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Single Phase Pressure Drop in Round Cylindrical Headers of Parallel Flow MCHXs

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ABSTRACT

This paper presents the investigation of the pressure drop in headers and development of correlation for pressure loss coefficient for single phase flow through round cylindrical headers of parallel MCHXs. The working fluid was compressed air flowing through header with 1 - 20 m/s based on smallest cross section while the velocity through micro-channels was in the range 6 - 30 m/s. The experimental results indicate that the pressure loss coefficient of inlet header is a linear function of the ratio of velocities through micro-channel tube and header, except for the first two micro-channel tubes; the pressure loss coefficient of outlet header is a quadratic function of the ratio of velocities through micro-channel tube and header, and decreases as the velocities through upstream micro-channel tubes increase. Correlations for predicting pressure drop of inlet header and outlet header are developed and agree for 98% of experimental data is within a ±15 Pa.

1. INTRODUCTION

Micro-channel heat exchangers show advantages over traditional fin-and-tube heat exchangers in compactness, lower refrigerant charge, etc. However, micro-channel heat exchangers face the problem of refrigerant distribution among parallel micro-channel tubes (Hrnjak, 2004; Hwang et al. 2007; Byun and Kim, 2011; Tuo and Hrnjak, 2012; Bowers et al., 2006; Dario et al. 2013; Ren et al., 2013) and pressure drop in the header strongly affects refrigerant distribution (Tuo and Hrnjak, 2013; Kim et al., 2004).

Single phase flow is common in inlet headers (gas coolers, condensers, etc.), and outlet headers (DX and FGB evaporators, etc.). In addition, it is logical to clarify the situation in single phase flow before addressing two phase flow. For two-phase evaporator, the refrigerant in an outlet header is usually in single phase state, and the pressure drop in the outlet header significantly affects the mass flow rate distribution through parallel micro-channel tubes and consequently degrades the heat exchanger performance (Tuo et al., 2012; Tuo and Hrnjak, 2013). As a result, it is needed to accurately predict the single phase pressure drop of micro-channel heat exchanger.

The single phase pressure drop of micro-channel heat exchanger could be broken down into two parts: pressure drop in micro-channel tubes and that in headers. The single phase pressure drops in the micro-channel tubes are well studied (Graham and Dunn, 1995; Heun and Dunn, 1995; Yin et al. 2001; Hrnjak and Tu, 2007), and the results showed that the correlation of Churchill (1977) has a good prediction for the laminar and turbulent flow regimes in the micro-channel tubes. The studies of pressure drop in the header are limited (Yin et al., 2002; Poggi et al., 2009). Yin et al. (2002) measured pressure drop in headers and generated a model based on the assumption that the pressure loss coefficients are uniform. Poggi et al. (2009) tested the pressure drop in a round inlet header, and the results show that the main pressure drop in the inlet header is caused by the contraction when flow passes over the first micro-channel tube. However, to the best of author's knowledge, there is still no information about non-uniformity of pressure loss coefficients in the headers. The change of pressure loss coefficient along the header results from the undeveloped flow in headers and the effect of flow through neighboring micro-channel tubes.

This paper presents the investigation of the pressure drop and development of correlation for pressure loss coefficient for single phase flow through round cylindrical inlet and outlet headers. The results presented in this study are obtained by compressed air. The velocity through headers based on smallest cross section range from 1 m/s to 20 m/s while the velocity through micro-channel tubes is in the range from 6 m/s to 30 m/s, which cover the most of realistic working conditions in the residential and automotive air conditioning system.

2. EXPERIMENTAL SETUP AND THE TEST FACILITY

The test system consists of a gas tank, a temperature pre-conditioner and a header, as shown in Figure 1. For the inlet header test flow passes through m_0, and then enters into the header. Part of gas leaves the header through the first 4 micro-channel tubes (MC tubes #1 to #4), and the rest exits to the atmosphere through the header outlet. For the outlet header test, the gas goes through the first five micro-channel tubes (MC tubes #1 to #5). The mass flow rate transducer m_0 is used to measure the mass flow rate through MC tube #5.

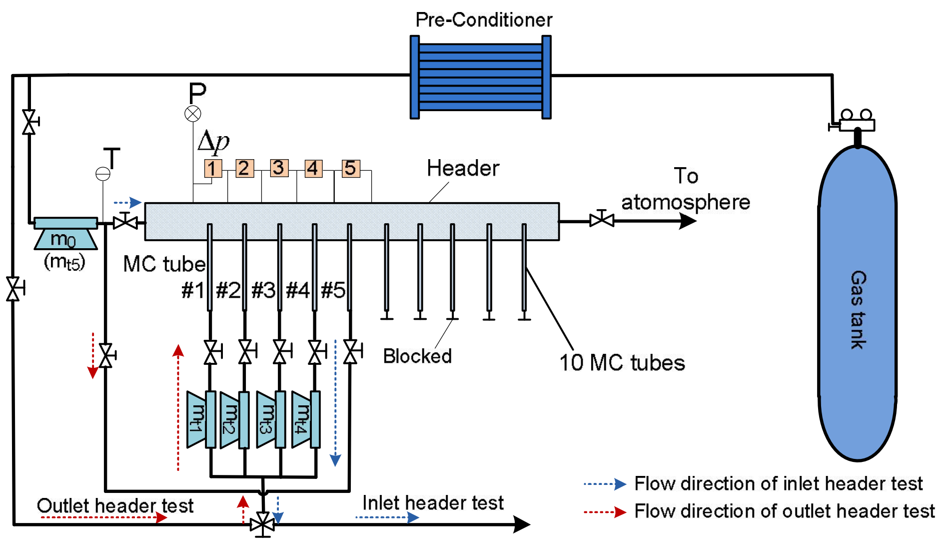


Figure 1: Test apparatus

The header consists of a transparent PVC tube and ten micro-channel tubes. The tube spacing is 12.0 mm and the protrusion is 50%. The inner diameter of the PVC tube is 18.4 mm and length is 200 mm. The micro-channel tube has 23 ports with 1.5 mm in thickness and the 17.9 mm in width. The length and width of each port is 0.84 mm and 0.64 mm, respectively.

3. DATA REDUCTION

The pressure drop at i^{th} section shown in Figure 2 consists of the diverging/converging loss pressure drop $\Delta p_{\zeta,\,i}$, acceleration pressure drop $\Delta p_{acc,\,i}$ and frictional pressure drop $\Delta p_{f,\,i}$, as shown in Equation (1). In Equation (1), the acceleration pressure drop and friction pressure drop are computed by Equation (2) and Equation (3), respectively. Thus, the diverging/converging loss coefficient in the i^{th} section is defined by the velocity at i^{th} section (see Figure 2) by Equation (4).

$$\Delta p_{\zeta,\,i} = \Delta p_i - \Delta p_{acc,\,i} - \Delta p_{f,\,i} \tag{1}$$

$$\Delta p_{acc,\,i} = \rho v_{c,i}^2 / 2 - \rho v_{c,i\text{-}1}^2 / 2 \tag{2}$$

$$\Delta p_{\mathrm{f,\,i}} = \lambda_{\mathrm{i-1}} \frac{l_i}{2D_{\mathrm{h}}} \frac{S_{\mathrm{eff}}}{S_{\mathrm{tot}}} \rho v_{\mathrm{c,i-1}}^2 / 2 + \lambda_{\mathrm{i}} \frac{l_i}{2D_{\mathrm{h}}} \frac{S_{\mathrm{eff}}}{S_{\mathrm{tot}}} \rho v_{\mathrm{c,i}}^2 / 2 \tag{3}$$

$$\zeta_{\mathrm{i}} \equiv \begin{cases} \dfrac{\Delta p_{\zeta,\,\mathrm{i}}}{\rho v_{\mathrm{c,i}}^2 / 2} & \text{Converging case} \\ \dfrac{\Delta p_{\zeta,\,\mathrm{i}}}{\rho v_{\mathrm{c,i-1}}^2 / 2} & \text{Diverging case} \end{cases} \tag{4}$$

where ρ is the density; $v_{\mathrm{c,i-1}}$ and $v_{\mathrm{c,i}}$ are the average velocities of flow through header at sections i-1 and i, respectively; $\lambda_{\mathrm{i-1}}$ and λ_{i} are the friction loss coefficient at sections i-1 and i, respectively; l_i is the header length of i$^{\mathrm{th}}$ section; D_{h} is hydraulic diameter of header; S_{eff} and S_{tot} are the effective friction perimeter and total perimeter of header, respectively, as shown in Figure 2.

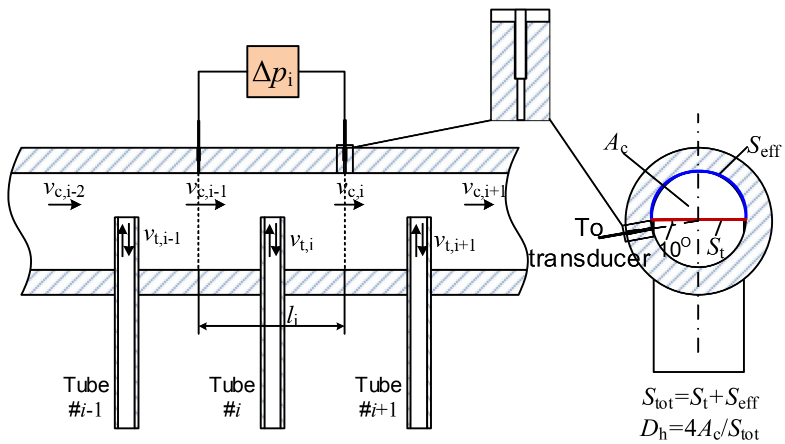


Figure 2: Schematic drawing of header at i$^{\mathrm{th}}$ section

4. RESULTS AND DISCUSSION

4.1 Inlet header

Figure 3 shows the effect of velocity through the i$^{\mathrm{th}}$ micro-channel tube on Δp_{i}. Figure 3(a) shows that under a certain flow velocity through header (v_{c0}), the velocity through micro-channel tube (v_{t1}) decreases the pressure drop (Δp_1) when flows pass over micro-channel tube #1. This is because the contraction makes a very strong secondary flow between the micro-channel tubes #1 and #2, and as the velocity through micro-channel tube #1 increases, the suction force from the micro-channel tube becomes stronger and pulls some of fluid into the volume between the micro-channel tubes #1 and #2 and below the line S_{t} shown in Figure 2. So, the flow through micro-channel tube affects the formation of secondary flow in the space behind. Figures 3(b) to 3(d) show that Δp_{i} decreases as the velocity through the i$^{\mathrm{th}}$ micro-channel tube increases. It is due to the increase of deceleration pressure drop.

Figure 4 shows the effect of the velocity through upstream micro-channel tubes on Δp_{i}. It shows that the velocity through micro-channel tube #1 (v_{t1}) has significant impact on Δp_2, but has no impact on Δp_3, as shown in Figures 4(a) and 4(b). When v_{t1} is low, the contraction effect is dominant, and an eddy zone is formed between micro-channel tubes #1 and #2, resulting in a local low pressure region in that area and a negative Δp_2. When v_{t1} increases, the eddy zone between micro-channel tubes #1 and #2 will be affected because the pressure behind the micro-channel tube #1 will increase and reduce pressure drop. The results of experiment also indicate that the velocities through micro-channel tubes #2 and #3 have no impact on Δp_3 and Δp_4, respectively, as shown in Figures 4(c) and 4(d). This is because velocity profile is more developed and so the effect of diverging flow through the tube on eddy zone behind the tube has almost no impact on the main flow through the header.

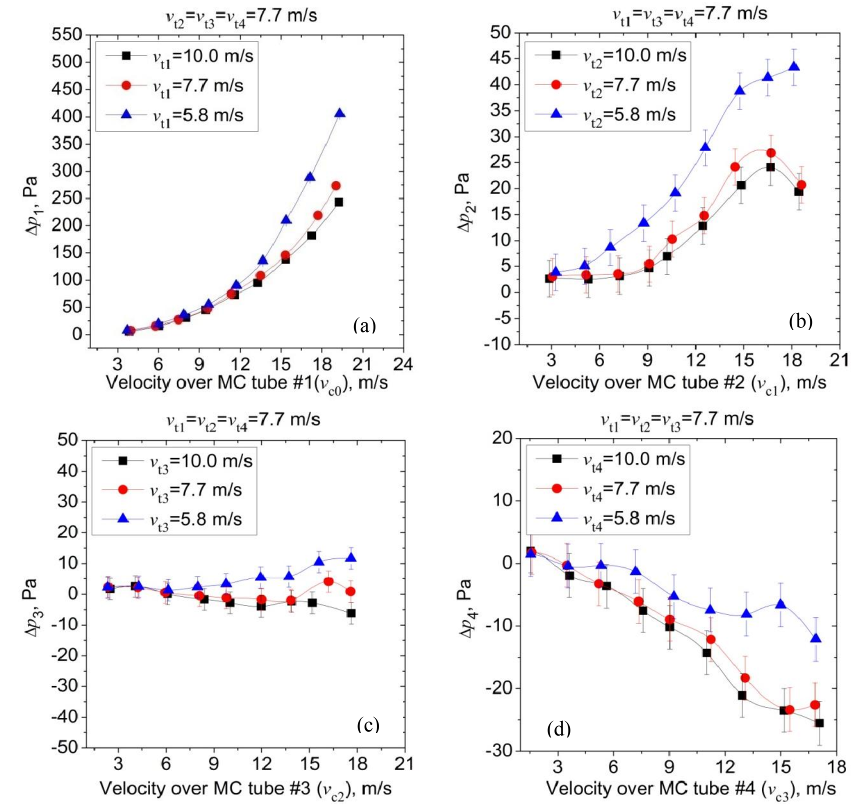


Figure 3: Effect of velocity on Δp_i in the i[th] micro-channel tube

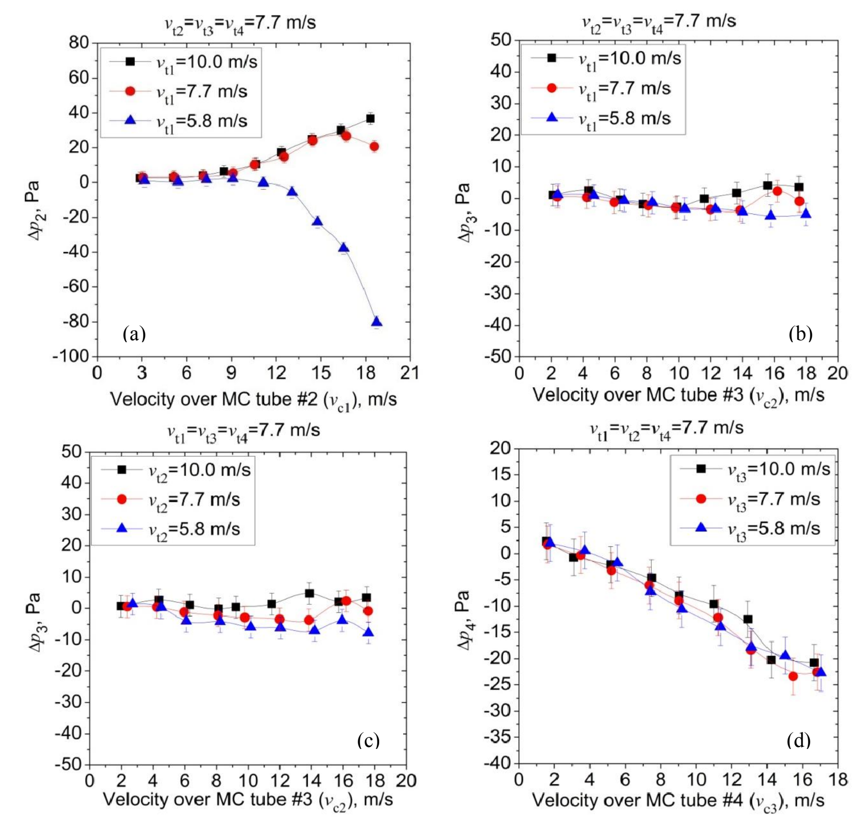


Figure 4: Effect of velocity through upstream micro-channel tubes on Δp_i of inlet header

Figure 5 shows the effect of the velocity through downstream micro-channel tubes $v_{t,\,i+1}$ on Δp_i. It shows that the velocity through downstream micro-channel tubes has no impact on the pressure drop. This is because the flow separation and eddy zone are formed after the flow diverging (Idelchik, 1994).

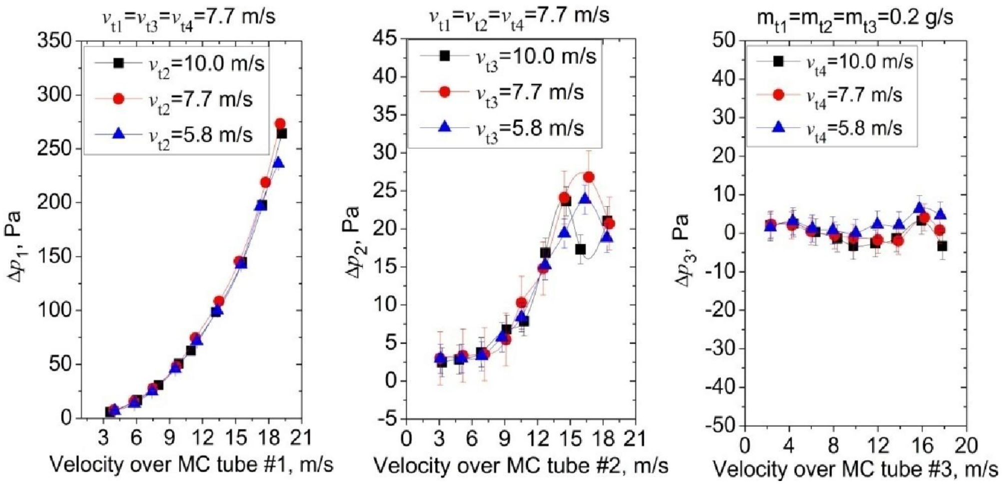


Figure 5: Effect of velocity through downstream micro-channel tubes on Δp_i

4.2 Outlet header

Figure 6 shows the effect of velocity through the i[th] micro-channel tube on Δp_i of outlet header. It shows that Δp_i increases as a quadratic function of velocity through the i[th] micro-channel tube.

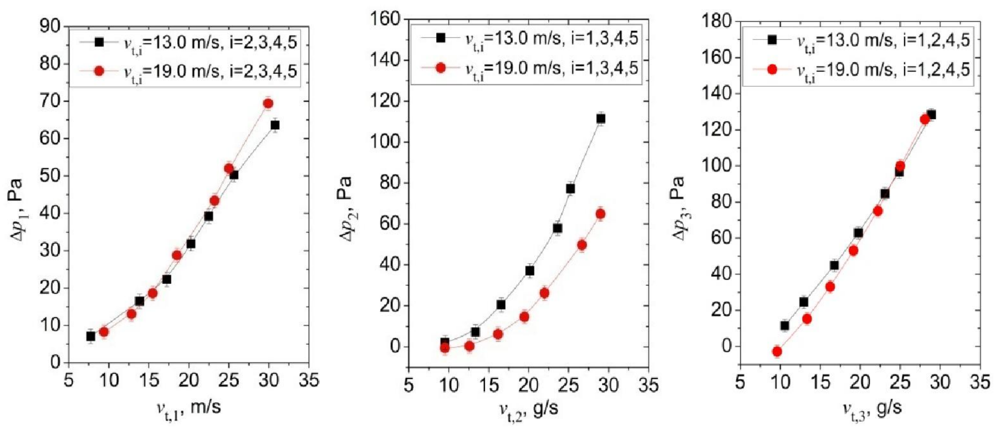


Figure 6: Effect of velocity through the *i*[th] micro-channel tube on Δp_i in outlet header

Figure 7 shows the effect of velocity through upstream micro-channel tubes on Δp_i of outlet header. It shows that Δp_i of outlet header decreases as the velocities through (i-1)[th] and (i-2)[th] micro-channel tubes increases. The reason is that as the velocity in upstream micro-channel tube increases, v_{ci} increase and consequently acceleration pressure drop decreases. In addition, this results in decreasing of v_{ti}/v_{ci}, resulting in the decrease of converging loss coefficient of current section (ζ_i), as shown in Figure 12.

Figure 8 shows the effect of velocity through downstream micro-channel tube on Δp_i of outlet header. It shows that like the inlet header case, the velocity through the (i+1)[th] micro-channel tube has no impact on Δp_i of outlet header except the velocity through micro-channel tube #2. As shown in Figure 8(a), the only different case is at high velocity through tube #2 (v_{t2}) and low velocity through tube 1 or through header (v_{c1}). The velocity through the (i+1)[th] micro-channel tube is so large that the flow through it almost blocks the flow through header.

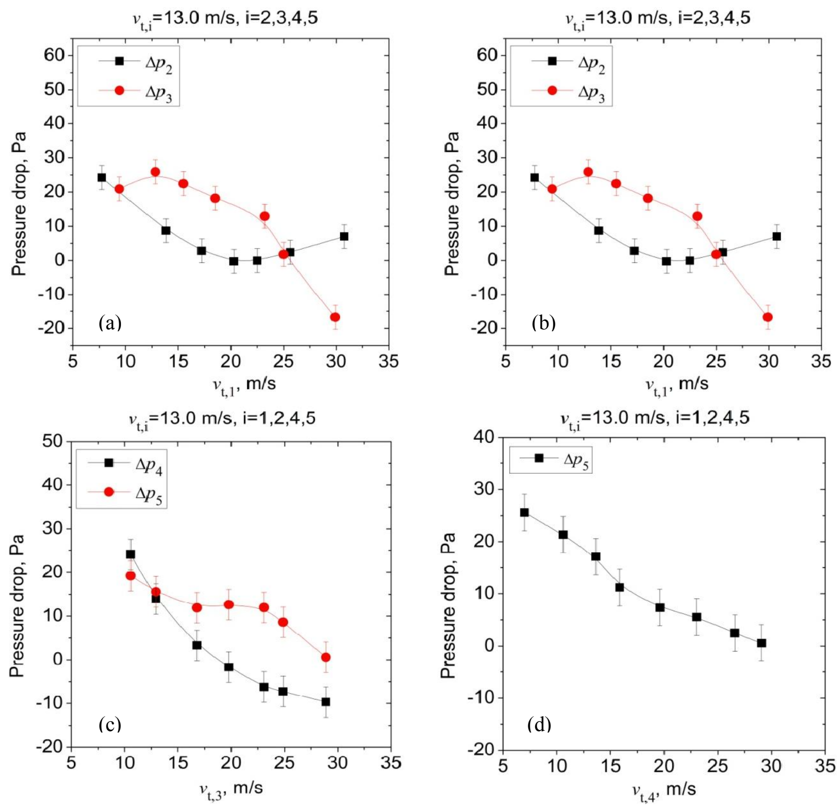


Figure 7: Effect of velocity through the upstream micro-channel tubes on Δp_i in outlet header

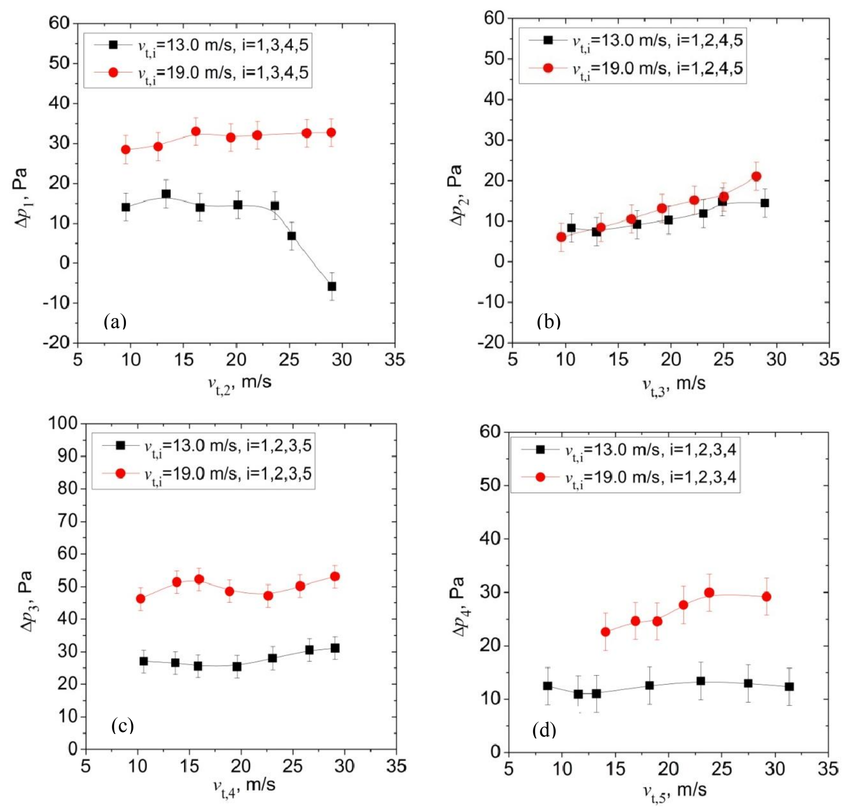
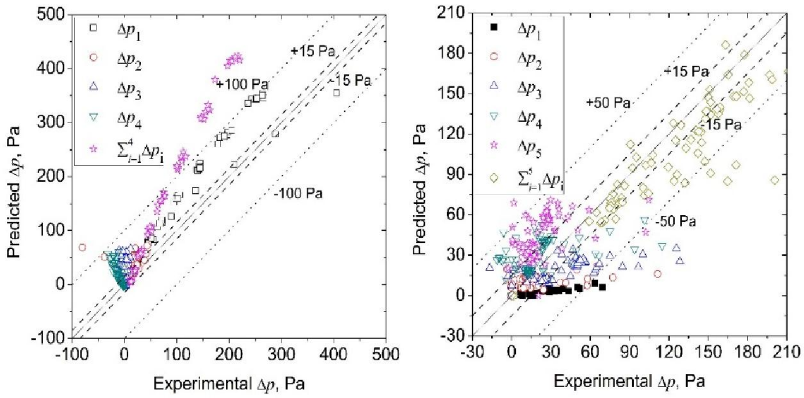


Figure 8: Effect of velocity through the (i+1)th micro-channel tube on Δp_i in outlet header

5. DEVELOPMENT OF THE CORRELATION

Until now, there is no correlation to reflect non-uniformity of pressure loss coefficient in headers. Yin et al. (2002) measured pressure drop in headers and generated a correlation based on the assumption that the pressure loss coefficients are uniform. As shown in Figure 9, Yin's correlation can provide satisfactory prediction to the total pressure drop of headers for the present experimental data, however, it is not suitable to predict the pressure drop of i[th] section (Δp_i) for both inlet and outlet headers. The possible reasons is that the pressure drop of i[th] section (Δp_i) for both inlet and outlet headers varies with locations and is a function of velocity through micro-channel tube and that through header. Therefore, the non-uniformity of pressure loss coefficient in headers and the impacts of velocities through micro-channel tube and headers should be reflected in the new correlation.



(a) Inlet header (b) outlet header

Figure 9: Comparison of the current experimental data with the predicted values obtained by Yin's correlation

5.1 Correlation for inlet header:

Based on the experimental results we can say that the pressure drop of i[th] section (Δp_i) is function of the velocity through the i[th] micro-channel tube ($v_{t,\,i}$) and the velocity though header before flow diverging at i[th] section ($v_{c,\,i-1}$). However, the velocities through the (i-1)[th] micro-channel tube ($v_{t,\,i-1}$) and (i+1)[th] micro-channel tube ($v_{t,\,i+1}$) has no impact on the pressure drop of i[th] section (Δp_i), except $v_{t,1}$ which has great impact on Δp_2 due to the contraction.

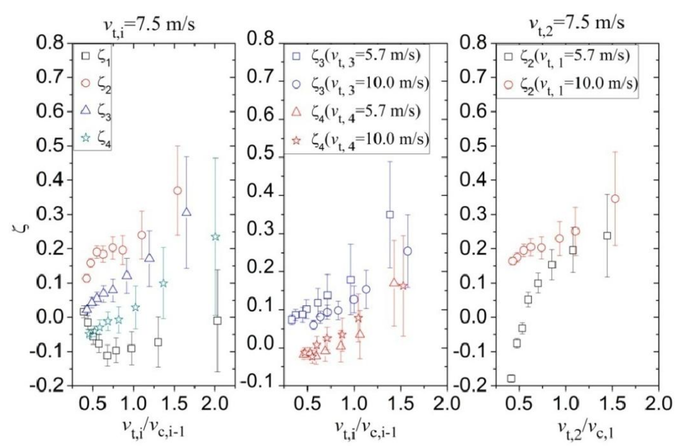
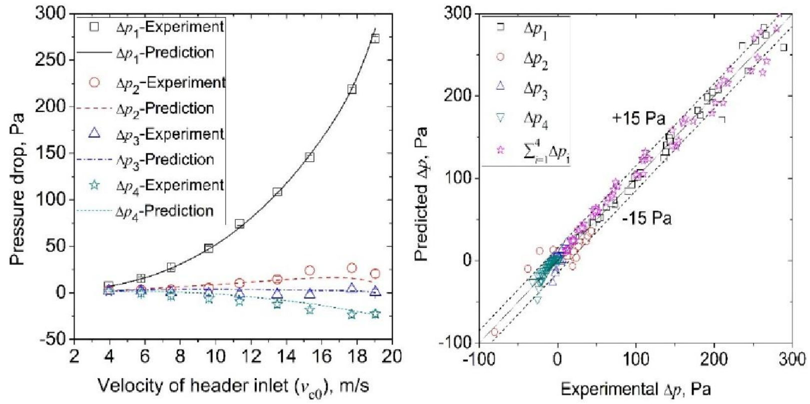


Figure 10: Pressure loss coefficient is a function of $v_{t,i}/v_{c,i-1}$, $v_{t,i}$, $v_{t,i-1}$ and location

Figure 10 shows the relations between the pressure loss coefficient of i[th] section (ζ_i) and the parameters of the i[th] micro-channel tube ($v_{t,\,i}$), the velocity though header before flow diverging at i[th] section ($v_{c,\,i-1}$) and locations, and the relations between ζ_2 and $v_{t,1}$. It shows that if "i" is equal or larger than 3, ζ_i is a linear function of $v_{t,\,i}/v_{c,\,i-1}$. As a result, the correlation for ζ_i should describe the pressure loss coefficient of contraction and diverging flow separately for ζ_1 and ζ_2, while others ζ_i (i≥3) can be treated equally as a linear function of $v_{t,\,i}/v_{c,\,i-1}$ while adding the effect of and location. Based on that the correlation for pressure loss coefficient in inlet header ζ_i is expressed as Equation (5). Using nonlinear fitting method the values of a_1 to a_{14} in Equation (5) are found to be: -14.582, 4.017, 0.111, -0.218, -24.230, 7.261, 0.242, -0.031, 0.269, 0.297, -0.044, 17.340, -1.715 and 0.165, respectively.

$$\zeta_i \equiv \frac{\Delta p}{\rho v_{c,\,i\text{-}1}^2/2} = \begin{cases} 0.75\exp\left(a_1\dfrac{v_{t,\,i}}{v_{c,\,i\text{-}1}}+a_2\right)+a_3\dfrac{v_{t,i}^2}{v_{c,i\text{-}1}^2}+a_4\dfrac{v_{t,\,i}}{v_{c,\,i\text{-}1}} & \text{if } i=1 \\ -0.4\exp\left(a_5\dfrac{v_{t,\,i\text{-}1}}{v_{c,\,i\text{-}2}}+a_6\right)+a_7\dfrac{v_{t,\,i}}{v_{c,\,i\text{-}1}}+a_8 v_{t,\,i}+a_9 & \text{if } i=2 \\ a_{10}\dfrac{v_{t,\,i}}{v_{c,\,i\text{-}1}}+a_{11}v_{t,\,i}+a_{12}\exp\left(a_{13}\cdot i\right)+a_{14} & \text{if } i\geq 3 \end{cases} \tag{5}$$

The correlation is validated by the experimental data, as shown in Figure 11. The predicted pressure drops of ith section of inlet header (Δp_i) using the correlation agree with 98% of the experimental data (292 points) obtained in the present study within a deviation of ±15 Pa, as shown in Figure 11(b). In addition, the predicted total pressure drop ($\Sigma\Delta p_i$) agrees with 95% of experiment data within a deviation of ±15 Pa, which means the developed correlation has a good accuracy to predict the total pressure drop of header.



(a) Predicted pressure drop along header (b) Accuracy of predicted pressure drop

Figure 11: Comparison of the predicted values with the experimental pressure drops in inlet header

5.2 Correlation for outlet header:

Based on the experimental results we can say that the pressure drop of ith section (Δp_i) is function of the velocities through the ith, (i-1)th and (i-2)th micro-channel tubes ($v_{t,\,i}$, $v_{t,\,i\text{-}1}$ and $v_{t,\,i\text{-}2}$) and the velocity though header before flow diverging at ith section ($v_{c,\,i\text{-}1}$). However, the velocity through the (i+1)th micro-channel tube ($v_{t,\,i+1}$) has no impact on the pressure drop of ith section (Δp_i).

Figure 12 shows that the pressure loss coefficient of ith section (ζ_i) is a quadratic function of $v_{t,\,i}/v_{c,\,i}$. In addition, when "i" is equal or larger than 3, ζ_i follows the same trend with $v_{t,\,i}/v_{c,\,i}$. As a result, the correlation for ζ_i has to be based on the fact that ζ_i is a quadratic function of $v_{t,\,i}/v_{c,\,i}$, and also $v_{t,\,i\text{-}1}/v_{c,\,i}$ and $v_{t,\,i\text{-}2}/v_{c,\,i}$ and in addition has the effect of location. Based on that, the formulation of ζ_i is expressed as Equation (6), and by using nonlinear fitting method, the values of a_1 to a_9 in Equation (6) are from our data determined to be: 0.048, -0.888, -1.273, 3.352, 0.059, -0.221, -0.276, -0.112 and 0.252 respectively.

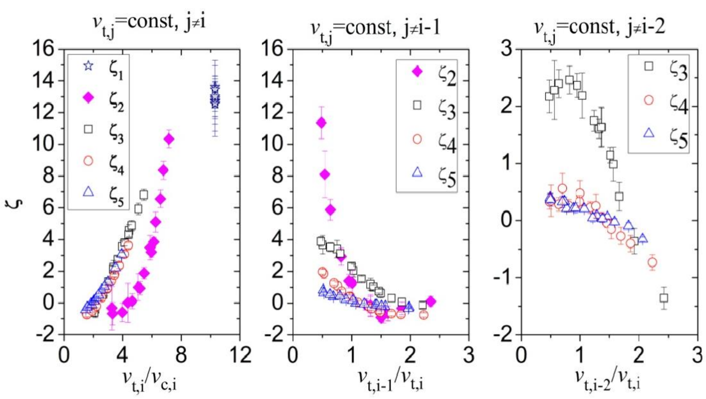
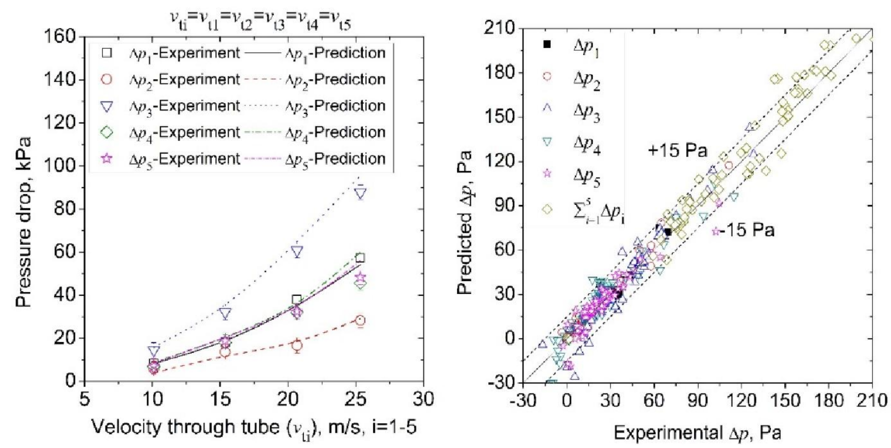


Figure 12: Pressure loss coefficient is a function of $v_{t,i}/v_{c,i}$, $v_{t,i\text{-}1}/v_{t,i}$, and $v_{t,i\text{-}2}/v_{t,i}$

$$\zeta_{\mathrm{i}} \equiv \frac{\Delta p}{\rho v_{\mathrm{c,\,i}}^2/2} = \begin{cases} 0.125\dfrac{v_{\mathrm{t,\,i}}^2}{v_{\mathrm{c,\,i}}^2} & \text{if } i=1 \\ \dfrac{v_{\mathrm{t,\,i}}v_{\mathrm{t,\,i\text{-}1}}}{v_{\mathrm{c,\,i}}^2}\cdot\left(a_1\dfrac{v_{\mathrm{t,\,i}}^2}{v_{\mathrm{c,\,i}}^2}+a_2\dfrac{v_{\mathrm{t,\,i}}}{v_{\mathrm{c,\,i}}}+a_3\ln\dfrac{v_{\mathrm{t,\,i\text{-}1}}}{v_{\mathrm{t,\,i}}}+a_4\right) & \text{if } i=2 \\ \dfrac{v_{\mathrm{t,\,i\text{-}1}}v_{\mathrm{t,\,i\text{-}2}}}{v_{\mathrm{c,\,i}}^2}\cdot\left(a_5\dfrac{v_{\mathrm{t,\,i}}^2}{v_{\mathrm{c,\,i}}^2}+a_6\dfrac{v_{\mathrm{t,\,i}}}{v_{\mathrm{c,\,i}}}+a_7\ln\dfrac{v_{\mathrm{t,\,i\text{-}1}}}{v_{\mathrm{t,\,i}}}+a_8\ln\dfrac{v_{\mathrm{t,\,i\text{-}2}}}{v_{\mathrm{t,\,i}}}+a_9\right) & \text{if } i\geq 3 \end{cases} \tag{6}$$

The correlation is validated by the experimental data, as shown in Figure 13. The correlation well predict the pressure drop along the headers as shown in Figure 13(a), and the predicted pressure drops of outlet header using the correlation agree with 98% of the experimental data (393 data points) obtained in the present study within a deviation of ±15 Pa as shown in Figure 13(b). In addition, the predicted total pressure drop ($\Sigma\Delta p_{\mathrm{i}}$) agrees with 90% of experiment data within a deviation of ±15 Pa.



(a) Predicted pressure drop along header (b) Accuracy of predicted pressure drop

Figure 13: Comparison of the predicted values with the experimental pressure drops of outlet header

6. SUMMARY AND CONCLUSION

The paper presented investigation of pressure drop and development of the correlation for pressure loss coefficients for single phase flow through round cylindrical inlet and outlet headers of micro-channel parallel flow heat exchangers. The correlation is experimentally validated for almost 600 cases and the deviation of pressure drop is within ±15Pa. Some other findings are as follows:

- The pressure drop of inlet header at the i$^{\mathrm{th}}$ section (Δp_{i}) is function of the velocity through the i$^{\mathrm{th}}$ micro-channel tube ($v_{\mathrm{t,\,i}}$) and the velocity though header before flow diverging at i$^{\mathrm{th}}$ section ($v_{\mathrm{c,\,i\text{-}1}}$). The velocities through the (i-1)$^{\mathrm{th}}$ micro-channel tube ($v_{\mathrm{t,\,i\text{-}1}}$) and the (i+1)$^{\mathrm{th}}$ micro-channel tube ($v_{\mathrm{t,\,i+1}}$) have no impact on the pressure drop of i$^{\mathrm{th}}$ section (Δp_{i}).
- The pressure drop of the outlet header at the i$^{\mathrm{th}}$ section (Δp_{i}) is a function of the velocity through the i$^{\mathrm{th}}$ micro-channel tube ($v_{\mathrm{t,\,i}}$) and the velocity though header after flow converging at i$^{\mathrm{th}}$ section ($v_{\mathrm{c,\,i}}$), and the velocities through the (i-1)$^{\mathrm{th}}$ and (i-2)$^{\mathrm{th}}$ micro-channel tubes ($v_{\mathrm{t,\,i\text{-}1}}$ and $v_{\mathrm{t,\,i\text{-}2}}$) have great impact on the pressure drop of i$^{\mathrm{th}}$ section (Δp_{i}).
- The diverging pressure loss coefficient of inlet header ζ_{i} is a linear function of $v_{\mathrm{t,\,i}}/v_{\mathrm{c,\,i\text{-}1}}$.
- The converging pressure loss coefficient of outlet header ζ_{i} is a quadratic function of $v_{\mathrm{t,\,i}}/v_{\mathrm{c,\,i}}$, and ζ_{i} follows the same trend with $v_{\mathrm{t,\,i}}/v_{\mathrm{c,\,i}}$ when i is equal or larger than 3.

NOMENCLATURE

D	Diameter (m)	**Subscript**	
l	Length (m)	acc	Acceleration
S	Perimeter (m)	c	Confluence flow in header
v	Velocity (m/s)	eff	Effective friction loss
Greek symbols		f	Friction
ζ	Diverging/converging loss coefficient	h	Hydraulic
λ	Frictional loss coefficient	i	i^{th} section of header
ρ	Density (kg m^{-3})	t	Tube
Δp	Pressure drop (Pa)	tot	Total

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