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Development of a refrigerant to refrigerant heat exchanger for high efficiency CO_2 refrigerant cycle

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ABSTRACT

The cycle performance of CO_2 refrigerant is worse than that of HFCs since the working pressure is higher which suffers larger compression and expansion losses. To enhance the cycle performance similar extent to HFCs, refrigerant to refrigerant heat exchangers need to be applied as one of the many components equipped into the CO_2 cycle. However, when such a heat exchanger is used as an economizer, which works between high temperature refrigerant after the gas cooler and low temperature refrigerant by reducing to the intermediate pressure, it is confirmed from the visualization that the refrigerant in low temperature side is observed as a vapor-liquid dispersed flow. In such a condition, the evaporating performance is low since most liquid flows in the vapor core as droplets.
In this study, aiming at weight saving and compactness for a refrigerant to refrigerant heat exchanger as an economizer, a newly designed aluminum double-tube heat exchanger in which outer channel is formed by multi-ports instead of annulus is introduced. Spiral fins are applied in the surface of inner channel to aim for attaching liquid flowing in vapor core to the inner tube wall. The local heat transfer coefficient was experimentally measured in a single tube and the performance of a heat exchanger assembly was estimated. These results show that this newly designed heat exchanger is effective to enhance heat transfer performance.

1. INTRODUCTION

Improvement of energy efficiency and reduction of environmental load for air-conditioners have been required in recent years. Especially, HFC refrigerants widely used in air-conditioners have high Global Warming Potential (GWP): therefore, their reduction has been targeted by the international agreements.
In terms of reducing environmental impact, R32 is considered the best balanced refrigerant according to the paper by Taira and Haikawa (2014) and globally spreading recently.
For refrigerant candidates to have potential further reducing GWP of air-conditioning systems, CO_2 refrigerant, which is widely used in water heat pump, refrigeration and freezer, is considered. While the refrigerant properties show environmentally pros, cons lie in cycle performance: the performance becomes lower than conventional HFCs due to the increase of discharge temperature in compression process and throttle loss resulting from high pressure difference in expansion process. In order to compensate these losses and achieve similar efficiency with conventional HFCs, the refrigerant cycle with internal heat exchangers is necessary.
In this study, newly designed aluminum double-tube heat exchanger is applied to the one of the internal heat exchangers, an economizer, and the effect is verified by measuring heat transfer coefficient in tube and estimating the performance of heat exchanger assembly in the CO_2 refrigerant cycle.

2. PRACTICAL CONDITIONS AND CONFIGURATIONS OF HEAT EXCHANGER

Figure 1 shows the condition of the economizer and refrigerant temperature distribution used in the CO_2 refrigerant cycle. Refrigerant in high temperature side is supercritical and that in low temperature side experiences two-phase

to superheated vapor. As seeing temperature difference, temperature gradually decreases in high temperature side since the refrigerant is supercritical, while temperature stays constant and then increases in low temperature side since the inflow refrigerant is two-phase and outflow is vapor. When hot refrigerant flows in outer channel and cold refrigerant flows in inner channel, flow pattern of cold refrigerant varies depending on the inlet pressure as shown in Figure 2. While the liquid film flows along the tube wall and vapor and droplets flow in the tube core in the low saturation temperature of the evaporator condition, thinner film flows and more droplets are dispersed in the tube core is observed in the relatively high saturation temperature of the economizer condition. This is mainly caused by the effect of the density ratio between vapor and liquid. The actual ratios are 7:1 in evaporator (Evaporation temperature, T_e = 7°C) and 3:1 in the economizer condition (T_e = 25°C). These ratios are much lower than the conventional HFCs. For example, the ratios of R410A are 16:1 and 29:1, respectively.
Such dispersed flow is significantly ineffective to heat transfer since working liquid is absent on the tube wall. In order to enhance the heat transfer, inner surface geometry having two-type of fins was introduced. The geometry of newly designed aluminum double-tube heat exchanger is shown in Figure 3. Large fins were designed to catch the droplets flow in the core and small fins were to keep the liquid film attached to the tube wall. Detail specifications of the heat transfer tube are tabulated in Table 1. The tube is twisted to setting helix angle, five degree, after extruding the aluminum to be specified cross sectional shape. The thicknesses of the tube wall *a*, *b*, *c* were decided by implementing elasto-plastic analysis to keep high pressure resisting strength.

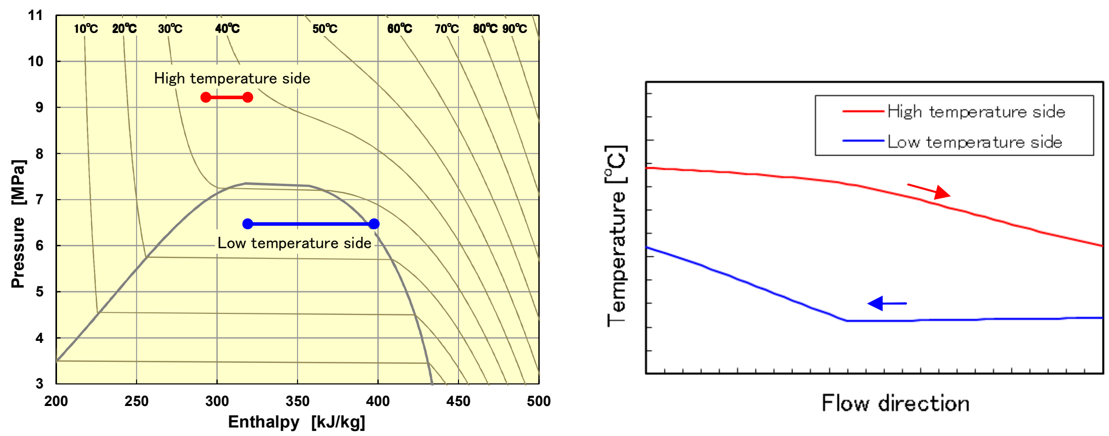
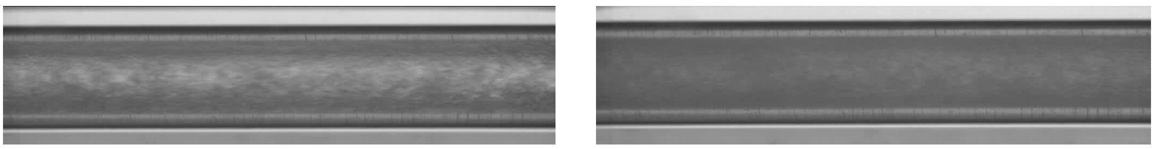


Figure 1: Condition and temperature distribution of an economizer in supercritical CO_2 cycle



(a) (b)

Figure 2: Flow visualization in low temperature side ((a) P = 4.18MPa(T_e= 7°C), G = 1000kg/m^{2}s, x = 0.5, d_t= 4.6mm , (b) P = 6.43MPa(T_e= 25°C), G = 1000kg/m^{2}s, x = 0.5, d_t= 4.6mm)

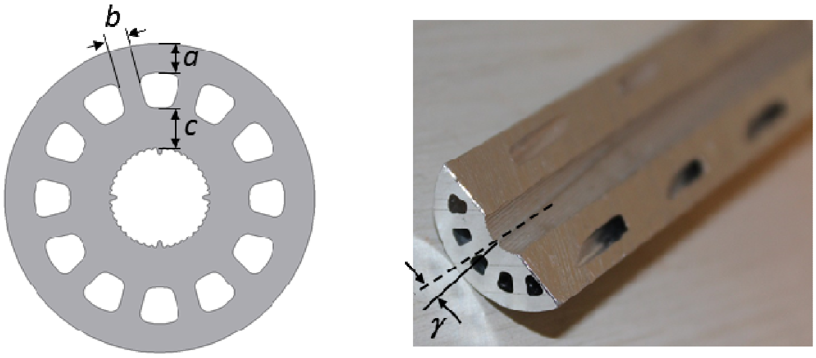


Figure 3: Geometry of newly designed aluminum double-tube heat exchanger

Table 1: Specification of newly designed aluminum double-tube heat exchanger

Outer diameter d_o [mm]	18.0	Number of outer hole N_o	12
Inner base diameter d_i [mm]	5.8	Equivalent outer diameter per hole d_e [mm]	2.51
Fin height (high) H_{fh} [mm]	0.45	Outer thickness a [mm]	1.7
Fin height (low) H_{fl} [mm]	0.15	Beam thickenss b [mm]	1.3
Number of fin N_f	40	Bottom thickness c [mm]	2.1
Helix angle γ [º]	5	Surface area enlargement ratio (inner side) [%]	131

3. EXPERIMENTAL SETUP

Figure 4 shows the schematic diagram of the experimental setup. The refrigerant supply system consists of a compressor, a coriolis mass flow meter, heat exchangers, an expansion valve to control the inlet refrigerant condition to a heat transfer tube such as pressure, temperature, mass flow rate and quality etc. In addition, a separated oil circulation line is located to enable providing lubricant oil within refrigerant flow. PAG (Poly alkyl glycol) was used in this study. The oil circulation rate (OCR) was measured by capturing refrigerant and oil mixture into a sampling tank set in the refrigerant circuit and weighed refrigerant and oil masses, respectively, according to ASHRAE standard (1996). Water flows into the heat exchanger counter-currently to refrigerant flow in order to supply heat source and circulates through a tank by using a pump. Water tank exposed to atmosphere is equipped with heating and chilling unit to keep a constant temperature. Platinum resistance thermometers were used to measure the refrigerant and water temperature. A pressure transducer and a differential pressure transducer were used to measure the pressure of refrigerant-side.
Figure 5 shows the heat transfer tube tested in this experiment. The effective length of heat transfer is 1016mm. Two types of heat transfer tube configuration were tested: One is specified in Figure 3, the other is the conventional tube having smooth inner surface without twist. The heat transfer tube is set horizontally and thermally insulated during the experiment. The experimental conditions are listed in Table 2. The heat flux is calculated based on inner base tube diameter.

To examine the effect of inner channel, the heat transfer coefficient of inner channel h_i is derived as following equations:

$$h_i = \left\{ \frac{1}{K} - \frac{A_i}{h_o A_o} - \frac{(0.5d_i + c)\cdot \ln[(0.5d_i + c)/0.5d_i]}{\lambda_{Al}} \right\}^{-1} \tag{1}$$

where overall heat transfer coefficient K is calculated from heat capacity of water side Q_w:

$$K = \frac{Q_w}{A_i \Delta T_{lm}} \tag{2}$$

In Equation (1), the unknown heat transfer coefficient of outer multi-port channels h_o can be experimentally obtained by implementing water-to-water experiment. At a constant water flow in outer channels, water flowing inner channel varies in several points. When the results plot with the inverse of obtained overall heat resistance as vertical axis and the inverse of inner channel velocity as horizontal axis, the intercept indicates that the heat resistance of inner channel is negligible; therefore, the heat transfer coefficient of outer channel h_o can be derived. This technique is known as Wilson plot method (reference in Shah (1985)).

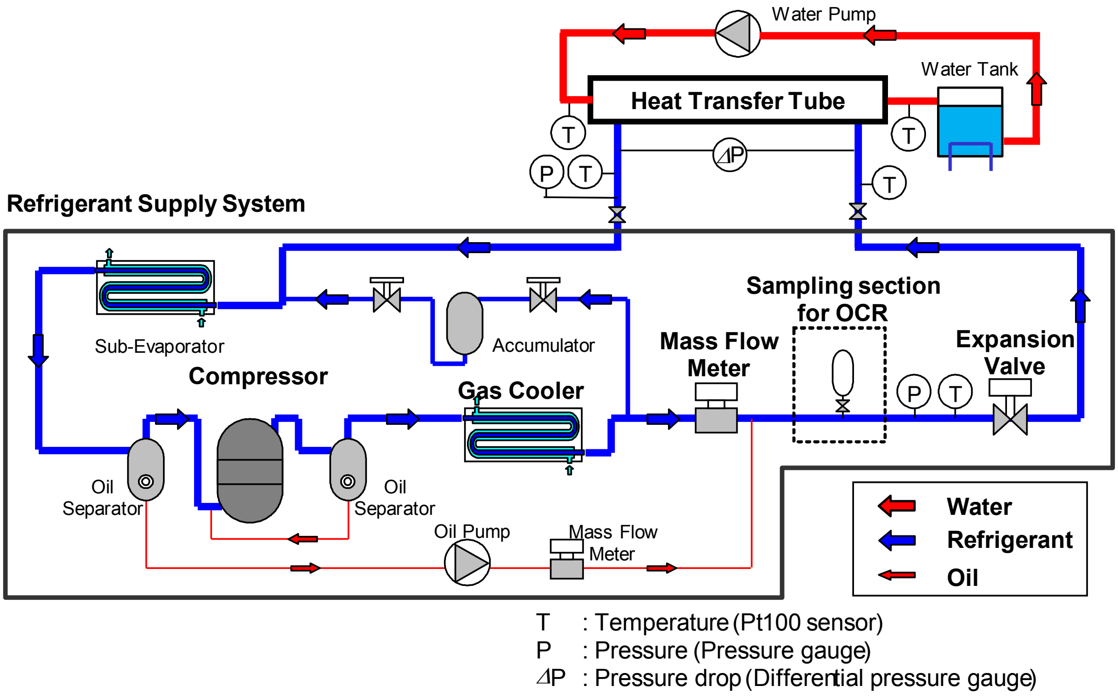


Figure 4: Experimental facility

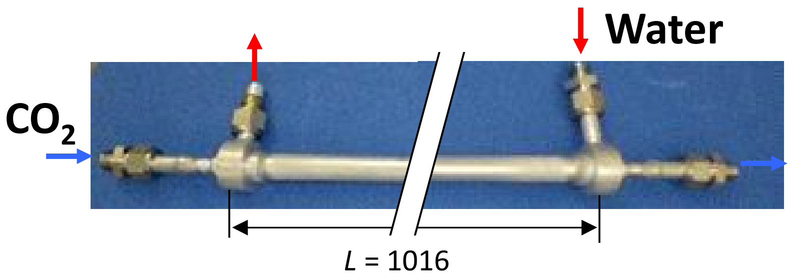


Figure 5: Heat transfer tube tested

Table 2: Experimental conditions

Heat flux Q [kW/m^2]	Mass flux G [kg/m^{2}s]	Evaporation temperature T_e [°C]	Quality [-]	OCR [weight %]
40	980	25	0-1	0, 0.5
40	500	25	0-1	0, 0.5

4. RESULTS AND DISCUSSION

Figure 6 shows the variation of evaporative heat transfer coefficient against quality. In the figure, the tube configuration specified in Figure 3 is referred to as "Spiral", while the conventional tube without twist and inner fins is referred to as "Smooth". It is found that the case of spiral tube without oil gives the highest heat transfer coefficient. The results show that, mostly, the heat transfer coefficient without oil gives higher than that with oil. The same trend was reported by Yoshioka et al (2008a, 2008b). They assume that the presence of oil in the tube wall will affect the heat transfer coefficient since the PAG oil is immiscible to CO_2 refrigerant and investigated the effect of oil retention rate on heat transfer coefficient.
Exceptions are in the middle quality in high flow rate and high quality regions where the results are inverted. In these regions, significant decrease of heat transfer initiates as indicated with dashed lines in the figure. This seems to be caused by the film comes off the inner wall to be dispersed. By adding oil, this phenomenon, referred to as "dryout" in the figure, is mitigated and delayed until higher quality. The effect on the heat transfer coefficient is more significant in higher flow rate since the significant decrease occurs in lower quality.
For smooth tube, comparing the effect of mass flow rate of pure CO_2, the heat transfer coefficient in high flow rate is larger than that in low flow rate in low quality. But this trend is inverted in the middle quality. It is thought that when the flow rate becomes high, the liquid tends to flow in the tube core as droplets: therefore, thinner liquid film on the tube wall causes insufficient liquid supply in evaporating inner surface.
When OCR = 0.5%, which occurs in the real situation in the refrigerant cycle, the heat transfer coefficient of spiral tube is higher than smooth tube in high flow rate. In low flow rate, although the heat transfer coefficient is similar up to middle quality, overall quality is higher than smooth tube since the significant decrease of heat transfer coefficient mitigated in high quality. For example, in the specific condition where quality equals 0.8, the heat transfer coefficient increases by 58% and 145% when G = 980kg/m^{2}s and G = 500kg/m^{2}s, respectively. As considering the surface enlargement ratio 1.31, the heat transfer enhancement by twist and fins is 20% and 87%, respectively.

Figure 7 shows the variation of refrigerant pressure drop against quality. From the figure the pressure drop increases as follows: smooth without oil < smooth with oil < spiral without oil < spiral with oil in order. The results can be well predicted by the homogeneous model based on Blasius's equation. Above results can be predicted by 1.0, 1.6, 2.6 and 4.6 times of homogeneous model, respectively. As the temperature decreases by pressure drop is 10kPa corresponds to 0.06 ^{o}C, the effect of refrigerant pressure drop of inner tube is minor in installing the heat exchanger into the refrigerant cycle.

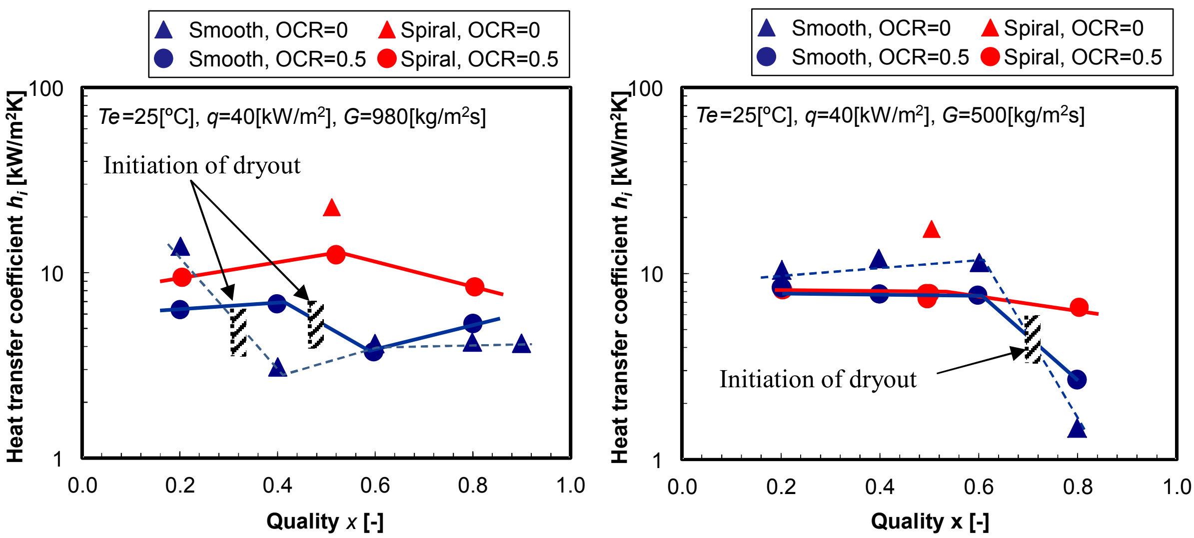


Figure 6: Variation of evaporative heat transfer coefficient with quality

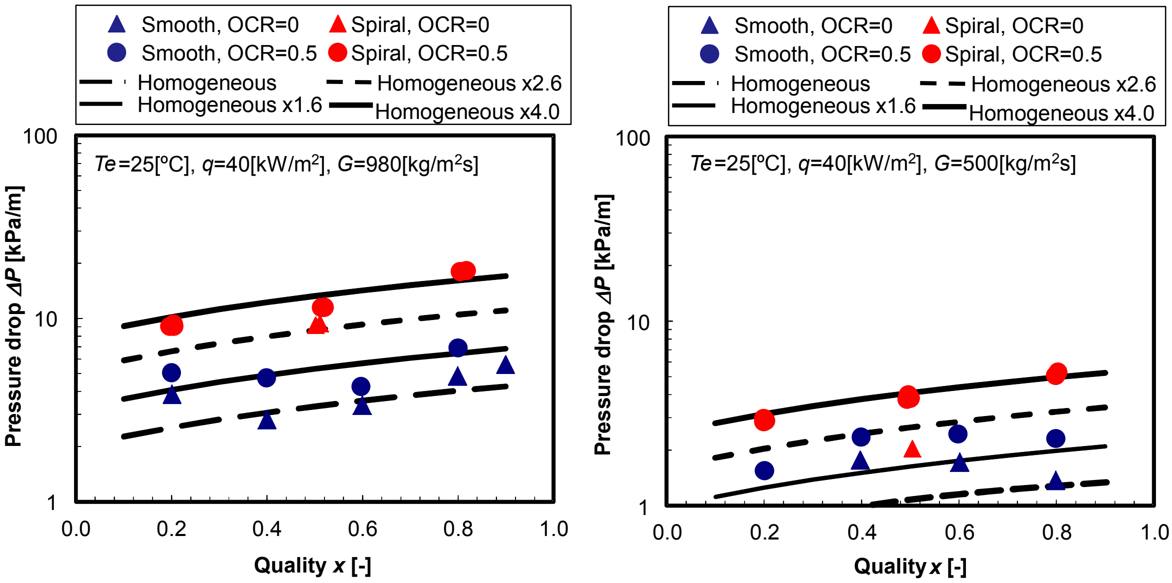


Figure 7: Variation of refrigerant pressure drop with quality

From the above, it is confirmed that the newly designed heat transfer tube with fins and twist is effective to enhance the heat transfer by not only increasing inner surface area but mitigating the dryout, significant decrease of heat transfer, to control the evaporative liquid located to the tube wall surface. It also confirmed that the heat transfer enhances in the condition with including oil, and the higher refrigerant flow is, the more effective heat transfer can be obtained.

5. VARIFICATION WITH HEAT EXCHANGER ASSEMBLY IN THE REFRIGERANT CYCLE

From the above results, a heat exchanger installed in the CO_2 refrigerant cycle was designed as shown in Figure 8 and the effect of newly designed heat exchanger is examined. The estimation was implemented at the economizer condition of rated cooling mode, listed in Table 3. In order to obtain the same heat capacity, the lengths of the two heat exchangers tested in this study were estimated to understand the effect of newly designed heat exchanger based on the experimental results obtained in previous section. For estimation, the length of heat exchanger is divided into 50 segments and the heat capacity of each segment is calculated by using two-phase heat transfer coefficient obtained in previous section and the heat transfer coefficient of vapor phase and high temperature side which are used the equation of single phase flow. The result is shown in Figure9. From the figure, it is found that the new type heat exchanger can reduce the size to 23%.

Figure 8: Heat exchanger configuration for installing the refrigerant cycle

Table 3: The condition of the economizer in the refrigerant cycle (Rated cooling mode)

High temperature side	
Inlet pressure	9.36 [MPa]
Inlet temperature	38.0 [ºC]
Mass flow rate	294 [kg/hr]
Low temperature side	
Saturation temperature	30.0[ºC]
Mass flow rate	97 [kg/hr]
Heat capacity	2.58 [kW]

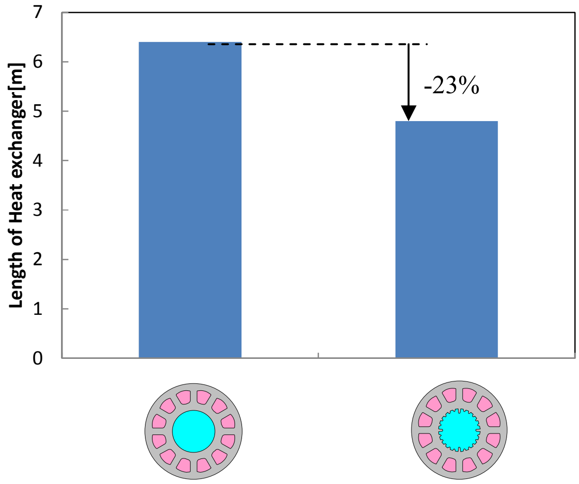


Figure 9: The effect of the newly designed heat exchanger with the same capacity estimated

6. CONCLUSIONS

For a refrigerant-to-refrigerant heat exchanger used as an economizer in the CO_2 refrigerant cycle for air-conditioner, a newly designed aluminum double-tube heat exchanger in which outer channel is formed surrounding multi-ports instead of outside annulus is proposed to aiming at weight saving and compactness.
It can be confirmed that the proposed inner surface geometry enhances the heat transfer and the effect is verified experimentally by measuring heat transfer tube and estimating the performance of the heat exchanger assembly in the refrigerant cycle.

NOMENCLATURE

A	surface area	(m^2)
a	outer thickness	(mm)
b	beam thickness	(mm)
c	bottom thickness	(mm)
d	diameter	(mm)
G	mass flux	(kg/m^2s)
h	heat transfer coefficient	(W/m^2K)
K	heat transfer rate	(W/m^2K)
L	length	(mm)
N	number	(-)
OCR	oil circulation ratio	(w%)
Q	heat capacity	(W)
q	heat flux	(W/m^2)

T_e	evaporation temperature	(^oC)
ΔP	pressure drop per length	(kPa/m)
ΔT_{lm}	log mean temperature	(K)
γ	helix angle	(^{o})
λ	thermal conductivity	(W/mK)

Subscript

Al	aluminum
f	fin
fh	high fin
fi	low fin
i	inner
o	outer
t	tube
w	water

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