Quantification of Liquid Refrigerant Distribution in Parallel Flow Microchannel Heat Exchanger Using Infrared Thermography

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ABSTRACT

This paper presents a method to quantify the distribution of liquid refrigerant mass flow rate in parallel flow microchannel heat exchanger from the infrared images. Quantification is achieved by building the relationship between the liquid mass flow rate through each microchannel tube and the air side capacity calculated from the infrared measurement of the wall temperature. After being implemented in a heat exchanger model, the quantification method is validated against experimental data. This method can be used for several types of heat exchangers: evaporators, condensers etc., also it can be applied to various heat exchanger designs: different inlet/outlet locations, different flow configurations etc.

1. INTRODUCTION

Microchannel heat exchangers have been widely used in air conditioning and refrigeration systems because of their compactness and enhancement for heat transfer performance. However microchannel heat exchangers suffer from the problem of refrigerant maldistribution, especially for evaporator. The none-uniform distribution of two phase refrigerant in evaporator can significantly deteriorate the heat exchanger performance as well as system efficiency. Many simulation results shows that the capacity degradation due to refrigerant maldistribution ranges from 5% to 23% (Kulkarni *et al*., 2004; Brix *et al*. 2009, 2010; Zou *et al*. 2012).

A good quantification of two phase refrigerant distribution is essential for evaluating maldistribution effect on heat exchanger and system performance. Many studies have been done to quantify refrigerant distribution using some intrusive techniques. The main idea is to measure the mass flow rate of liquid and vapor phase through each microchannel tube which branches from a realistic heat exchanger header. Vist (2003) investigated the distribution of two-phase CO_2 and R134a in horizontal heat exchanger headers for upward and downward flow configuration. Strong effect on distribution of vapor fraction at header inlet has been seen in the experiments. An empirical model was developed to describe phase split in the header. Lee *et al*. (2004) examined the distribution of air-water two-phase annular flow in a vertical heat exchanger header. The effect of tube protrusion was examined and the most uniform distribution could be achieved by adjusting the depth of protrusion. Hwang *et al*. (2007) studied the distribution of R410A and R134a in horizontal heat exchanger headers. They concluded that the refrigerant distribution was largely affected by header inlet location and vapor mass flux. A set of empirical correlations for refrigerant distribution were developed. Lee (2009) investigated the distribution of air-water two phase flow in a vertical header. The distribution at the front part of the header was found to be only affected by the upstream flow regime, while the distribution at the rear part of the header was mainly influenced by the downstream flow configuration due to flow recirculation. The existing T-junction phase split models were found to be working well for the front part of the header but not the rear part. Ahmad *et al*. studied the two phase HFE 7100 distribution in a horizontal header including both upward and downward flows. The effects of numerous operational and geometrical parameters on distribution were investigated and a map of flow regime in the header was developed. Kim *et al*. (2011) examined the effect of inlet location on two phase R134a distribution in a horizontal header. Parallel inlet configuration was found to have the worst performance, while normal and vertical inlet configuration yield similar refrigerant distribution. Empirical distribution correlations for all three inlet configurations were formulated, which were functions of Reynolds number of the upstream gas in the header. Zou and Hrnjak (2013a, 2013b) performed experimental studies for two phase R134a and R410A distribution in vertical heat exchanger headers. They found that the best distribution was achieved at high flux and low quality. Empirical distribution correlations were developed for both fluids. The intrusive method could provide accurate information of distribution which are the liquid and vapor mass flow rates through each microchannel tube, but none of the aforementioned experiment setups used real microchannel heat exchangers. Some of the experiments were conducted under adiabatic condition. Even for the diabatic cases, it was extremely hard to represent the heat load of each microchannel in a real heat exchanger; also some of the experimental facilities didn't have outlet header, so the outlet header induced maldistribution, which was proved to be very important (Tuo and Hrnjak, 2013), was ignored.

Many attempts have been made to quantify two-phase refrigerant distribution through non-intrusive methods. Litch and Hrnjak (1999) related the air exit temperature of a microchannel condenser with the inside refrigerant distribution. Song *et al*. (2002) used the frosting pattern to identify maldistribution in an outdoor microchannel coil. Infrared (IR) thermography has also been used by many researchers to examine refrigerant distribution, but only qualitative information was obtained (Litch and Hrnjak, 1999; Dschida and Hrnjak, 2008; Qi *et al*., 2009; Shi *et al*., 2011; Tuo and Hrnjak, 2012). Bower *et al*. (2010) proposed a distribution rating parameter which is based on the uniformity of the surface temperature of the heat exchanger. Some quantification of distribution could be acquired from the IR image by using this parameter. Bower *et al*'s method have been adopted by many authors (Tuo and Hrnjak, 2012; Li and Hrnjak, 2013).

Non-intrusive method has the advantage of saving time and cost over the intrusive method, but typically uncertain quantitative information about refrigerant distribution is obtained. In this study, an IR thermography based method is proposed to quantitatively estimate liquid refrigerant distribution in parallel flow microchannel heat exchangers. This method is validated against experimental results in conjunction with the heat exchanger model.

2. METHOD DEVELOPMENT

2.1 Heat transfer basis for the quantification method

The cooling capacity of an evaporator comes mainly from the latent heat of the liquid refrigerant. So distribution of liquid is more important than vapor. The refrigerant inside of tubes could be either two phase or superheat vapor as shown in Figure 1.

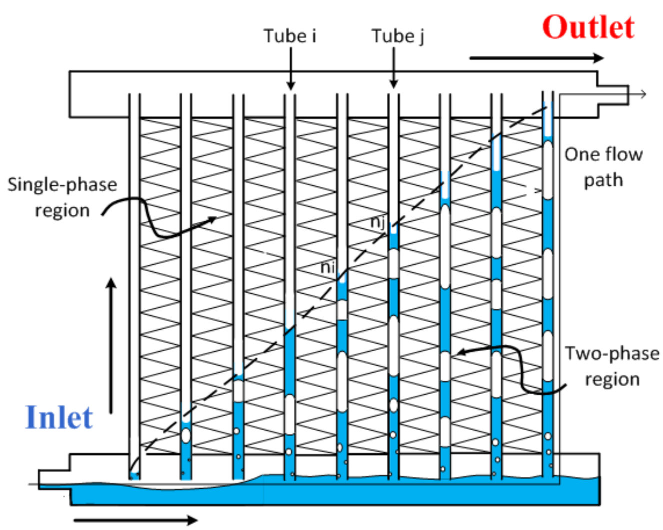

Figure 1: Snapshot of a simplified situation in parallel flow heat exchanger

From the modeling perspective (finite volume approach), a heat exchanger has several parallel microchannel tubes and each tube is divided into elements. The following assumptions are made to simplify the model without losing generality: 1) Uniform temperature and velocity profile of the incoming air; 2) Uniform distribution of refrigerant among parallel ports within the same tube; 3) Uniform wall temperature within each element; 4) No conduction through the fins or wall along tube direction.

For each element in the two-phase region, because of the uniform wall temperature, the energy balance on the air side can be calculated using the ε -NTU method:

$$Q_{air,element,tp} = \varepsilon C_{\min}(T_{air,in} - T_{wall,element}) \tag{1}$$

$$C_{\min} = c_{p,air}\dot{m}_{\mathrm{air}} \tag{2}$$

$$\varepsilon = 1 - \exp(-NTU) \tag{3}$$

$$NTU = \frac{h_{air}A_{air}}{C_{\min}} \tag{4}$$

Assuming uniform temperature and velocity profile of the incoming air, A_{air} , $T_{air,in}$, $\dot{m}_{\mathrm{air}}$, $c_{p,air}$ and h_{air} stays constant for each element, ε and $C_{\min}$ is also constant.

In the two-phase region, the refrigerant-side capacity is the latent heat of the liquid refrigerant, so the following energy balance equation can be formed in each element:

$$Q_{ref,element,tp} = \Delta\dot{m}_{ref,liq,element}\, i_{fg} \tag{5}$$

For each microchannel tube, refrigerant side capacity in the two phase region is calculated by summing up the capacity of each element together:

$$Q_{ref,tube,tp} = i_{fg}\sum_{1}^{n}\Delta\dot{m}_{ref,liq,element} = i_{fg}\,\dot{m}_{ref,liq,tube} \tag{6}$$

（n is the last element of the two phase region）

Based on energy balance between refrigerant-side and air-side,

$$Q_{ref,tube,tp} = \sum_{1}^{n} Q_{air,element} = \varepsilon C_{\min}\sum_{1}^{n}(T_{air,in} - T_{wall,element}) \tag{7}$$

the ratio of liquid refrigerant mass flow rate through any two microchannel tubes can be calculated as following:

$$\frac{(\dot{m}_{ref,liq,tube})_i}{(\dot{m}_{ref,liq,tube})_j} = \frac{\left(\sum_{1}^{n_i}(T_{\mathrm{air},in} - T_{wall,element})\right)_i}{\left(\sum_{1}^{n_j}(T_{\mathrm{air},in} - T_{wall,element})\right)_j} \tag{8}$$

Given the above ratio, liquid refrigerant distribution among parallel microchannel tubes can be fully described by the following equation:

$$\frac{(\dot{m}_{ref,liq,tube})_i}{\dot{m}_{ref,liq,HX}} = \frac{\left(\sum_{1}^{n_i}(T_{\mathrm{air},in} - T_{wall,element})\right)_i}{\sum_{1}^{\mathrm{tube\#}}\sum_{1}^{n_j}(T_{\mathrm{air},in} - T_{wall,element})} \tag{9}$$

IR camera provides the wall temperature of each element. As long as the boundary between single phase region and two phase region can be identified on the IR image of the heat exchanger (see section 2.2), the distribution of liquid refrigerant mass flow rate can be then determined.

2.2 Procedure of applying the quantification method

First, IR image is smoothed using an algorithm for total variation minimization. The difference between the IR images before and after smoothing can be seen in Figure 2. Temperature scale is the same in the smoothed image as in the raw IR image which is adjusted to be narrowest possible.

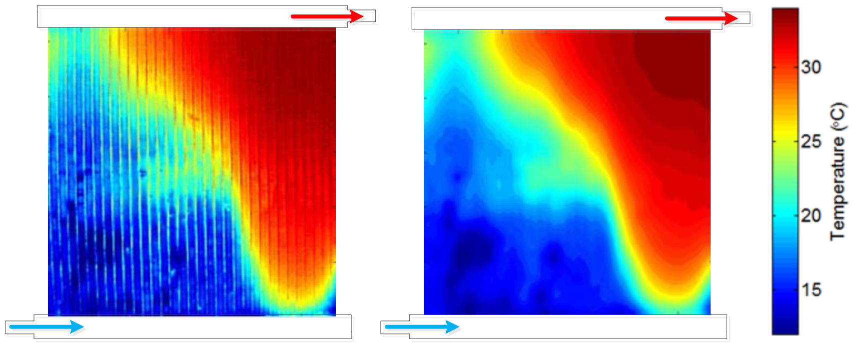

Figure 2: Comparison of IR image before and after smoothing

After obtaining the smoothened IR image, a transition line is drawn to identify the two-phase zone. In each tube with liquid at the inlet and superheated exit, there will be a certain location at which the last drop of liquid refrigerant evaporates. The element which starts from that location is defined as the transition element. It is assumed to have saturated single phase vapor refrigerant at the inlet. When heat exchanger model is not available, the pressure at the transition element can be estimated as the average of evaporator inlet and outlet pressures, also average mass flow rate which equals to the total mass flow rate divided by the tube number is used to calculate the refrigerant-side heat transfer coefficient.

$$\dot{m}_1 = \dot{m}_2 = \ldots = \dot{m}_i = \ldots = \dot{m}_n \tag{9}$$

$$\dot{m}_i = \dot{m}_{l,i} + \dot{m}_{v,i} \tag{10}$$

Based on the air-side and refrigerant-side thermal resistances, the wall temperature of the transition element in each tube can be then determined. The locations with the calculated temperature in the IR image define the transition line (red line in Figure 4).

When higher calculation accuracy is required, the mass flow rate at each transition location can be determined through an iterative process using heat exchanger models. In the first iteration, average mass flow rate which is corresponding to uniform distribution case is used to calculate the transition temperature. The resultant transition line will be used to determine liquid mass flow rate through each tube using the IR quantification method introduced in section 2.1. In the heat exchanger model (Li and Hrnjak (2013) is chosen in this study), the working fluid flows along different paths, each of them consisting of one microchannel tube and the corresponding distances in the inlet and outlet header as shown in Figure 3. Because each path starts from the evaporator inlet and ends at the outlet, the pressure drop should be the same. 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. This quality distribution function is now replaced by the information of liquid refrigerant distribution ($\dot{m}_{l,i}$) acquired by the IR quantification method. Knowing the liquid mass flow rates distribution ($\dot{m}_{l,i}$), the vapor mass flow rate through each tube ($\dot{m}_{v,1}, \dot{m}_{v,2} \ldots \dot{m}_{v,i} \ldots, \dot{m}_{v,n}$) can be determined using the following pressure drop (function of $m_{l,i}$ and $m_{v,i}$) equality relationship along each flow path and the mass conversation equation of the vapor refrigerant.

$$\Delta P_{1^{st}\,path} = \Delta P_{2^{nd}\,path} = \ldots = \Delta P_{n^{th}\,path} \tag{11}$$

$$\dot{m}_{v,1} + \dot{m}_{v,2} + \ldots + \dot{m}_{v,n} = x\,\dot{m}_{inlet} \tag{12}$$

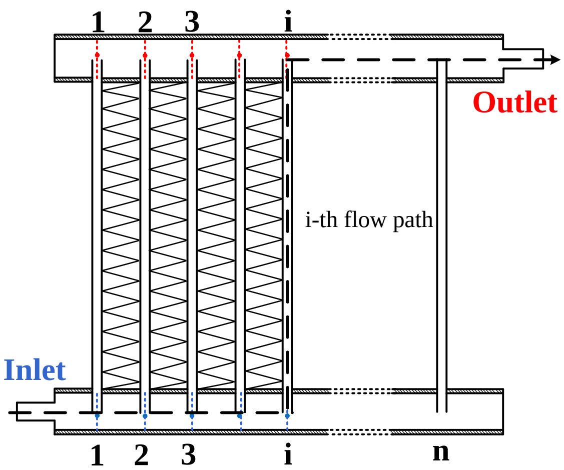

Figure 3: Configuration of microchannel heat exchanger

Using the newly calculated mass flow rates ($\dot{m}_1, \dot{m}_2 \ldots \dot{m}_i \ldots \dot{m}_n$), new transition line is identified. Iteration will stop when mass flow rates reach convergence. The resultant transition line is shown as the blue line in Figure 4. If more precision is needed in the model, actual pressure at the transition location will be determined iteratively instead of assumed to be the average of inlet and outlet pressures.

Comparing these two transition lines, it can be found that in the most superheated tubes, the blue line is located above the red line because these tubes receive less than average mass flow rates, resulting in lower refrigerant-side heat transfer coefficient and higher transition wall temperature; In the least superheated tubes, the blue line is below the red line, because the mass flow rates through these tubes are higher than average, contributing to a higher refrigerant-side heat transfer coefficient and lower transition wall temperature. Even though the red line is determined using average mass flow rate (no need for heat exchanger model), it can still provide a reasonable estimation which makes the quantification method easier to apply.

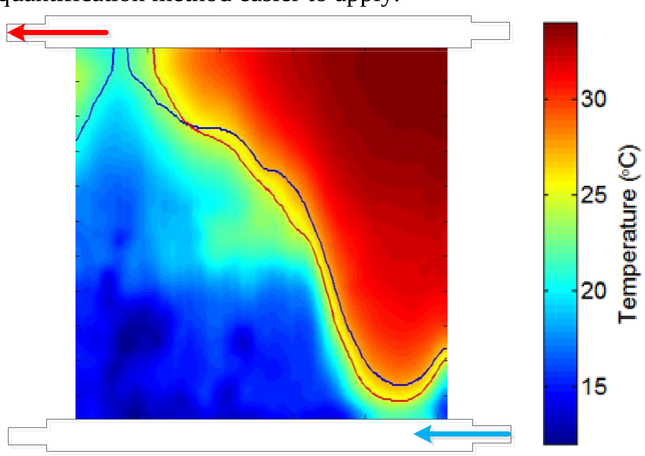

Figure 4: Determination of transition line

3. METHOD VALIDATION

The newly proposed quantification method of liquid refrigerant distribution is integrated into Li and Hrnjak's (2013) heat exchanger model as introduced in section 2.2. The new model is validated against Bielskus's (2011) experimental data. A microchannel evaporator taken from a mid-size, major brand vehicle was used in his study. The experiments were conducted under three indoor/outdoor conditions (25℃/25℃, 35℃/ 35℃, 40℃/40℃) and four different superheats at compressor inlet (5℃, 10℃, 15℃, 20℃). Since Li and Hrnjak's model is inherited from an experimentally validated (using Bielskus's (2011) data) model developed by Tuo *et al.* (2012), the comparison of the simulation results with experimental data presents the validation of the method.

The inlet and outlet location of the heat exchanger is indicated on the top left part in the following two figures. Figure 5 shows the comparison of the simulated surface temperature (using modified Jin's distribution function and IR quantification method) and experimental results under the same indoor temperature but different superheats at the compressor inlet. The case of 35 °C indoor temperature is used as an example.

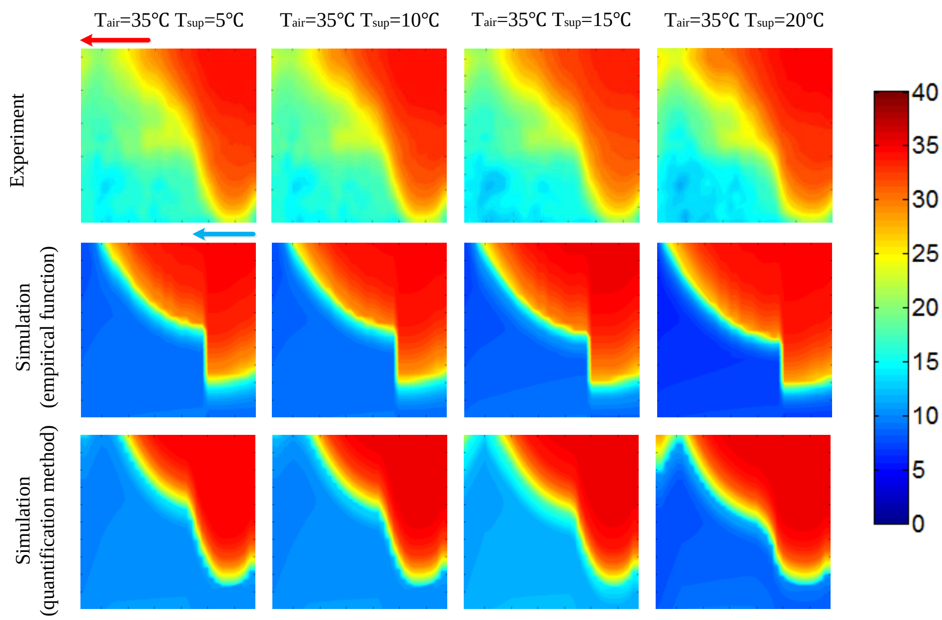

Figure 5: High similarity of the shape of the superheated zones in experiment and simulation validates the IR quantification method presented

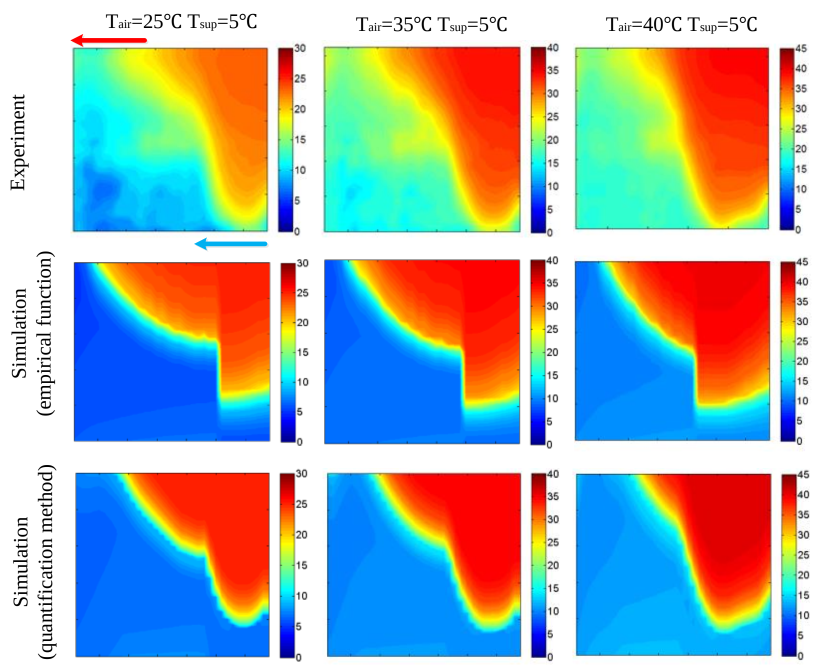

Figure 6: Same as in Figure 6 but for different conditions

Figure 6 shows the comparison of the simulated surface temperature and experimental results under the same compressor inlet superheat but different indoor temperatures. The 5 °C superheat condition is chosen as a show case.

From Figure 5 and Figure 6, it can be seen that the shape of the superheat region calculated using IR quantification method is smoother and more similar with the experimental results than the ones calculated using empirical distribution functions. The high similarity of the shape of the superheat regions between simulations (using IR quantification method) and experiments indicates accurate estimation of refrigerant distribution among parallel microchannels. The simulated surface temperature is found to be consistently lower than the experimental results for 2-3 °C.

Good agreement (±5%) between simulated and measured capacity shown in Figure 7 provides additional assurance of the fidelity of the model with IR quantification method.

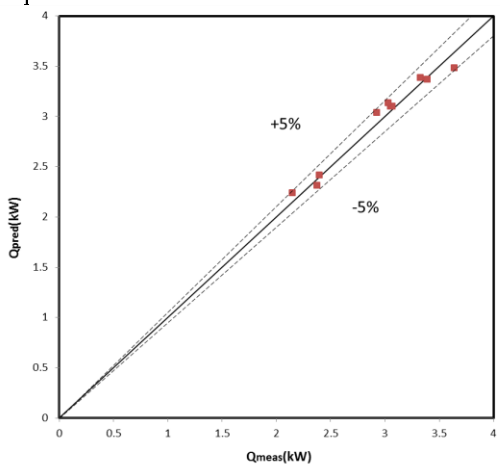

Figure 7: Capacity validation

4. APPLICATION

This quantification method can be applied to various heat exchanger designs. Since refrigerant maldistribution is less important in condenser (Pottker and Hrnjak, 2012), the following examples only focus on evaporators.

Figure 8 shows the liquid refrigerant distribution inside of a microchannel heat exchanger with inlet and outlet on the opposite side. For the IR image on the left side, liquid mass flow rate fraction calculated using the IR quantification method is presented on the right side in blue. The red line indicates the liquid mass flow rate distribution using empirical distribution function developed by Jin (2006) (modified by Tuo *et al.* (2012)). The similarity of two liquid refrigerant distribution profiles indicates that far simpler and less costly IR quantification method provides even slightly better results.

Figure 9 illustrates that the quantification method is cable of providing good estimation of liquid refrigerant distribution regardless of the inlet and outlet locations.

Figure 10 demonstrates that this method can also handle the estimation of liquid refrigerant distribution in the heat exchanger with vertical headers and horizontal tubes.

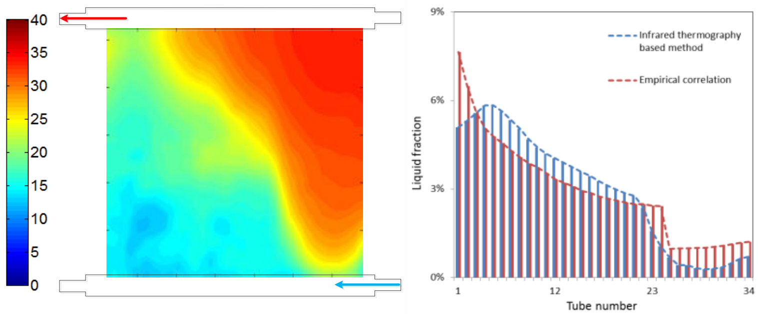

Figure 8: Example for heat exchanger with inlet and outlet on the opposite side

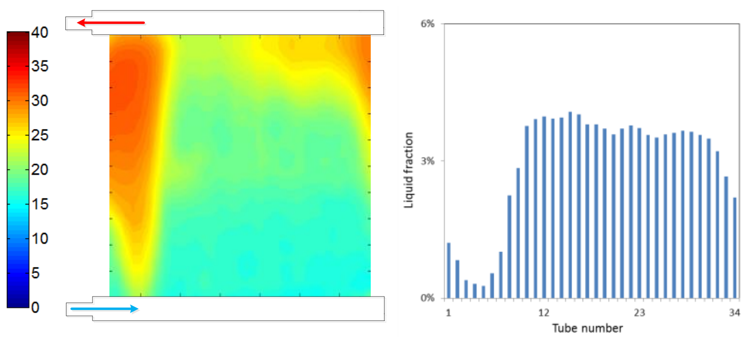

Figure 9: Example for heat exchanger with inlet and outlet on the same side

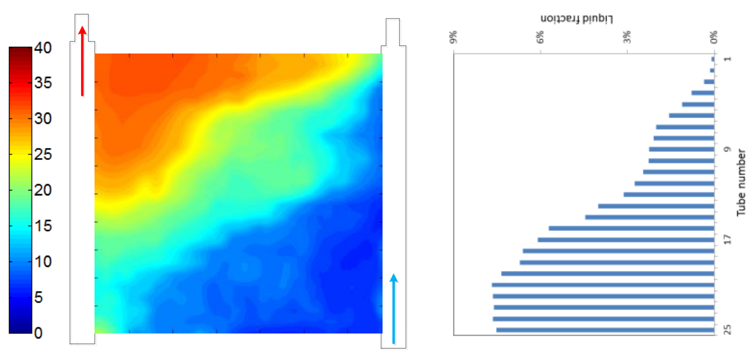

Figure 10: Example for heat exchanger with vertical header and horizontal tube

The existence of flooded tubes in a heat exchanger creates limitation for this method. The quantification of liquid refrigerant distribution is based on the premise that all the liquid has turned into vapor at the end of the two phase region. So this method has difficulties working in situations like the first pass in a multiple circuits evaporator or the last pass in a multiple circuits condenser where many flooded tubes exist (shown in Figure 11).

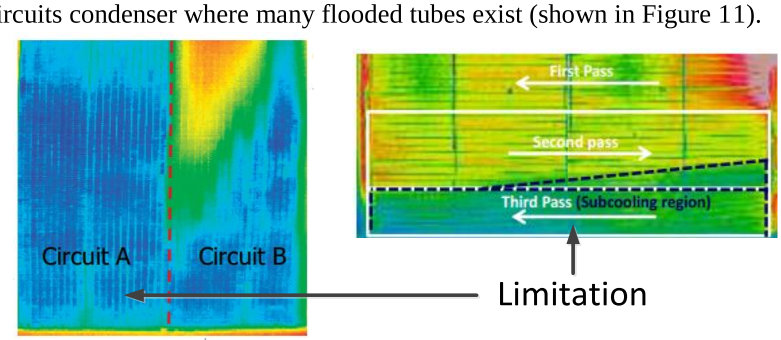

Figure 11: Limitation of the quantification method

5. CONCLUSIONS

In this study, an IR thermography based method is proposed to quantify liquid refrigerant distribution in parallel flow microchannel heat exchangers. After being implemented in a heat exchanger model, this method is validated against experimental data. Good agreement has been achieved between simulation and experiment for surface temperature and capacity.

This quantification method is non-contacting and none-intrusive, thus is easy to apply. It can be used to evaluate the refrigerant distribution in operating heat exchangers under various conditions. The paper demonstrates the application of the methods to different heat exchanger orientations and designs. Authors are currently working on application of this method to develop more general correlation/process to predict refrigerant distribution among parallel microchannels.

NOMENCLATURE

A	area	(m^2)
C	heat capacity	($kJkg^{-1}K^{-1}$)
ε	heat exchanger effectiveness	(–)
h	heat transfer coefficient	($kWm^{-2}K^{-1}$)
i	specific enthalpy	($kJkg^{-1}$)
$\dot{m}$	mass flow rate	(kgs^{-1})
NTU	number of transfer units	(–)
P	pressure	(kPa)
Q	capacity	(kW)
T	temperature	(K)
U	heat exchanger effectiveness	(–)
Subscript		
air	air side	
element	element	
f	fluid	
g	gas	
HX	heat exchanger	
in	inlet	
l	liquid	
min	minimum	
p	pressure	
path	path	
ref	refrigerant	
tp	two phase	
tube	tube	
v	vapor	

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ACKNOWLEDGEMENT

The authors thankfully acknowledge the support provided by the Air Conditioning and Refrigeration Center at the University of Illinois at Urbana-Champaign.