandlikar (1990) as a reference. This gives reasonable prediction for the results of smooth tubes but discrepancies are seen at lower and higher quality. As seen in the figure, when OCR is 0.5wt%, the results of herringbone tubes have the least deterioration rate of heat transfer, while the results of spiral tubes drop remarkably and are similar to smooth tubes. Furthermore, the heat transfer coefficient of herringbone tubes increases with quality that was also observed in HFC refrigerants, Ebisu et al. (1997).

In Figure 7, the results show that the pressure drop increases with quality for every tubes. The broken line indicates the correlation by Friedel (1979) as a reference. The correlation slightly overpredicts the results of smooth tubes. For spiral and herringbone tubes the effect of oil is less in contrast to smooth tubes for which the difference between with and without oil becomes larger in higher quality. This can be explained as follows. The oil partially miscible to the liquid phase of refrigerant extracts when quality becomes higher and flows slowly along the inner surface of the smooth tubes whereas between ditches of the grooved tubes. The oil layer causes the drag for smooth tubes, but inner surface geometry is more dominant on pressure drop for spiral and herringbone tubes.

From the above results, it was found that the heat transfer coefficient and pressure drop were different depend on the inner grooved geometry and herringbone tubes were effective to mitigate the deterioration of heat transfer coefficient due to the PAG oil. This can be estimated the oil flow configuration inside the tube is related. To clarify this, flow visualization inside tubes were carried out.

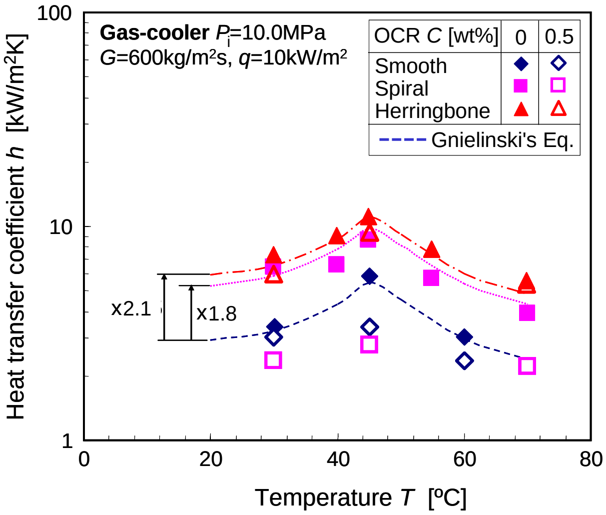


Figure 4: Effect of temperature on heat transfer coefficient in gas cooler condition.

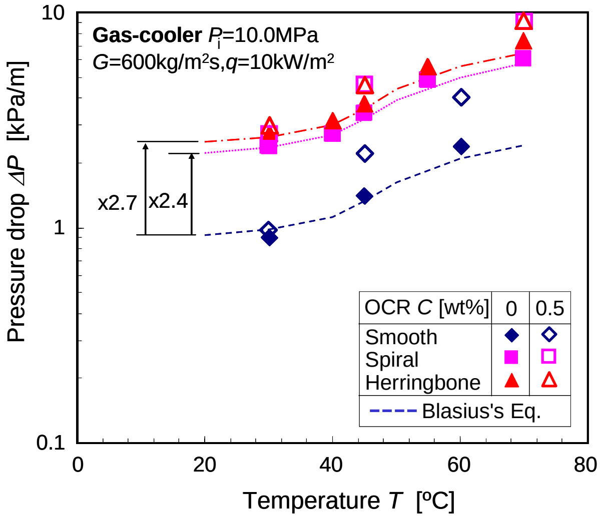


Figure 5: Effect of temperature on pressure drop in gas cooler condition.

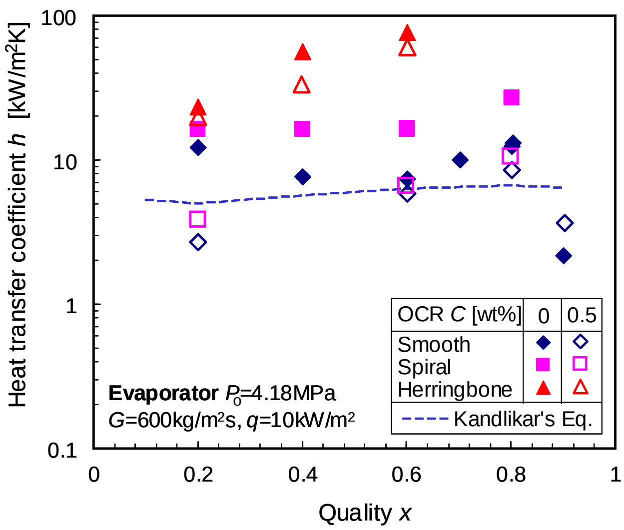


Figure 6: Effect of quality on heat transfer coefficient in evaporator condition.

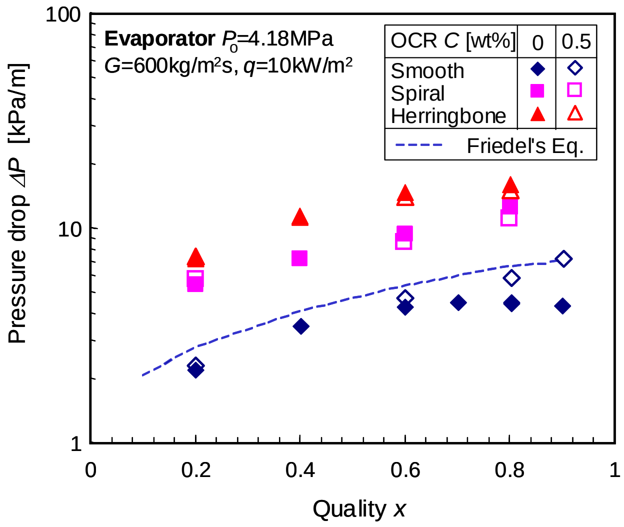
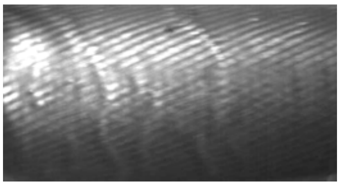


Figure 7: Effect of quality on pressure drop in evaporator condition.

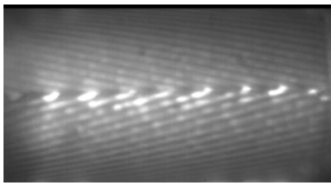
3.3 Results of flow visualization

Figure 8 shows the images of spiral and herringbone tubes in gas-cooler condition. Visualization made possible by using tubes cut partially which were inserted to the sapphire glasses with 7.1mm of internal diameter that is slightly larger than the outside diameter of the tubes. Images were taken by using high-speed camera through the transparent glasses. It can be observed that oil flows as film along the tube wall and droplets in the tube core. For spiral tubes great amount of oil film forms waves along the inner tube wall which covers over the ditches. In contrast, for herringbone tube, these waves disappear and oil flows as drops between ditches. Additionally, since the grooves are discontinuous for herringbone tubes, these drops meet at the aggregating point to grow larger oil drops and tear off to the tube core. In contrast, relatively thinner oil might exist at separating point. From these observations, it was estimated that thinned oil film made the deterioration of heat transfer to ease.

Flow direction ⟹



(a) Spiral tube



(b) Herringbone tube

Figure 8: Visualization of oil flow inside heat transfer tubes
(Gas cooler condition, P_i=10MPa, T=30°C, G=600kg/m^{2}s, C=0.7wt%).

3.4 Performance of heat exchangers

From above, it was found that the deterioration of heat transfer due to PAG oil was the least for the herringbone tubes in both gas-cooler and evaporator conditions. This section describes the evaluation of air-cooled heat exchangers to confirm the effect of enhancement of heat transfer coefficient by grooved tubes.

Figures 9 and 10 show the results in gas-cooler and evaporator. The ordinate indicates the ratio of heat transfer rate to the performance of smooth tube without oil. As for gas-cooler, shown in Figure9, when no oil exists, the heat transfer performance becomes higher in order of smooth, spiral and herringbone tubes. When OCR is 0.5wt%, the deterioration rates are 0.9%, 7.9% and 2.5% for smooth, spiral and herringbone tubes, respectively. For heat exchangers of herringbone tubes, it is higher than those of smooth tube but better than those of spiral tubes. Heat

exchangers using herringbone tubes perform 7.5% higher heat transfer than those using smooth tubes. As for evaporator, shown in Figure 10, alike to the gas cooler the heat transfer performance becomes higher in order of smooth, spiral and herringbone tubes when no oil exists. When OCR is 0.5wt%, the deterioration rates are 9.6%, 8.3% and 4.3% for smooth, grooved and herringbone tubes, respectively. Heat exchangers of herringbone tubes show 17.7% higher performance than those of smooth tubes.
From these, it was confirmed that the improvement of heat exchanger performance could be obtained by using herringbone tubes. It can be seen that greater effect is gained in evaporator than gas cooler.

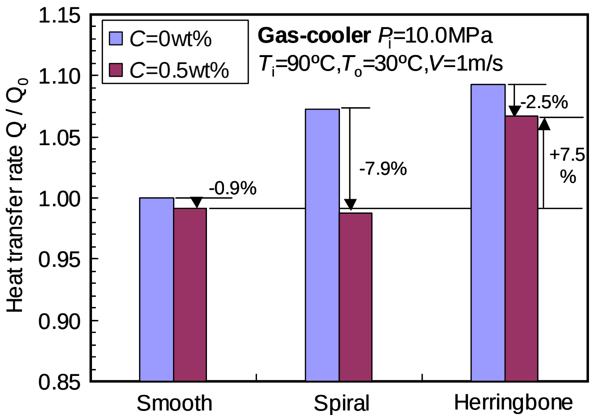


Figure 9: Results of heat transfer rate with and without PAG oil for each tube in gas cooler.

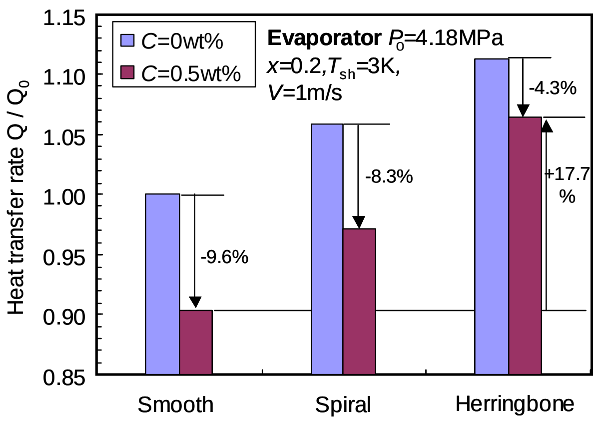


Figure 10: Results of heat transfer rate with and without PAG oil for each tube in evaporator.

4. CONCLUSIONS

The effect of geometries of inner grooved tube on the heat transfer performance was experimentally investigated by using three types of tubes and the heat exchangers assembly using the tubes. From the experimental results through this study, following conclusions can be drawn:

- From the experiments of horizontal heat transfer tubes, it was found that herringbone tubes showed the highest heat transfer coefficient and the least effect of PAG oil. In gas cooler condition, improvement of heat transfer coefficient depends only on surface area expansion rate of inner surface of grooved tubes but geometries. In evaporator condition, improvement by geometries of inner surface can be seen.

- Flow visualization inside tubes reveals that the thicker oil film seen in spiral tubes do not exist in herringbone tubes. This clarifies the effect of geometry of inner surface on oil removal adhere to the wall. The removal of oil can be the key to mitigate the deterioration of heat transfer performance.
- Heat exchangers using herringbone tubes can mitigate the deterioration of heat transfer rate when PAG oil is mixed. Increase in 7.5% and 17.7% of the heat transfer performances were obtained in gas cooler and evaporator compare with smooth tubes, respectively, when OCR is 0.5wt%.

NOMENCLATURE

C	oil circulation rate	(wt%)
G	mass flux	(kg/m^{2}s)
H	heat transfer coefficient	(kW/m^{2}K)
m	mass flow rate	(kg/s)
P	pressure	(MPa)
ΔP	pressure drop	(kPa)
q	heat flux	(kW/m^2)
Q/Q_0	ratio of heat transfer rate	(-)
T	temperature	(ºC)
V	air frontal velocity	(m/s)
x	quality	(-)
V	air frontal velocity	(m/s)

Subscripts

i	inlet
o	outlet
oil	oil
ref	refrigerants
sh	superheat

REFERENCES

ANSI/ASHRAE Standard 41.4-1996 (RA 2006), 2006, Standard method for measurement of proportion of lubricant in liquid refrigerant, *American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.*, Atlanta.

Blasius, H., 1913, Des Aehnlichkeitsgesetz bei Reibungsvorgangen in Flussigkeiten, *VDI Mitt. Forschungsarb.*, no. 131: p. 1-39.

Ebisu, T., Fujino, H., Torikoshi, K., 1998, Heat transfer characteristics and heat exchanger performances for R407C using herringbone heat transfer tube, *International Refrigeration and Air Conditioning Conference*, Purdue University, West Lafayette. IN: paper 434.

Friedel, L., 1979, Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow, *European Two-phase Flow Group Meeting*, Ispra, Italy: paper E2.

Gnielinski, V., 1976, New equations for heat and mass transfer in turbulent pipe and channel flow, *International Chemical Engineering*, vol. 16, no. 2, p. 359-368.

Higashiue, S., Kawahara, K. Yanachi, S., Koyama, S., 2007, Experimental investigation on heat transfer characteristics of supercritical carbon dioxide inside a horizontal micro-fin copper tube during cooling process, *Proc.ASME-JSME Thermal Engineering Summer Heat Transfer Conference*, Vancouver: HT2007-32724.

Hihara, E., Dang, C., 2007, Boiling heat transfer of carbon dioxide in horizontal tubes, *Proc. ASME-JSME Thermal Engineering Summer Heat Transfer Conference*, Vancouver: HT2007-32885.

Kandlikar, S.G., 1990, A general correlation for saturated two phase flow boiling heat transfer inside horizontal and vertical tubes, *ASME J. Heat Transfer*, vol. 112: p. 219-228.

Katsuta, M., Okuma, K., Kishi, S., Miyachi, N., 2007, The evaporating heat transfer characteristics on refrigeration cycle of CO2 in the refrigerator oil mixture state, *Proc. JSRAE Annual Conference*, JSRAE: E103 (In Japanese).

Yoshioka, S., Kim, H., Kasai, K., 2008a, Effect of PAG oil circulation rate on the heat transfer performance of air-cooled heat exchanger in carbon dioxide heat pump system, *Proc. IEA Heat Pump Conference*, Zurich: paper 3.10.

Yoshioka, S., Kim, H., Kasai, K., 2008b, Heat transfer performance and oil behavior for R744 with PAG oil in air-cooled heat exchanger, *Proc. 8th IIR Gustav Lorentzen Conference on Natural Working Fluids*, Copenhagen: paper M4-04.