Determination of air side heat transfer coefficient in a mini-channel heat exchanger using Wilson Plot method

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Abstract. In this study, the air side heat transfer coefficient of an aluminium mini-channel heat exchanger was investigated for single-phase flow in the mini-channel, with water in the tubes and air on the outside. Research methods included hydraulic tests on a single mini-channel tube, Wilson Plot experiments and experiment validation. Results obtained from the hydraulic test showed that turbulent flow occurred in the tube at a Reynolds number of 830. Wilson Plot experiments were conducted to determine air side heat transfer coefficient of the heat exchanger. The tube side Reynolds number was maintained above 1000 to ensure turbulent flow and tube side heat transfer coefficient was calculated using Gnielinski equation for turbulent flow. The air side heat transfer coefficients obtained from the Wilson Plot experiments were in good agreement with known correlations. The outcome of this study is to use the air side heat transfer coefficient to calculate the performance of refrigerant condensers for different tube pass ratios and flow pass configurations.

1. Introduction
Past research on mini-channel heat exchangers had focused on condensation heat transfer, pressure drop, two-phase flow patterns and system configuration while evaporator mini-channels research has gained much popularity in recent years. Due to the high price of copper, aluminium Mini-Channel Heat Exchangers, or in short MCHX, are gaining popularity to replace the conventional copper-aluminium air-conditioning heat exchangers. MCHX are made from extruded aluminium tubes brazed to aluminium louvered fins in a temperature controlled environment using a process such as the Nocolok brazing. The tubes are brazed at both ends to aluminium manifolds or headers. Thus the entire construction is from aluminium.
Air is drawn by a blower and passes through the matrix of the fins and tubes of the MCHX while the refrigerant passes through the tubes. The advantages of MCHX are high heat transfer surface area per unit volume (A/V), light weight and resistance to galvanic corrosion.

As in the conventional heat exchangers, the MCHX is also subject to air and refrigerant side flow maldistribution. An example of air side maldistribution in conventional finned-heat exchangers had been studied by Chin and Raghavan [1]. Various factors that contribute to refrigerant flow maldistribution include the heat exchanger geometry, operating conditions, multiphase flow and fouling [2]. Maldistribution causes a degradation in the heat exchanger performance which would mean less efficient heat exchange as well as more energy required to overcome the pumping losses.

Other problems that would bring about a high pressure drop and performance degradation to the mini-channel heat exchanger are the tube pass ratio and number of passes.

![Figure 1. Division of the Mini-Channel according to Tube Pass Ratio.](image)

| Table 1. Type of Mini-Channel used in this Study according to Different Tube Pass Ratio. |
|-----------------------------------------------|
| **Type** | 1  | 2  | 3  | 4  | 5  |
| n       | 50 | 60 | 70 | 80 | 90 |
| m       | 50 | 40 | 30 | 20 | 10 |
With equal flow passes in all configurations shown above, the heat transfer and pressure drop would be sub-optimal. At the inlet the volume flow rate would be high in view of the high vapor fraction; the velocities would be high leading to a good thermal performance. Due to condensation of vapor, the velocities will be lower in the second pass giving rise to low pressure drop but also poor heat transfer. The aim of good design would be to balance the number of tubes in each pass as well as the number of passes in such a way as to get the maximum heat transfer at minimum pressure drop. In turn this would mean a higher flow area in the first pass.

The overall scope of this work is to investigate different tube pass ratio which includes 50/50 as a baseline case, 60/40, 70/30, 80/20 and 90/10 and different number of flow configuration which includes 2 as a baseline case, 3 and 4 pass. Water is used as the tube-side working medium in the hydraulic test and Wilson Plot experiments. For the Wilson Plot experiments, the water and air side velocities range between 1.0-3.0 m/s.

2. Literature Review

A literature survey was undertaken to review the research on various design considerations for heat exchangers that cause degradation of their performance, with focus given to mini-channel heat exchangers. The outcome of this survey showed that the proposed work has not been carried earlier out by any researcher.

A study of flow maldistribution was conducted by Bassiouny and Martin [3,4] in plate heat exchangers (PHE). The study used a theoretical calculation method to determine flow distribution and pressure drop in plate heat exchangers for the U-type, where both the inlet and outlet ports were on the same side and also the Z-type, where the inlet and outlet ports were on opposite sides.

Another study conducted by Rao et al [5] looked into the effect of port-to-channel flow maldistribution on the thermal behavior of multipass plate heat exchangers (PHE). The variation of the heat transfer coefficient due to flow variation from channel to channel was also considered. It was found that multipass heat exchangers were generally found to be less affected by flow maldistribution compared to single pass heat exchangers.

Hwang et al [6] studied the refrigerant inlet location in the manifold and compared the behavior of side-inlet and end-inlet locations. The side-inlet location showed better liquid refrigerant flow...
distribution than the end-inlet location by more effectively mixing the liquid and vapor refrigerant from the inlet. They found that by improving refrigerant maldistribution through the manifold inlet, the mini-channel evaporator performance could be improved.

Kandlikar et al. [7] stated that the pressure drop penalty needs to be carefully evaluated when selecting the channel size. The pressure gradient (pressure drop per unit length) increases drastically with channel size reduction [8] and therefore mini-channel heat exchangers need to be suitably designed to provide short flow lengths to limit the overall pressure drop.

Cavallini et al. [9] studied condensation heat transfer and pressure drop in a horizontal smooth tube using a new HFC refrigerant. The effects of vapour quality, mass velocity, saturation temperature, driving temperature difference and reduced pressure were discussed. It was discovered that low-pressure and mid-pressure refrigerants such as R22 perform better than high pressure refrigerants such as R410A and that at the same mass velocity and vapor quality, low pressure fluids such as R22, on the other hand, show a higher pressure drop penalty than R410A.

Del Col et al. [10] studied the effect of the cross sectional shape on the condensation behaviour in a single square mini-channel and compared it to the circular mini-channel. It was discovered that at lower mass fluxes the square channel yielded higher heat transfer coefficients than that of a circular channel. This was thought to be due to the effect of surface tension pulling the liquid towards the corners and reducing the average thermal resistance.

Kandlikar and Schmitt [11] studied the effect of surface roughness on the pressure drop in single phase flow in a mini-channel pipe and found that the average relative roughness value $\varepsilon / D_h$ was greater than the threshold value of 0.05 used in the Moody diagram. They introduced three surface characterization parameters together with equivalent roughness and constricted hydraulic diameter. Brackbill and Kandlikar [12] proved that using these parameters, the relationship between the critical Reynolds value and $\varepsilon / D_{h,ef}$ agree theoretically and experimentally.

Using known correlations and computer software, Subramaniam and Garimella [13] studied the parameters that influence the performance of R410A micro-channel condensers and suggested an optimum condenser design from a baseline design.

3. Methodology
The effects of the different tube pass ratios and number of flow passes of the mini-channel are to be compared in terms of thermal and hydraulic performance. In order to do this, an experiment is conducted using the Wilson Plot Method to determine the air side heat transfer coefficient, $h_o$ and overall heat transfer coefficient, $U_o$. As the flow of refrigerant inside the tube must be turbulent flow for the Wilson Plot experiment, the turbulent flow Reynolds number, $Re_t$ is first determined from flow experiments using a single tube.

For the single tube test, an experimental rig (figure 3) is set-up and water is used as the medium flowing inside a single mini-channel. Temperature and pressure drop on the water side across the tube and water flow rate are measured using resistance temperature detectors (RTD), differential pressure transmitter and magnetic flow meter respectively. Data collected from this experiment are used to calculate Re and Fanning friction factor, $f_F$ and using (1) and (2):

$$Re = \frac{\rho v h}{\mu} \quad (1)$$

where $\rho$ is the density of the liquid, $v$ is the velocity of the fluid, $d_h$ is the hydraulic diameter of the mini-channels and $\mu$ is the viscosity of the liquid.

$$f_F = dP \left( \frac{d_h}{L} \right) \frac{1}{2 \rho v^2} \quad (2)$$
where \( f_f \) is the Fanning friction factor, \( L \) is the tube length, \( v \) is the flow velocity and \( dP \) is the pressure drop across the tube.

\[
\Delta P = \frac{f_f}{4L} \frac{v^2}{2g}
\]

**Figure 3.** Schematic of Experimental Rig to determine \( Re_t \)

The \( f_f \) and \( Re \) values are then used to plot a \( f_f \) vs. \( Re \) chart from which \( Re_t \) is determined.

Water is used as the medium inside the mini-channel and the water side heat transfer coefficient \( h_i \) is calculated using the Gnielinski equation.

\[
h_i = \frac{(Re-1000)Pr(f/2)(k/d_h)}{(1+12.7(Pr^{2/3}-1)(f/2)^{0.5})}
\]

(3)

where \( Re \) is the Reynolds number, \( Pr \) is the Prandtl No, \( f \) is Fanning the friction factor, \( k \) is the thermal conductivity and \( d_h \) is the hydraulic diameter of the mini-channel.

From the Wilson Plot experiment data, \( h_o \) and \( U_oA_o \) can be determined.

\[
\frac{1}{U_o} = \frac{1}{n_o h_o} + R_w + \frac{A_o}{h_i A_i}
\]

(4)

where \( U_o \) is the overall heat transfer coefficient, \( h_o \) is the air side heat transfer coefficient, \( h_i \) is the water side heat transfer coefficient, \( \eta_o \) is the surface effectiveness, \( R_w \) is the tube wall thermal resistance, \( A_o \) is the is the total heat transfer area of external fin and tube and \( A_i \) is the internal tube heat transfer area.
where $A_f$ is the entire fin surface area, $A$ is the total surface area (fin plus exposed base), $\eta_f$ is the efficiency of a single fin, $h$ is the convective coefficient of the fin, $k$ is the thermal conductivity of the fin and $t$ is the thickness of the fin.

\[
\eta_o=\frac{A_f}{A}(1-\eta_f)
\]  

(5)

\[
\eta_f=\frac{\tanh(mL)}{mL}
\]  

(6)

\[
m=\left(\frac{2h}{kt}\right)^{1/2}
\]  

(7)

\[\frac{1}{U_o} \text{ versus } \frac{1}{V^{0.8}}\] determined from Wilson Plot Experiment.

Figure 4. $1/U_o$ versus $1/V^{0.8}$ determined from Wilson Plot Experiment.
The Wilson Plot test set-up is shown in figure 5. The air side and water side temperatures are measured using resistance temperature detectors (RTD). The air and water flow rate are measured using air nozzle and flow meter respectively. Air temperature is controlled at the air side inlet to the mini channel heat exchanger (coil) using chilled water fan coil unit and air heaters while water temperature is controlled at the water side inlet to the heat exchanger using water heaters. A frequency inverter is used to control the water pump in order to maintain the flow rate of the water.

All $h_0$ values determined from the Wilson Plot are used to calculate the performance of the mini-channel heat exchanger. A discrete element analysis is used to calculate the performance of the mini-channel heat exchanger for different tube pass ratios and flow pass configurations. This is done by dividing the heat exchanger into a number of discrete elements and using the $\varepsilon$-$NTU$ method to evaluate the local heat transfer rate $q_i$ of each element. The sum of $q_i$ values yields the total heat transfer rate $Q$, of the mini-channel heat exchanger. The pressure drop across the mini-channel heat exchanger is calculated using Bassiouny and Martin analysis method [3,4].

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**Figure 5.** Wilson Plot Test Rig Set-Up.
4. Results and Discussion
The Reynolds numbers for turbulent flow to initiate in the mini-channel tube is 830. See figure 6. The Wilson Plot Test are run at $Re$ above 1000. The Wilson Plot experiment results are shown in table 2.

![Friction Factor vs. Reynolds Number in Single Minichannel Tube](image)

**Figure 6.** Friction factor vs. Reynolds Number in Single Mini-Channel Tube.

**Table 2.** Air Side Heat Transfer Coefficient vs. Frontal Air Velocity.

| $n_h$, (W/m²·K) | 243.38 | 225.11 | 199.58 | 185.82 | 178.37 | 121.64 |
|------------------|--------|--------|--------|--------|--------|--------|
| Frontal Air velocity (m/s) | 8.20   | 7.29   | 6.39   | 5.47   | 3.64   | 1.83   |
A graph is plotted as shown in figure 7. The fitted equation of the curve is $y = 95.6x^{0.42}$. The exponent is 0.42. This agrees well with the general equation for flow over cylinders shown in (8) [14]. The small deviation is due to the flattened tube over which air flows in the MCHX.

$$h = 2.755V^{0.471} / D^{0.529}$$  \hspace{1cm} (8)

The value of $h_o$ thus determined is to be used to calculate the overall heat transfer coefficient $U$ of the mini-channel heat exchanger. In order to arrive at the best arrangement, the outlet state of the refrigerant is to be computed for different tube pass ratios and flow pass configurations, by utilizing several correlations for MCHX passages.

5. Conclusions
The Reynolds number at which turbulent flow initiates in the mini-channel tube is 830 as determined from the hydraulic test. Results of the Wilson Plot experiment show a close agreement with known correlations for the air-side. Therefore the $h_o$ values determined from the experiment are suitable to be used to calculate the performance of the mini-channel heat exchanger for different tube pass ratios and flow pass configurations which is a sequel to this study.

Acknowledgements
The first author would like to thank Professor Vijay R. Raghavan for his invaluable advice and OYL Research & Development Centre Sdn Bhd who has financially supported the experiment rig and test equipment.
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