# Investigation of Port Level Refrigerant Flow Maldistribution in Microchannel Heat Exchanger

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**Abstract**

This paper presents an investigation of port level refrigerant flow maldistribution in microchannel heat exchanger (MCHX). The objective is to observe and quantify port level refrigerant flow maldistribution induced by the uneven heat transfer caused by air condition change in air flow direction. A microchannel heat exchanger model which is comprehensively validated by experimental data is employed to simulate a 4-pass 36-tube microchannel condenser with 18 rectangular ports in each tube. The effects of refrigerant mass flux, refrigerant inlet quality, refrigerant saturation delta T and air velocity are explored. The coefficient of variation (CoV) is introduced to quantify the port level flow maldistribution. For condensation tube, the mass flow rate increases first, then decreases along air flow direction. There is a peak in the concave maldistribution curve where the corresponding port holds the maximum mass flow rate. The parametric study shows that large refrigerant mass flux, large inlet vapor quality and small air inlet velocity tends to bring the peak of the maldistribution curve forward. The maximum CoV of maldistribution observed is 0.1424 and the maximum MCHX capacity percentage degradation for the sample inlet tube is 33% and the largest capacity degradation for the entire MCHX is 3.66%. This study indicates that port level maldistribution has significant impact on heat exchanger performance. The conclusions of this study serve as a contribution, which can enhance the design of MCHX geometry and its operation condition to make the refrigerant flow distribution more uniform in port level.



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*Keywords*: microchannel heat exchanger, port level, flow maldistribution, simulation.

## 1. Introduction

Microchannel heat exchangers (MCHX) have been widely used in automotive industry and being increasingly applied in residential heat pumps because of their compactness, significant charge reduction, lower refrigerant pressure drop and lower air-side fan power consumption compared to tube-fin heat exchangers. MCHX has two limitations which restrict its applicability. The first is the condensate drainage problem. Compared with tube-fin heat exchanger, the condensate accumulated on fins and flat tubes blocks air from passing through the exchanger smoothly and thus severely deteriorates its performance. The second limitation is the flow maldistribution problem. Due to the flow maldistribution, part of the MCHX is filled with single phase fluid which causes significant performance degradation. The flow maldistribution can be induced by MCHX geometry such as header orientation, micro-channel tube diameter, tube pitch, tube length, as well as MCHX operation conditions such as inlet quality, refrigerant mass flux etc. For evaporators, refrigerant maldistribution can cause dry-out phenomena. For condensers, maldistribution can create zones of high liquid accumulation.

To study MCHX refrigerant maldistribution, Kim and Sin [17] investigated the effects of tube protrusion depths, tube outlet direction, refrigerant mass flux, inlet vapor quality on flow maldistribution of MCHX tubes. Marchitto [20] investigated the effect of liquid and gas superficial velocity of MCHX inlet header and explored the influence when inlet nozzle was inserted into the header. Ahmad et al. [1] conducted experiments in three different MCHX tube orientations to study different structural and functional parameters that can influence the flow distribution. Kim [18] conducted experiments for three different inlet orientations of MCHX headers using R134a in a parallel flow MCHX to investigate the effects of the inlet flow orientation and operating conditions including refrigerant mass flux and vapor quality. Nielsen et al. [23] investigated the effects of channel thickness on refrigerant maldistribution inside MCHX tubes. Kim et al 2013 [19] conducted a study to improve flow distribution by using devices such as wire mesh, perforated plates and perforated tubes. Hwang et al. [13] investigated the effects of MCHX manifold geometry on refrigerant distribution in a horizontal header and vertically oriented tubes. Zou and Hrnjak [28] experimentally studied the effects of upward flow in vertical header on distribution of R134a. Zou and Hrnjak [29] compared the distribution of R134a and R410A in MCHX with vertical header and drew the conclusion that in general, refrigerant is evenly distributed for high mass flux and low inlet quality with an exception in the case of R410A which provides uniform distribution even at high inlet quality. Zou et al. [30] investigated the effects of fluid properties on two phase flow distribution and drew the conclusion that fluids with high liquid to vapor density ratio provide better flow distribution. Huang et al. [12] adopted a CFD and effectiveness-NTU based co-simulation approach to investigate the flow maldistribution in MCHX headers. Bowers et al. [3] and Brix et al. [14] investigated the effects of MCHX refrigerant maldistribution by experiment and simulation respectively, both studies show that flow maldistribution in MCHX evaporator can lead to significant degradation on heat exchanger performance. Zou and Hrnjak [28] developed empirical correlations to predict liquid take-off ratio in MCHX headers for R134a and R410A. Although many studies have been conducted to investigate the refrigerant flow distribution inside MCHX headers and tubes, to the author's knowledge, very few literature have appeared in the literature accounting for MCHX port level refrigerant flow maldistribution. Among these few literature, Dario et al. [8] experimentally investigated the microchannel port level refrigerant maldistribution induced by the orientation of the header and the position of the inlet feeder tube on the header. There is no paper found in the literature to investigate port level refrigerant maldistribution induced by uneven heat transfer caused by air condition change in air flow direction.

Compared to the time-consuming MCHX product designing and testing process, simulation tools are now extensively used in the performance evaluation and design of MCHX as well as other air-to-refrigerant heat exchangers. Several models for simulating MCHX performance are available in the literature. Yin et al. [27] developed a finite volume, first principle-based CO2 gas cooler model. They employed empirical correlations to predict heat transfer coefficients, pressure drop, and fin efficiency. Jiang et al. [16] presented a simulation and optimization tool for the design of air-cooled MCHX. The effectiveness-NTU method was employed to simulate the dry surface condition, while wet surface conditions were handled by McQuiston [21] enthalpy difference method. The tool, CoilDesigner® [16], incorporates a matrix representation for convenient designing and analyzing complex coil circuiting. A segment-by-segment approach that accounts for two-dimensionally non-uniform air flow distribution across the coil face has been implemented in the software. The model tracks and captures the significant change of thermo-physical properties as the refrigerant transitions between vapor, two-phase, and liquid regimes. It also provides a user-friendly graphical interface, and a choice of a wide variety of working fluids and air and liquid side heat transfer / pressure drop correlations. Schwentker et al. [24] verified the prediction of CoilDesigner® against experimentally measured data for eight R-134a microchannel condensers. The model was able to predict the condenser heat load within 2.25% for 80% of the 35 experimental data points. The average error, average absolute error, and the maximum error in the heat load prediction were -0.84%, 1.6%, and 4.6%, respectively.

Huang et al. [10] continued the effort to develop new MCHX model in CoilDesigner®, the new model is capable of simulating refrigerant flow distribution into different ports of each tube. In one of the most comprehensive microchannel condenser and gas cooler performance validation efforts, Huang et al. [10] validated their new model against 227 experimental data points for eight different working fluids including R410A and 18 MCHX geometries from seven different data sources. The average absolute deviation between the predicted and measured values of the heat duty and the refrigerant pressure drop was found to be 2.7% and 28%, respectively. More recently, Huang et al. [11] validated the model against experimental data for both condenser and evaporator applications using R410A and R32. 65 data points, including 45 condenser points and 20 evaporator points for eight different MCHXs were validated. Without using any correction factors on the heat transfer correlations, the absolute average capacity prediction errors ranged from 1.75% to 3.1%, while the pressure drop deviations ranged from 11.14% to 16.71%.

It is thus clear that MCHX port refrigerant flow maldistribution phenomena have not been explored in depth. Thus, the primary objective of this paper is to observe the refrigerant port level maldistribution, then investigate the effect of different operating parameters on port level flow maldistribution. Furthermore, this study aims to explore the impact of port level refrigerant maldistribution on MCHX performance.

## 2. Simulation Procedure

This paper adopts the MCHX model described in Huang et al. [10] for all simulations. Since the air inlet state is different from the first port to the last port for a given tube along air flow direction, the heat transfer difference from air to different ports can induce refrigerant mass flow maldistribution. The top level solution methodology of this model is presented in Figure 1. The air-side and refrigerant side heat transfer coefficients (HTCs) are calculated on a per port base. The air-side propagation is conducted iteratively and air-side condition is passed to the next port in air flow direction. In between every iterations, the per port air HTCs are updated based on the updated air side condition.

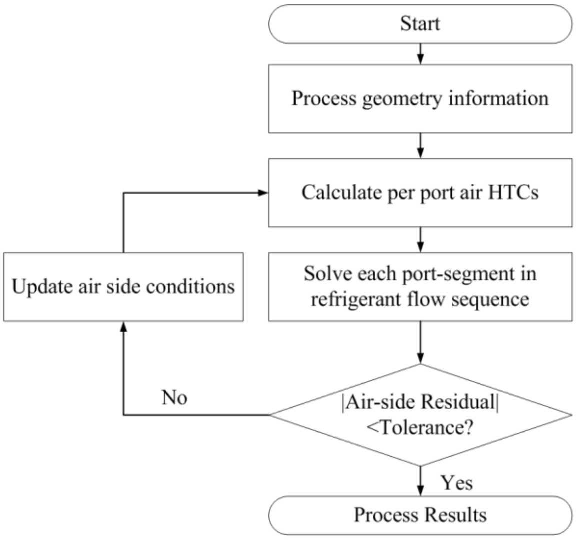


Figure 1: Solution methodology for Huang et al. [9] model

The MCHX condenser simulated in this study is shown in Figure 2. This heat exchanger has the optimal performance according to the study done by Menhendale et al. [22]. The geometries of this MCHX are shown in Table, those structural parameters are adopted from a commercial air-conditioning unit. The MCHX consists of 36 tubes. Each tube consists of 18 rectangular ports. The pass configuration is 13-13-6-4, meaning the first pass contains 13 tubes, the second, third and fourth pass contain 13, 6 and 4 tubes respectively. This pass configuration is the optimal design obtained from Menhendale et al. [22].

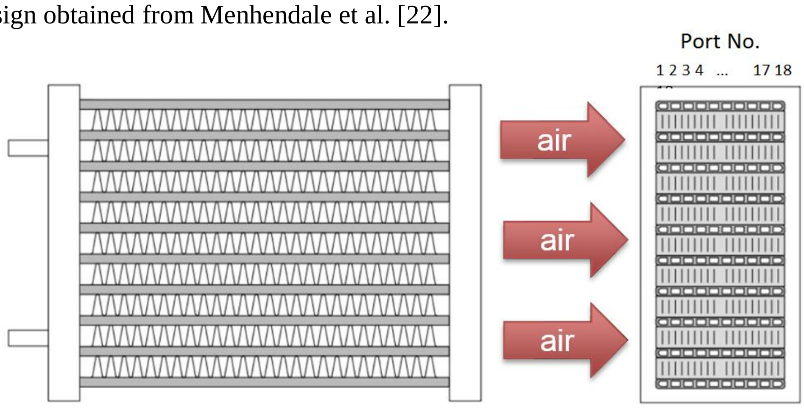


Figure 2: Schematic of MCHX simulated in this study

Table 1: MCHX structural parameters

| | |
|---|---|
| Tube Length (m) | 0.562 |
| Total number of tubes | 36 |
| Tube depth (m) | 0.0254 |
| Tube thickness (m) | 0.0018 |

| | |
|---|---|
| Port diameter (m) | 0.001 |
| Number of ports per tube | 18 |
| Number of passes | 4 |
| Pass arrangement | 13-13-6-4 |
| Fin density (FPI) | 20 |
| Louver length (m) | 0.0104 |
| Louver angle (deg.) | 30 |
| Louver pitch (m) | 0.00152 |
| Average Fin height (m) | 0.01641 |

In this study, R410A is chosen as the refrigerant because it is very commonly used in modern residential and commercial HVAC&R applications. Tables 2 lists the range of air and R410A side operating conditions. These operating conditions are adopted from ARI Standard 201/240 cooling mode A test condition [2]. Table 3 lists the pressure drop and heat transfer correlations used in simulation, those correlations are picked up based on their application ranges.

Table 2: Air and R-410A operating conditions used in the analysis

| Operation Mode | Air dry bulb (°C) | Air wet bulb (°C) | Average air velocity (m/s) | R410A inlet Tsat (°C) | R410A inlet quality | R410A mass flux (kg/m2s) |
|---|---|---|---|---|---|---|
| Condenser | 35 | 23.9 | 0.5 - 4 | 45 | 0 - 1 | 200 - 600 |
| Evaporator | 26.7 | 19.4 | 0.5 - 4 | 8.5 | 0.1-0.5 | 200 - 600 |

Table 3: Heat transfer and pressure drop correlations used in the analysis

| | Air side | R-410A side | | | |
|---|---|---|---|---|---|
| | | Vapor | Two-Phase | | Liquid |
| Heat transfer correlation | Chang and Wang [6] | Dittus-Boelter [9] | Shah [26] for condenser | Shah [25] for evaporator | Dittus-Boelter [9] |
| Pressure drop correlation | Chang et al. [5] | Blasius [14] | Chen et al. [7] | | Blasius [14] |

Two assumptions are made during the simulation. First, air flow at the coil face is assumed uniform. This paper focuses on port level flow maldistribution induced by air property change along air flow direction, thus the two dimensional air flow maldistribution which is perpendicular to the air flow direction is not of interest. Second, the distribution of refrigerant inside the header is assumed uniform. Because the prediction of refrigerant maldistribution in MCHX header relies on the accurate prediction of pressure drop inside header. However, the header pressure drop not only consists of single phase and two phase header frictional pressure drop, but include gravitational pressure drop as well. Thus the orientation of the header, i.e. whether the header is vertical or horizontal positioned, has an impact on header pressure drop as well as refrigerant flow maldistribution in headers. For the sole purpose to study port level maldistribution inside flat tube, it is not necessary to involve header flow maldistribution.

Based on the assumption that the header refrigerant flow distribution inside is uniform and considering the goal of this study to explore the port level flow maldistribution, it is unnecessary to evaluate the performance of the entire MCHX. Therefore, this study chooses one tube at the first pass as the objective to study port level maldistribution. This particular tube is chosen based on two principles. First, it is convenient to control the inlet condition of the tube at inlet pass of MCHX. Second, among all 13 tubes in the $1^{st}$ pass, the first tube has different heat transfer area, because it may or may not have extension fins, while the last tube adjacent to the second pass can interact with the second pass and bring some end-effects. Therefore, as an intermediate tube, the $7^{th}$ tube at the first pass is chosen as a sample to study and in the following section, all results except Figure 9 are referring to this single tube.

## 3. RESULTS AND DISCUSSION

Figure 3 shows port level mass flow distribution result for a condenser tube. The x-axis is the MCHX port number, air flows from port #1 to port #18, the y-axis is the normalized mass flow rate defined in Equation (1).

$$Normalized\ mass\ flow\ rate = \frac{m_i}{\bar{m}}, where\ \text{port\# i} = 1, 2, \ldots, 18 \tag{1}$$

Figure 3 indicates that, for condensation tubes, the mass flow rate first increases, then decreases along air flow direction. There is a peak for each curves where the corresponding port holds the maximum mass flow rate. When the mass flux increases, the peak will be brought forward closer to the MCHX air inlet.

Figure 4 shows a tube from the MCHX evaporator. The maldistribution trend in an evaporator tube is very different from that in a condenser tube. The mass flow rate monotonously increases along each port in air flow direction. The larger the mass flux is, the more significant the maldistribution is. The different trends for mass flow distribution in condensation and evaporation attribute to different processes of thermal-physical properties change.

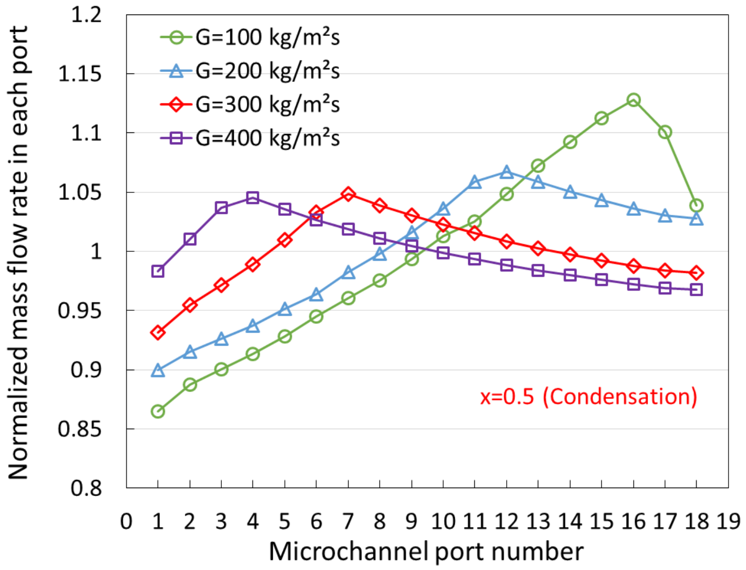


Figure 3: Port level flow distribution in a condenser tube

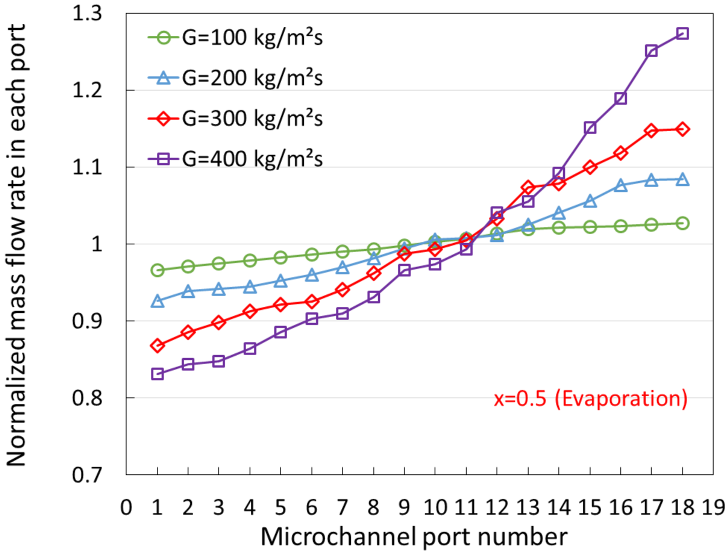


Figure 4: Port level flow distribution in an evaporator tube

In order to explore the effect of refrigerant inlet vapor quality, the condenser is simulated under different inlet qualities as in Figure 5. The maldistribution profile is still a concave. It indicates that large quality can bring the “peak” port containing the maximum mass flow rate forward.

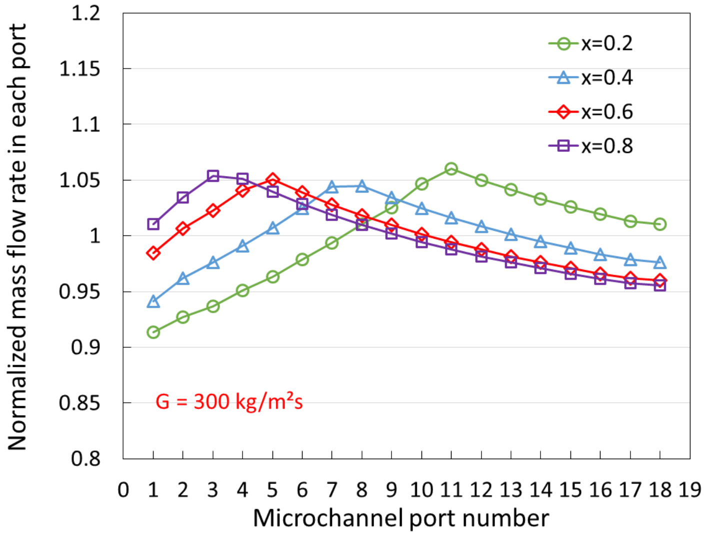


Figure 5: Port level flow distribution under different inlet qualities

Since the inlet of a MCHX condenser is usually in superheat region and the outlet tube is in sub-cooling liquid region, it is worthwhile to explore the maldistribution for superheat and sub-cooling tubes. In Figure 6, the inlet condition is set as 5K sub-cooling, 20K superheat, two phase with vapor quality x=0.2 and x=0.8 respectively. Figure 6 shows that sub-cooling liquid has perfect uniformity, because liquid refrigerant has very small thermal properties change during heating or cooling process compared with vapor phase and two phase refrigerant. Two phase case with low quality (x=0.2) shows a “concave” profile, while the two phase with high quality (x=0.8) shows that the mass flow decreases along air flow direction. As it has been discussing in Figure 5, high inlet quality can bring the peak of this concave forward, thus this monotonic trend can be explained as the quality is so large that the peak has been dragged beyond the first port. In addition, the superheat tube shows very similar monotonic maldistribution trend as the high inlet quality (x=0.8) tube.

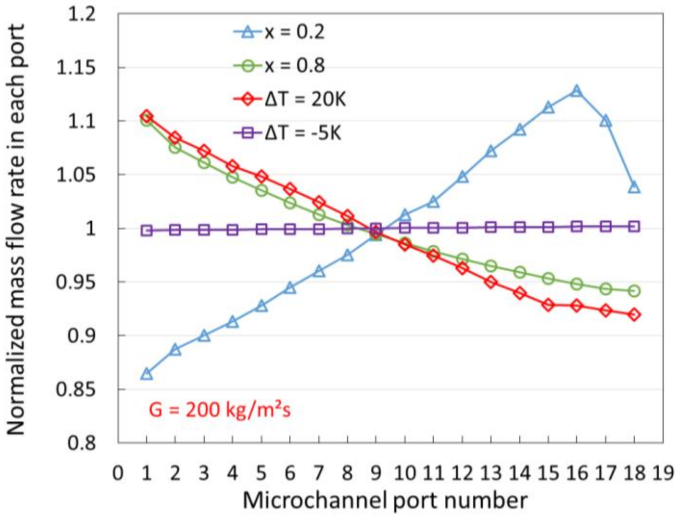


Figure 6: Port level flow distribution at the inlet, intermediate and outlet tubes of a condenser

The port flow maldistribution in this study is induced by air property change, thus frontal air velocity is another operating parameter which has impact on flow maldistribution. Figure 7 shows the simulation results for the condenser tube under different air velocity. It indicates that the small air velocity tends to bring the "peak" port with maximum mass flow forward. And when the air velocity is enlarged to a certain value, the peak of the concave curve is beyond the last port, then the mass flow monotonously increases along air flow direction.

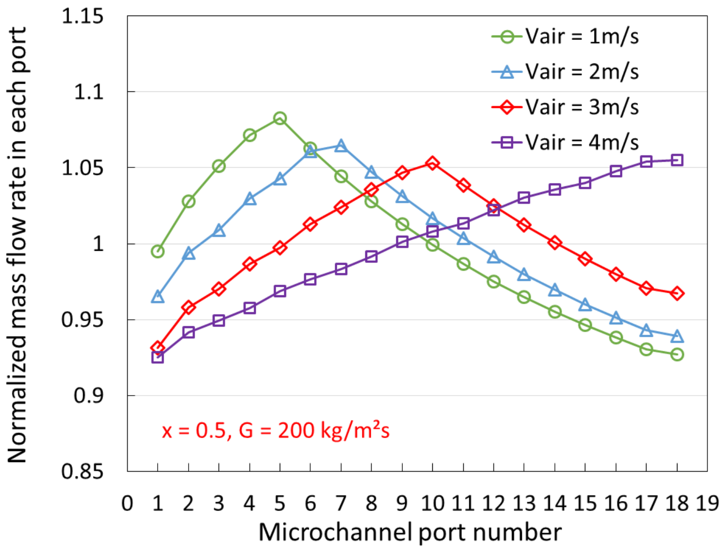


Figure 7: Port level flow distribution under different frontal air velocity

In order to assess the impact of port level refrigerant flow maldistribution on MCHX performance, the coefficient of variation (CoV) is introduced to define the degree of flow maldistribution as in Equation (2). CoV can vary from 0 to large positive value, and zero value of CoV indicates the ideally uniform distributed flow.

$$CoV = \frac{\text{standard deviation}(\sigma)}{\text{average}(\overline{m})} = \frac{\left[\frac{\sum_{i=1}^{N}(\dot{m}_i - \overline{m})^2}{N}\right]^{0.5}}{\overline{m}} \quad (2)$$

To evaluate MCHX performance difference with and without port level flow maldistribution, capacity degradation number in Equation (3) is used. The positive value of this number indicates the uniformly distributed refrigerant flow has larger capacity than mal-distributed refrigerant flow. The value of this number indicates the capacity percentage degradation induced by maldistribution.

$$\text{Capacity degradation number} = \frac{Q_{uniform} - Q_{maldistribution}}{Q_{uniform}} \tag{3}$$

Figure 8 shows the capacity degradation number for the sample inlet tube (tube #7). Figure 9 shows the capacity degradation number for the entire MCHX. In both figures, the degradation numbers are positive, indicating port level maldistribution is detrimental to MCHX performance. The largest CoV observed in this study is 0.1424, and the largest capacity degradation for the sample inlet tube is 33%, while the largest capacity degradation for the entire MCHX is 3.66%. These numbers indicate that port level maldistribution has significant impact on MCHX performance. It can be seen that the degradation number is not monotonously correlated to CoV, which means large CoV not necessarily induces large capacity degradation.

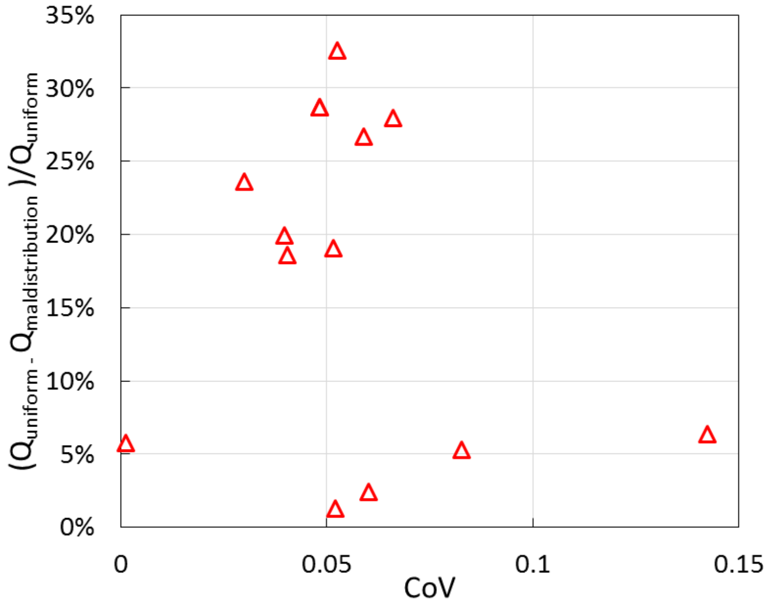


Figure 8: Port level mass flow distribution impact on the capacity of a single inlet tube

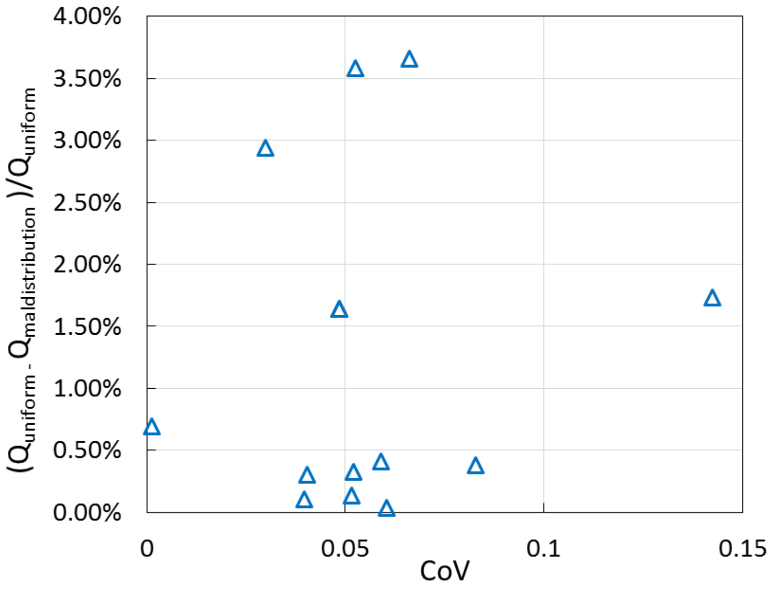


Figure 9: Port level mass flow distribution impact on the capacity of entire MCHX

## 4. CONCLUSION

This paper presents an investigation of port level refrigerant flow maldistribution in microchannel heat exchanger. The objective is to explore the effect of port flow maldistribution on heat exchanger performance and to quantify this maldistribution by coefficient of variance. A microchannel heat exchanger model which is comprehensively validated by experimental data is employed to simulate a 4-pass, 36-tube MCHX with 18 rectangular ports in each tube.

The result shows that condensation tube has different port flow maldistribution trend compared with evaporation tube. For evaporation tube, mass flow rate monotonously increases along each port in air flow direction and larger refrigerant mass flux can induce more significant flow maldistribution. For condensation tube, along each ports in air flow direction the refrigerant mass flow rate increases first, then decreases. There is a peak in the concave curve where the corresponding port holds the maximum mass flow rate. The parametric study shows that large refrigerant mass flux, large inlet vapor quality and small air inlet velocity can bring the peak of the maldistribution curve forward to be closer to the air inlet. And compared with two phase tube, superheat tube has similar maldistribution trend as the high vapor quality tube, while sub-cooling liquid tube shows perfect distribution uniformity.

It is found the maximum coefficient of variance of maldistribution observed in this study is 0.1424 and the maximum MCHX capacity percentage degradation for the sample inlet tube is 33% and the largest capacity degradation for the entire MCHX is 3.66%. It indicates that port level maldistribution has significant negative impact on heat exchanger performance. It is worthwhile to further investigate this topic and a variable port size profile can be expected to improve the refrigerant flow distribution. The conclusions of this study serve as a contribution, which can enhance the design of MCHX geometry and its operation condition to make the refrigerant flow distribution more uniform in port level.

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