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An Experimentally Validated Model for Microchannel Heat Exchanger Incorporating Lubricant Effect

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ABSTRACT

In this study, a microchannel heat exchanger model is developed in which the thermodynamic and transport properties of refrigerant-oil mixture are taken into account as well as their impact on boiling heat transfer and pressure drop. A newly proposed infrared thermography based method is implemented in the model to describe the quality distribution in the inlet header instead of an empirical correlation. The new model is validated against experimental results (R134a-PAG 46 oil) at various oil circulation rates (0.1%-8.3%). The agreement between measurement and prediction is ±4% for capacity, ±10% for pressure drop and ±3℃ for superheat at compressor inlet. Simulation results also indicate that lubricant addition improves refrigerant distribution and the infrared thermography based method enables the model to fully capture lubricant effect on capacity.

1. INTRODUCTION

Lubricants in a vapor compression refrigeration system are mainly used to reduce friction and minimize wear of the compressor. In automobile air conditioning systems, where oil separators are less commonly used, the lubricant pumped out of the compressor is mixed with refrigerant and travels throughout the system. Adverse effects of lubricating oil on capacity and COP has been identified experimentally by many researchers (McMullan et al., 1988; McMullan et al., 1992; Lottin et al., 2003; DeAngelis and Hrnjak, 2005). These adverse effects can also be predicted by numerical models (Youbi-Idrissi et al., 2003; Lottin et al., 2003).

The fundamental reason for the above mentioned lubricant effects is the changes of thermodynamic and transport properties due to oil addition. Many studies have been performed to investigate the refrigerant-oil properties including solubility, viscosity, density surface tension, heat capacity, thermal conductivity and etc. for numerous refrigerant and oil combinations (Filippov and Novoselova, 1955; Liley and Gambill, 1973; Jensen and Jackman, 1984; Baustian, 1986; Conde, 1996; Seeton and Hrnjak, 2009).

In a microchannel heat exchanger manifold, due to the properties changes, especially the increasing surface tension caused by oil addition, the flow regime become significantly different with that of pure refrigerant. A comprehensive review of the flow regime visualization of the refrigerant-oil mixtures has been done by Filho et al.(2008). After summarizing numerous studies, they concluded that the transition from stratified wavy regime to annular flow regime of refrigerant-oil mixture happened earlier than pure refrigerant. Foaming was identified in most of the studies with different quantity and the authors believed that foaming characteristics were related to particular physical properties of each refrigerant and lubricant pair. Bowers and Hrnjak (2010) illustrated the flow of R134a and 1.7% (OCR) PAG oil (46 cSt at 40 oC) in a horizontal tube (ID=8.7 mm) .They discover that the flow regime became Stratified/Annular when it is Stratified/Wavy for pure refrigerant and the addition of oil created numerous droplets in the vapor and lots of bubbles or froth in the liquid layer. Zou and Hrnjak (2013) observed the flow regime of R134a and PAG46 oil mixture in a vertical microchannel manifold. Foam was formed after a large amount of oil (2.5% and 4.7%) was introduced, making the flow regime more homogeneous.

Apart from the manifold, oil effect can also take place in the parallel tubes in a microchannel heat exchanger. The change of the thermodynamic and transport properties will affect the heat transfer and pressure drop characteristics of the working fluid. Several critical reviews of oil effect on refrigerant-side heat transfer and pressure drop could be found in the literature (Thome,1995; Thome,1999; Mermond et al.,1999; Shen and Groll, 2005a; Shen and Groll, 2005b; Thome et al. 2008; Youbi-Idrissi and Bonjour, 2008).All studies identified that oil addition increased refrigerant-side pressure drop. But regarding to the influence of oil on heat transfer, no agreement has been reached. Most of the studies reviewed in the aforementioned four papers used conventional tubes and very little information could be obtained with respect to oil effect in microchannel. Burr and Hrnjak (2005) visualized the flow regime of R134a and POE oil mixture of high quality (0.75-0.95) in parallel microchannels (1.54mm hydraulic diameter). They found that most of the flow observed had the separated regime and the maldistribution was exacerbated by oil addition. Through experimental measurements, the authors also discovered that the pressure drop was increased and the void fraction was depressed by the presence of lubricant. Field and Hrnjak (2007) observed the flow regime of R134a and POE68 oil with different concentration in a transparent microchannel tube (0.5 mm hydraulic diameter). Bubble-slug, slug, slug-annular and annular flow regimes were identified and the transition between bubble-slug and slug flow were found to be affected by oil addition. Increasing viscosities and concentrations of the lubricant were shown to elevate the pressure drop of refrigerant-oil mixture. In the end, the authors developed a mechanistic model which could be used to predict the pressure drop of pure refrigerant and refrigerant-oil mixture.

Refrigerant maldistribution is a severe issue for microchannel heat exchanger which can greatly decrease the thermal performance of the heat exchanger, especially for evaporator. The existence of lubricant can affect distribution as well which is a consequence of combined oil effect in the manifold and oil effect in microchannel tubes. DeAngelis and Hrnjak (2005) experimentally studied the oil effect on small R744 system which has a microchannel evaporator. They found that decreasing viscosity is beneficial for capacity and COP, but OCR effect was not clear to them at that time. Zou and Hrnjak (2012) investigated oil effect on refrigerant distribution in vertical inlet headers. At 0.5% oil concentration, distribution was found to become worse, which might be due to the significant increase of viscosity that creates difficulty for the working fluid to reach the top tubes. While at 2.5% and 4.7% oil concentration, distribution became improved which owe to the large amount of foams making the flow regime more homogeneous.

The aforementioned studies are mainly experimental demonstration of the importance of lubricant effect. Many attempts have been made to take account of the effect using the modeling approach. According to Thome's studies (1995a, 1995b, 1999), there were two typical ways to study oil effect. The first one treated oil as the contamination and used a correction factor to take lubricant influence into account. The second one considered the working fluids as refrigerant-oil zeotropic mixture instead of pure refrigerant and mixture properties were used in the model. The second approach was believed to be thermodynamically right. And it was more generalized, having potential to be used for any miscible refrigerant-lubricant combinations. This thermodynamic approach has been adopted by many researchers. Youbi-Idrissi et al. (2003) developed a simplified thermodynamic model of enthalpy calculation for the refrigerant and oil mixture and showed that higher OCR would result in higher non-evaporated quantity of refrigerant thus lower specific enthalpy difference in the evaporator. This model has been experimental validated by Youbi-Idrissi et al. (2004). Lottin et al. (2003) built a numerical model to simulate the consequences of oil addition on refrigeration system performance and found that oil effects were negligible when OCR was less than 0.5, but beyond that value, the system performance could be significantly decreased. Lottin et al. (2003) modeled the oil effects on evaporator and condenser and predicted that oil addition generally decreased the performance of heat exchanger but optimum performance of the evaporate was observed at 0.1% OCR.

The evaporator type used in Youbi-Idrissi et al's studies is shell and tube while it is plate heat exchanger in Lottin et al's research. There are very few studies including oil effect in the microchannel heat exchanger model. Li and Hrnjak (2013) incorporated the lubricant effect using the thermodynamic approach in a microchannel heat exchanger. The lubricant influences on pressure drop, heat transfer, distribution were all included in the model. This paper presents the improvement of the model by including a newly proposed infrared (IR) thermography based method. The new model is validated against experimental data. Effect of Lubricant on distribution and evaporator capacity is studied based on the model.

2. EXPERIMENT DESCRIPTION

The experimental data used in this study were taken from a Mobile Air Conditioning (MAC) test facility. The schematic drawing of the facility is shown in Figure 1. The variable speed compressor and the microchannel condenser are realistic components which are used in a major brand vehicle. The microchannel evaporator has a single pass and single slab which is specially designed for the experimental purpose. Details of the experimental facility including component geometry and measurement uncertainty can be found in Tuo and Hrnjak (2012)'s paper.

Experiments uses R134a as the refrigerant and PAG46 as the lubricant. The Oil Circulation Rate (OCR) in the test varies from 0.1% to 8.3%. Oil sampling method is employed to measure the OCR. The indoor/outdoor ambient temperature is fixed at 35℃/35℃, the superheat at compressor inlet is maintained at 15℃ and the subcooling at condenser outlet is kept at 10℃.

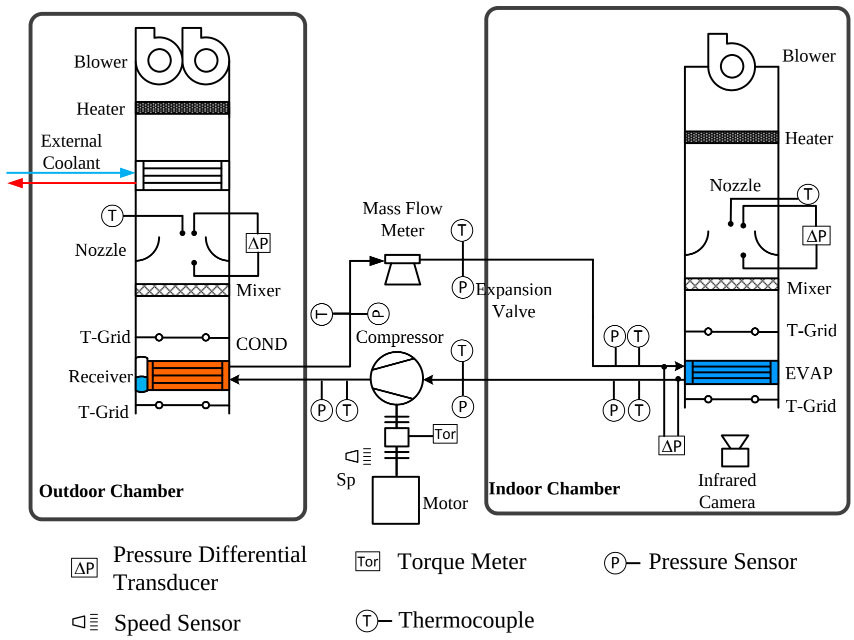

Figure 1: Schematic drawing of the test facility

3. MODELING DESCRIPTION

The microchannel heat exchanger model developed by the same authors (Li and Hrnjak, 2013) is used in this study. Refrigerant-oil zeotropic mixture is treated as the working fluid. In each microchannel tube, mixture properties are incorporated in the heat transfer and pressure drop correlations, thus the oil effect on the flow resistant within each tube is implicitly counted. In Li and Hrnjak's model, the two-phase refrigerant distribution in the inlet header is originally described by an empirical quality distribution function. Among available distribution functions, we have initially chosen Jin (2006) (modified by Tuo et al. (2012)) which assigns the inlet quality for each microchannel tube. Since this distribution function was mainly developed for pure refrigerant, the lubricant effect in the inlet header is not fully taken into account at this point.

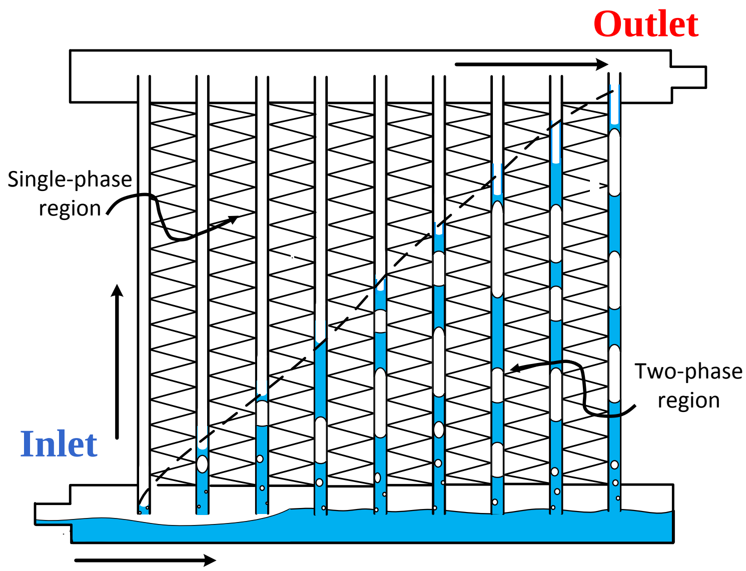

Figure 2: Snapshot of a simplified situation in a parallel flow heat exchanger

This model can be further modified by replacing the existing quality distribution function with an IR based quantification method (Li and Hrnjak, 2014 Purdue conference, paper 2147). The refrigerant inside of microchannel tubes could be either two-phase or superheat vapor as shown in Figure 2. Because the refrigerant-side capacity in the two-phase region is mainly the latent heat of the liquid refrigerant, the main idea of this IR quantification method is to correlate the liquid mass flow rate through each individual tube with the corresponding air-side capacity in the two-phase region. The ratio of the liquid mass flow rates through any two microchannel tubes can be formulated as follow:

$$\frac{(\dot{m}_{ref,liq,tube})_i}{(\dot{m}_{ref,liq,tube})_j} = \frac{(\dot{m}_{ref,liq,tube})_i h_{fg}}{(\dot{m}_{ref,liq,tube})_j h_{fg}} = \frac{\left(\sum_1^{n_i} \varepsilon C_{\min}(T_{\text{air},in} - T_{wall,element})\right)_i}{\left(\sum_1^{n_j} \varepsilon C_{\min}(T_{\text{air},in} - T_{wall,element})\right)_j} = \frac{\left(\sum_1^{n_i} (T_{\text{air},in} - T_{wall,element})\right)_i}{\left(\sum_1^{n_j} (T_{\text{air},in} - T_{wall,element})\right)_j}$$

（Each tube is divided into elements, n is the last element in the two phase region）

By assuming uniform temperature and velocity profile of the incoming air, ε and $C_{\min}$ can be eliminated, because they are constant for each element. Knowing the evaporator air inlet temperature measured by thermocouple grid and heat exchanger surface temperature measured by infrared camera, the liquid refrigerant distribution can be fully described. Then the distribution of the vapor refrigerant will be determined by the mass conservation of both phases as well as the pressure drop equality relationship of each flow path (details are introduced in Li and Hrnjak, 2014 Purdue conference, paper 2147).

By using the IR based quantification method, oil effect on distribution including the influence in the inlet header as well as in the tubes is fully captured in the infrared image and then transferred into the model.

4. VALIDATOIN

Simulation results including capacity, pressure drop and superheat are validated against experimental data. Normally models using pure refrigerant either do not consider the existence of lubricant or modify the mass flow rate by multiplying (1-OCR). The modeling approach incorporating lubricant effect is compared with two above-mentioned pure refrigerant models and superiority of the oil approach is demonstrated.

Figure 3 illustrate the capacity validation of two pure refrigerant models and the refrigerant-oil mixture model against experimental results. The capacity prediction of the model with lubricant effect has a root-mean-square-deviation (RMSD) error of 5.5%. While the pure refrigerant models with and without modifying the mass flow rate by subtracting OCR have RMSD of 13.4% and 20.7%. They have tendency of over predicting the capacity (more than 8%) when OCR is higher than 5%.

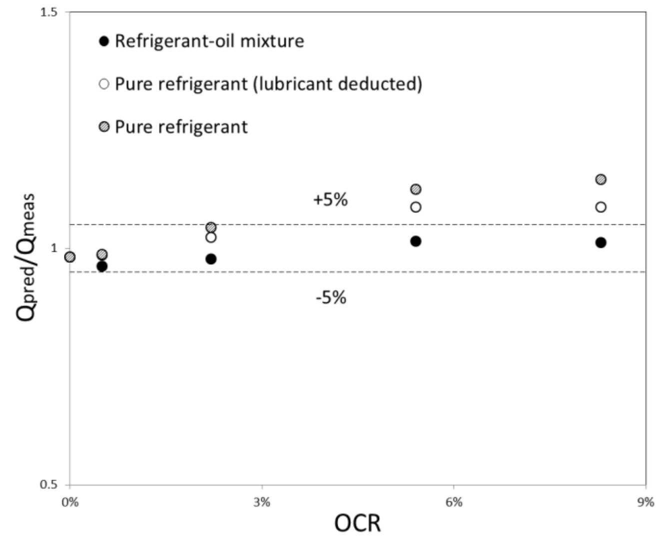

Figure 3: Capacity validation of three different modeling approaches

The pressure drop validation of three different approaches is demonstrated in Figure 4. It can be seen that the lowest RMSD is achieved in the lubricant model and most of the prediction is within 10% agreement with the experimental data. Two pure refrigerant models tend to underpredict the pressure drop especially under high OCRs.

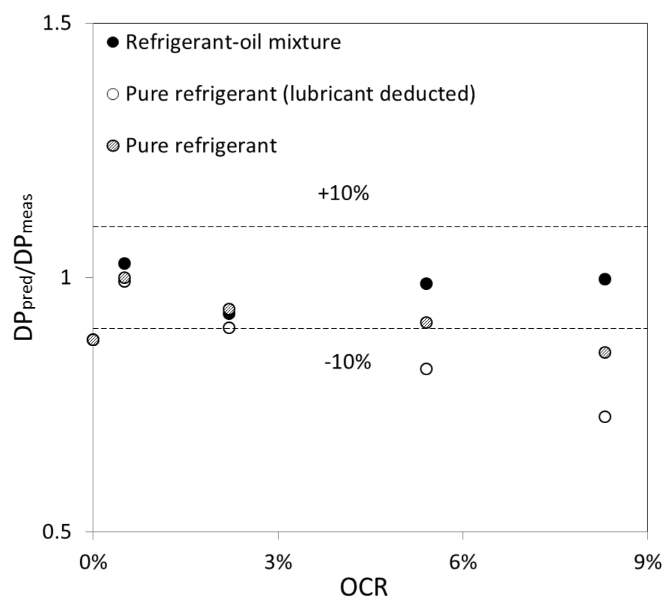

Figure 4: Pressure drop validation of three different modeling approaches

The superheat prediction of three approaches is also validated against experimental data which is shown in Figure 5. There is no obvious superiority of model incorporating lubricant effect than pure refrigerant model in superheat prediction. The RMSD error of the model considering lubricant is 3.01℃.

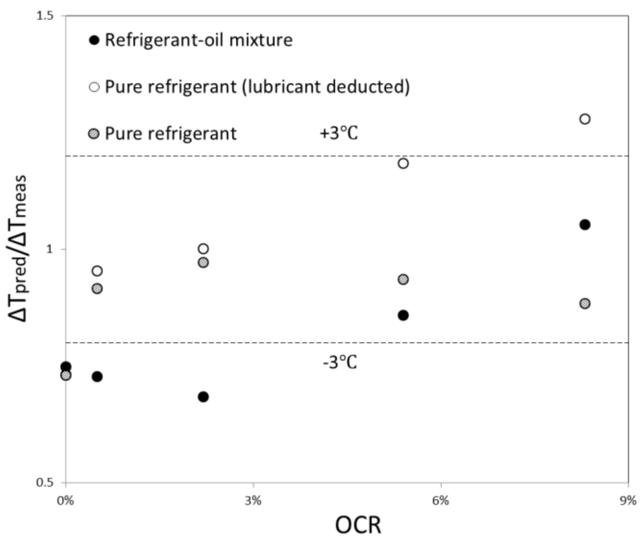

Figure 5: Superheat validation of three different modeling approaches

5. DISCUSSION

5.1 Lubricant Effect on Distribution

Using the IR quantification method introduced in section 3, the mass flow rate of liquid refrigerant-oil mixture through each microchannel tube is quantified and then distribution is rated by the coefficient of variance defined as the following:

$$\sigma = \frac{1}{\dot{m}_l}\sqrt{\frac{1}{n}\sum_{1}^{n}(\dot{m}_{l,i} - \overline{\dot{m}_l})} \tag{11}$$

It can be seen from Figure 6 that the addition of lubricant improves refrigerant distribution since the coefficient of variance decreases.

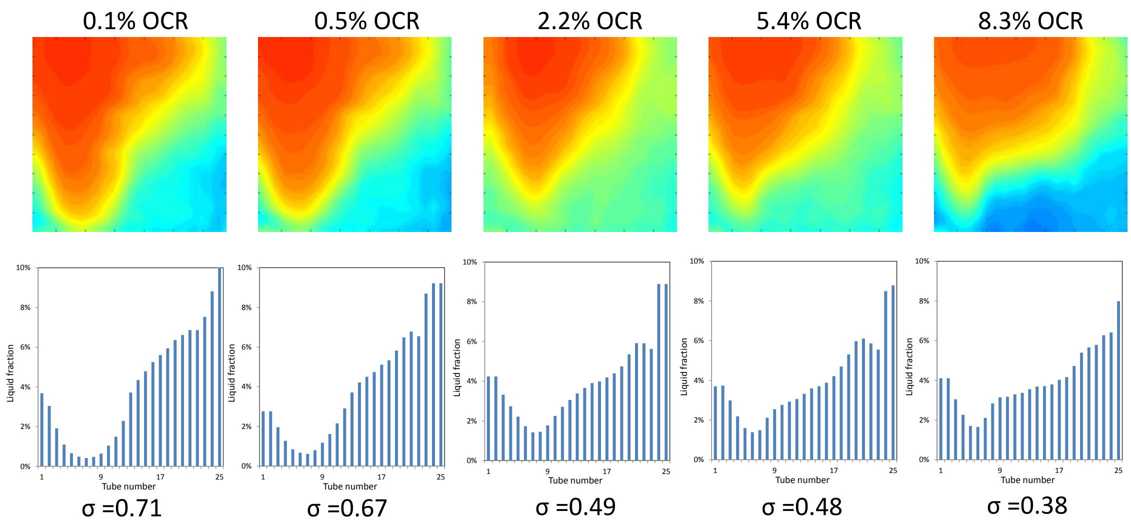

Figure 6: Effect of lubricant on distribution

5.2 Lubricant Effect on Capacity

The improvement of distribution creates potential for capacity increase. It is witnessed in the experiment that the capacity spikes around 2% OCR. The detailed interpretation of the experimental results can be found in Li and Hrnjak (2014a, 2014b). It can be seen from Figure 7 that the simulation results from the model utilizing the IR quantification method precisely capture the trends of the capacity variation corresponding to OCR increase; while the model using empirical distribution function (modified Jin's function) fails to do so, because the lubricant effect on distribution in the inlet header is not correctly considered.

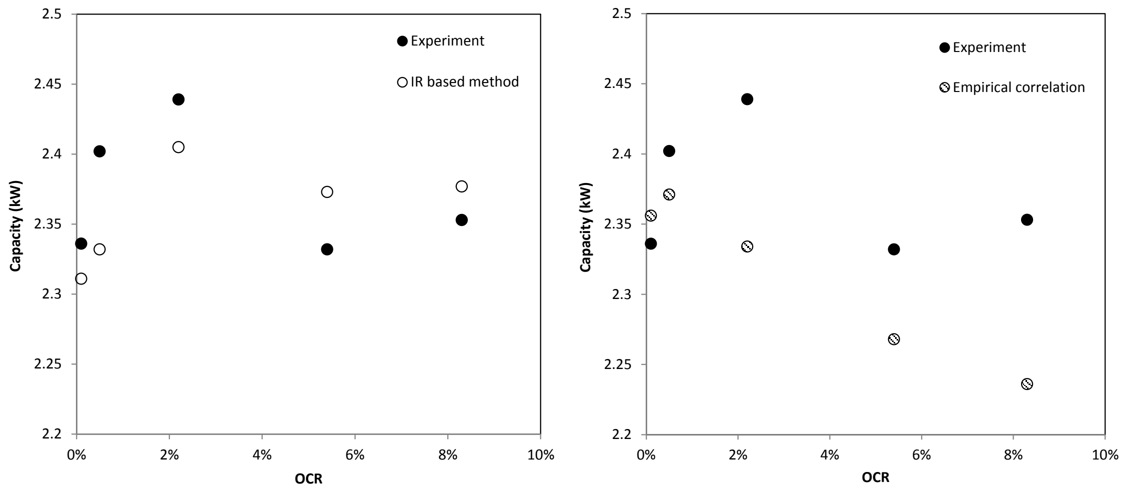

Figure 7: IR based quantification method enable the heat exchanger model capture lubricant effect on capacity more precisely

6. CONCLUSION

In this study, Li and Hrnjak's (2013) model is enhanced by the inclusion of an IR based quantification method. The new model is validated against experimental data, showing superiority over models using pure refrigerant in capacity and pressure drop prediction.

Lubricant effect on distribution is quantified using the IR quantification method. It is found that the addition of oil is beneficial for distribution. The inclusion of the IR method enables the model capturing the effect of lubricant more comprehensively (include the lubricant effect in the inlet header and tubes) thus predict the capacity more precisely.

NOMENCLATURE

C	heat capacity	($kJkg^{-1}K^{-1}$)
ε	heat exchanger effectiveness	(–)
h	specific enthalpy	($kJkg^{-1}$)
$\dot{m}$	mass flow rate	(kgs^{-1})
OCR	Oil Circulation Rate	(–)
T	temperature	(K)
σ	coefficient of variance	(–)
Subscript		
air	air side	
element	element	
f	fluid	
g	gas	
in	inlet	
l/liq	liquid	

min	minimum
ref	refrigerant
tube	tube

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