



## Experiments on Measurement of Heat Transfer Rate of Gas Flow Through Microchannel with Constant Wall Temperature

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### ABSTRACT

Experimental results for the measurement of heat transfer rates in microchannel gas flow are crucial for gas to gas micro heat exchangers. A critical overview of the main factors that play an important role in the determination of heat transfer rate is presented. The experimental and numerical data obtained from authors' previous studies are used in order to highlight the characteristics of convective heat transfer of gas through microchannels with constant wall temperature. It is suggested to obtain heat transfer rates by determining the difference between the gas enthalpy at the inlet and outlet by measuring local temperature and pressure. The heat transfer rates obtained in the present study were compared with those determined by the difference in total temperatures and incompressible flow Nusselt number.

#### Keywords:

Heat transfer rate; gas flow; convective heat transfer; total enthalpy and temperature; microtube

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## 1. Introduction

Understanding of heat transfer characteristics in microchannels is becoming an important topic in the miniaturization of electronic devices [1]. Since the pioneer work of Tuckerman and Pease [2], numerous experimental and numerical investigations on convective heat transfer of microchannel flow have been undertaken.

In the case of microchannel liquid flow, flow and heat transfer characteristics can be accurately predicted by conventional correlations [3,4], with minor deviations caused by experimental errors [5].

However, in the case of microchannel gas flow, it is well understood that rarefaction, surface roughness and compressibility significantly affect the flow and heat transfer characteristics

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separately or simultaneously [6,7]. The effect of compressibility is more dominant on flow and heat transfer characteristics than those of surface roughness and rarefaction for microchannels with hydraulic diameter longer than  $10\ \mu\text{m}$ . The compressibility effect leads to a significant flow acceleration along the microchannel length and a large gas expansion near the outlet. This results in a decrease of the gas temperature and an additional heat transfer from the wall to the gas near the outlet [8]. Therefore, conventional correlations for the prediction of Nusselt number to obtain heat transfer fail in the presence of significant compressibility effect [9].

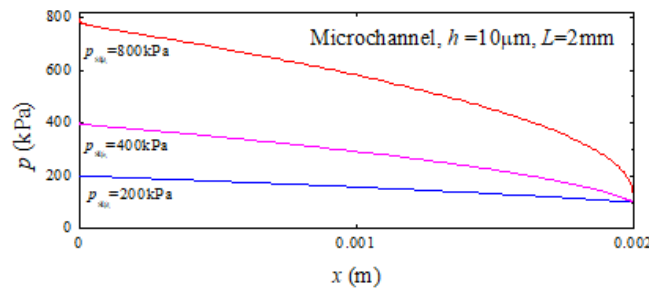
Experimental investigations on heat transfer characteristics of gas flow in microchannels have been performed by a number of researchers using most common thermal boundary conditions, namely constant heat flux (CHF) and constant wall temperature (CWT) with emphasis on CHF [9-12]. The literature was well surveyed and critically reviewed by Lin *et al.*, [10] and Morini *et al.*, [11]. It is reported that the thermal behavior at microscale of gas and liquid flows through microchannel in terms of convective heat transfer coefficients can be strongly affected by scaling and micro-effects but also by practical issues linked to the geometry of the testing, the real thermal boundary conditions, the presence of fittings and position and type of the sensors, and so on [11]. All these aspects have to be taken into account during the data post processing in order to obtain the correct evaluation of heat transfer characteristics. Choi *et al.*, [13] carried out experiment using nitrogen gas flow in micro-tubes with CWT with inner diameters ranging from  $3\ \mu\text{m}$  to  $81\ \mu\text{m}$ . They observed very low values of Nusselt number compared to the prediction of the conventional correlations. However, the detailed measurement of gas temperature was not described. This motivated the present study having the goal to measure the total temperature to quantitatively determine the heat transfer rate through a micro-tube with constant wall temperature with an overview on authors' previous studies.

## 2. Convective Heat Transfer of Gas Flow Through Microchannels with CWT

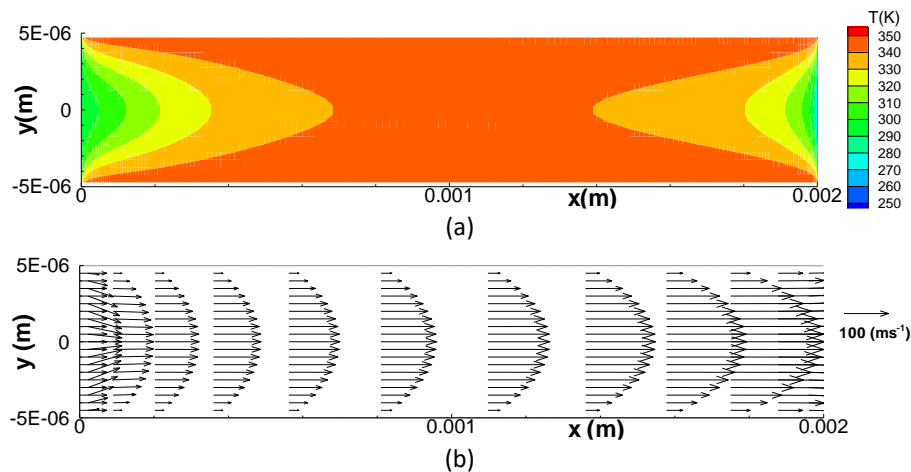
The axial pressure distributions obtained numerically for a typical gas flow through a parallel plate microchannel of  $h=10\ \mu\text{m}$  and  $L=0.002\ \text{m}$  is plotted in Figure 1 [14].

The channel walls are maintained to an uniform value ( $T_w=350\ \text{K}$ ) and the gas enters with a lower stagnation temperature ( $T_{\text{stg}}=300\ \text{K}$ ). Under the conditions investigated in [14], the compressibility effects are evident by observing the non-linear axial trend of the pressure along the channel (Figure 1) obtained for large values of the stagnation gas pressure at the inlet. The pressure gradient becomes steep due to flow acceleration near the outlet as the stagnation pressure increases. By reducing the value of the inlet stagnation gas pressure (i.e.  $p_{\text{stg}}=200\ \text{kPa}$ ), the pressure distribution was almost linear, and the compressibility effects are negligible. Increasing the inlet gas pressure, the pressure distribution becomes more and more nonlinear due to the compressibility effects. For the stagnation pressure,  $p_{\text{stg}}=800\ \text{kPa}$  in the figure, velocity vectors and the contour plots of the temperature are presented in Figure 2(a) and Figure 2(b), respectively. The reference arrow in the figure represents a velocity of  $100\ \text{m/s}$ . Figure 2(a) shows that the velocity profile becomes nearly parabolic because of the viscosity. Beyond the entrance region, however, the flow in the channel is accelerated and the average velocity increases. This is due to the volume expansion of the gas caused by large pressure drop. Figure 2(b) represents the temperature contours in the channel. The temperature in the core region of the channel decreases as a result of energy conversion into kinetic energy causing an additional heat transfers from the wall to gas. Therefore, the heat flux from the wall along the length in Figure 2(b) is plotted in Figure 3. It is possible to highlight that the heat flux has a positive value when the gas temperature is lower than the wall temperature. In the case of slow flow ( $p_{\text{stg}}=200\ \text{kPa}$ ) represented by the blue line, the heat flux from the wall converges to zero asymptotically similar to that of the incompressible flow. On the other hand, in the case of fast flow

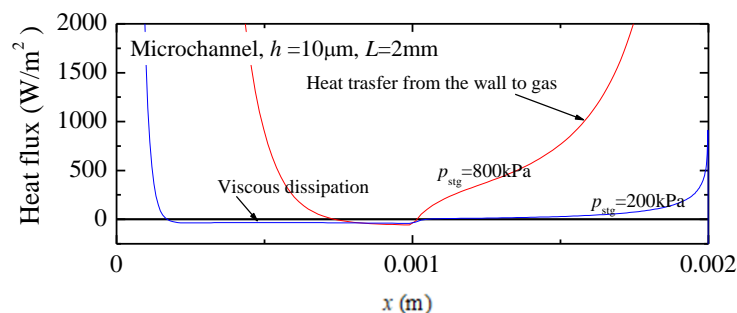
( $p_{stg} = 800$  kPa) represented by the red line, part of the thermal energy converts into kinetic energy near the outlet. This results in a decrease in temperature and additional heat transfer from the wall to the gas near the outlet. That increases the heat flux near the outlet. Therefore, the total temperature is higher than the wall temperature. This is numerically and experimentally demonstrated in a series of previous studies [8,14-18].



**Fig. 1.** Pressure distributions as a function of  $x$



**Fig. 2.** (a) Contour plot of temperature and (b) velocity vector



**Fig. 3.** Heat flux from the wall

### 3. Heat Transfer Rate

The heat transfer rate of gas flow is expressed as [19]

$$\dot{Q}_h = \dot{m} (h_{T,out} - h_{stg}) \quad (1)$$

For an ideal gas, the enthalpy is a function of temperature as

$$dh = c_{p,ave} dT \quad (2)$$

And the heat transfer rate is

$$\dot{Q}_T = \dot{m} c_p (T_{T, out} - T_{stg}) \quad (3)$$

For the case of slow flow of an ideal gas, the heat transfer rate can be obtained since the bulk temperature and the total temperature are equal by means of,

$$\dot{Q}_{slow} = \dot{m} c_p (T_{b,incomp} - T_{in}) \quad (4)$$

where the bulk temperature for incompressible flow is expressed as a function of mean Nusselt number and axial location as (e.g., Burmeister [20])

$$T_{b,incomp} = T_w - (T_w - T_{stg}) e^{-4Nu_m X^*} \quad (5)$$

The mean Nusselt number of a turbulent fully developed flow in a duct is defined by [21]

$$Nu_m = 0.022 Re^{0.8} Pr^{0.5} \quad (6)$$

$X^*$  is the inverse of Graetz number, defined by

$$X^* = \frac{x}{D Re Pr} \quad (7)$$

Many researchers have investigated the turbulent heat transfer characteristics of duct flows and Nusselt numbers in the literature (e.g., Kays and Crawford [21] and Shah and London [22]). In the present study, the Nusselt number obtained by Kays and Crawford [21] in the range of  $0.5 < Pr \leq 1.0$  for a fully developed flow in tubes was used for the calculation of the bulk temperature for incompressible flow.

In order to obtain heat transfer rates, the total enthalpy difference between the inlet and outlet is used as show in Eq. (1). The enthalpy of a gas is determined from the pressure and the temperature,  $h = h(p, T)$ . Therefore, to obtain the total enthalpy at the microtube outlet, a device was fabricated with similar structure to that proposed by Yamada *et al.*, [23] for our recent studies [24-26] as shown in Figure 4.

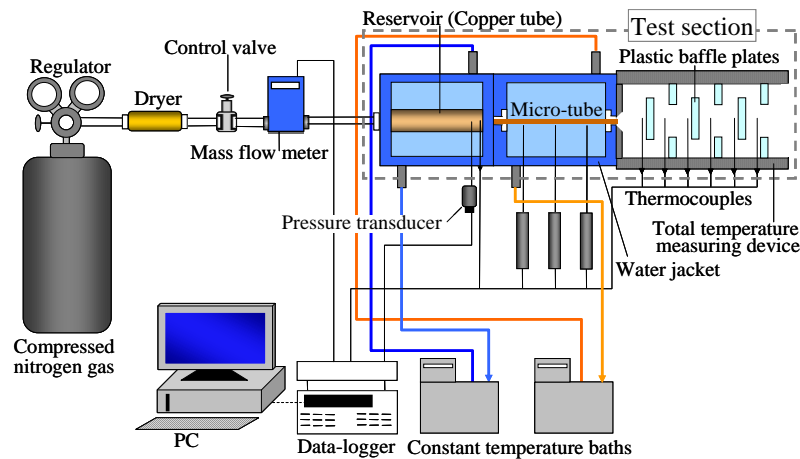
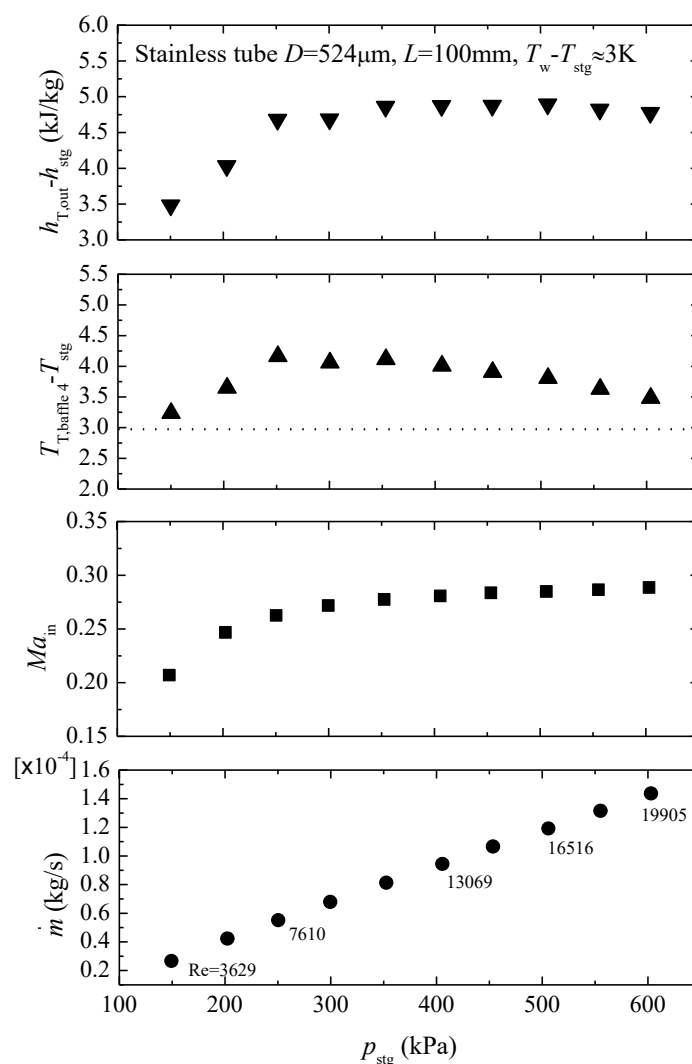


Fig. 4. A schematic diagram of experimental setup

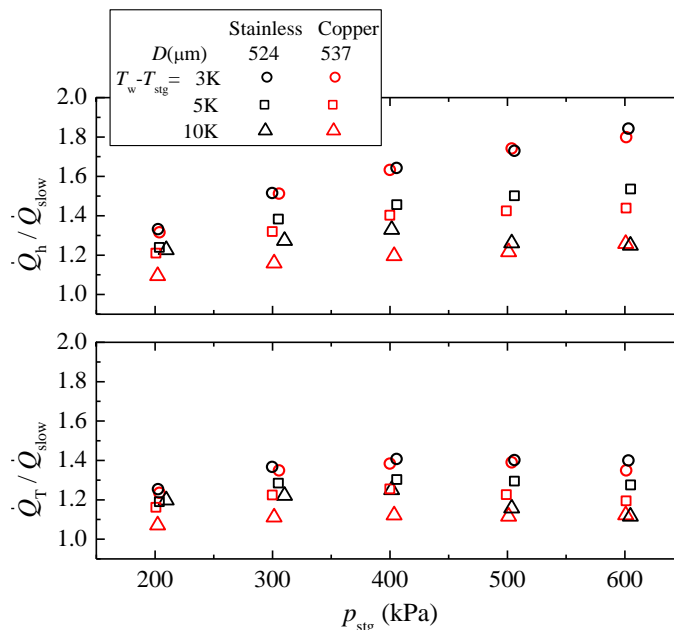
The heat transfer in the figure consists of three components: an inlet tube, a microtube and a temperature measuring device. The inlet tube made of copper is placed inside the upper water jacket to control the gas temperature. The pressure transducer and the thermocouple are inserted in the inlet tube in order to measure the gas temperature and pressure. The measured gas temperature and pressure are considered to be the stagnation temperature,  $T_{stg}$  and the stagnation pressure,  $p_{stg}$ , since the gas is at rest. The microtube is changed from a test to another one using different tube material (stainless steel, copper and fused silica) and diameters ( $D \cong 150 \sim 530\mu\text{m}$ ) microtubes [24-26]. The microtube is placed inside the lower water jacket to keep the wall temperature constant. Water is circulated between the water jacket and a thermostatic bath. A thermally insulated exterior foamed polystyrene tube with six baffle plates fabricated where the gas velocity reduces and the kinetic energy is converted into the thermal energy, was attached to the outlet of the microtube as a total temperature measuring device. More details of the total temperature measuring device are documented in Yamada *et al.*, [23] and our recent studies [24-26].

Here, Figure 5 shows the measured total temperature difference between the outlet and the stagnation temperature,  $T_{T,out} - T_{stg}$  as a function of the stagnation pressure. The temperature difference between the wall and the stagnation temperature,  $T_{wall} - T_{stg}$  is also plotted in the figure with a dotted line. The total enthalpy difference between the micro-tube outlet and the stagnation,  $h_{T,out} - h_{stg}$  is plotted in the figure with the values of the mass flow rate and the inlet Mach number. The enthalpy was obtained as a function of pressure and temperature. The figure is the results for the case of  $T_w \cong 294\text{K}$  and  $T_{stg} \cong 297\text{K}$  ( $T_w - T_{stg} \cong 3\text{K}$ ). This flow is turbulent flow since  $3630 \leq Re \leq 19900$ . The measured total temperature at the microtube outlet is higher than wall temperature since additional heat was transfer from the wall to gas due to thermal energy conversion into kinetic energy. Similar results were obtained by the previous numerical study [16]. Inlet Mach number increases as the stagnation pressure increase, and nearly levels off in the range of  $p_{stg} \geq 300\text{kPa}$ . This reflects that the flow becomes choked under higher stagnation pressure. The value of  $T_{T,out} - T_{stg}$  increases with increase in the stagnation pressure and slightly decrease in the range of  $p_{stg} \geq 300\text{kPa}$  due to gas expansion from a large pressure difference between the stagnation and the outlet pressure. However, the value of  $h_{T,out} - h_{stg}$  increases with increased stagnation pressure and maintains a specific value in the range of  $p_{stg} \geq 300\text{kPa}$ . The value of  $h_{T,out} - h_{stg}$  remains almost constant in the range when the flow is choked even though the stagnation pressure increases. The enthalpy difference between the inlet and the outlet of the tube corresponds to the heat transfer rate from the wall. For the reason, the heat transfer rate remains unchanged even if the flow becomes a choked flow.

The heat transfer rates were obtained by Eq. (1), Eq. (3) and Eq. (4) with measured data. The heat transfer rates normalized by  $\dot{Q}_{\text{slow}}$ ,  $\dot{Q}_h/\dot{Q}_{\text{slow}}$  and  $\dot{Q}_T/\dot{Q}_{\text{slow}}$  are plotted as a function of the stagnation pressure in Figure 6. Part of the thermal energy converts into kinetic energy near the outlet when the flow is fast. This results in the decrease of the bulk temperature and addition of heat transfer from the wall to gas. As results of this both values of  $\dot{Q}_h/\dot{Q}_{\text{slow}}$  and  $\dot{Q}_T/\dot{Q}_{\text{slow}}$  are higher than unity. In the case of  $T_w - T_{\text{stg}} \cong 10\text{K}$ , the maximum increment of both values due to addition heat transfer is less than 20 %. However, in the cases of  $T_w - T_{\text{stg}} \cong 3\text{K}$  and  $T_w - T_{\text{stg}} \cong 5\text{K}$ , the maximum increment is more than 80% and 40 %, respectively. The value of  $\dot{Q}_T/\dot{Q}_{\text{slow}}$  increases with the stagnation pressure and levels off for choked flows ( $p_{\text{stg}} \geq 300 \text{ kPa}$ ). Also, for choked flows, the measured  $T_{T, \text{out}}$  slightly decreases as the stagnation pressure increases as shown in Figure 5. Therefore,  $\dot{Q}_T/\dot{Q}_{\text{slow}}$  obtained by temperature difference,  $T_{T, \text{out}} - T_{\text{stg}}$ , remains almost constant as the stagnation pressure increases.



**Fig. 5.** Total temperature, total enthalpy,  $Ma_{\text{in}}$  and mass flow rate as function of  $p_{\text{stg}}$



**Fig. 6.** Heat transfer rates as function of  $p_{stg}$

However, the value of  $\dot{Q}_h/\dot{Q}_{slow}$  increases as the stagnation pressure increases since  $\dot{Q}_h$  was obtained by the enthalpy difference determined from a pressure and a temperature,  $h_{T,out} - h_{stg}$ . Therefore, the value of  $\dot{Q}_h/\dot{Q}_{slow}$  is higher than that of  $\dot{Q}_T/\dot{Q}_{slow}$ . And the difference between  $\dot{Q}_h$  and  $\dot{Q}_T$  is large with increasing the stagnation pressure because of the Joule-Thomson effect for a real gas. As a result of that, in the case of presence in large gas expansion between the inlet and outlet a heat transfer rate should be obtained from the enthalpy differences determined from a pressure and a temperature.

#### 4. Conclusions

In this study, the experimental determination of the convective heat transfer rate of gas flow through a microchannel with constant wall temperature were described. The following conclusions were obtained.

- i. For convective heat transfer of gas flow through microchannel with constant wall temperature, the total temperature is higher than the wall temperature. This is due to an additional heat transfer from the wall to gas since gas static temperature decreases because of conversion of thermal energy into kinetic energy.
- ii. The value of  $T_{T,out} - T_{stg}$  increases with the increase of the stagnation pressure and slightly decreases in the range where the flow is under-expanded (choked). However, the value of  $h_{T,out} - h_{stg}$  increases with the increase of the stagnation pressure and maintains a specific value.
- iii. Both  $\dot{Q}_h$  and  $\dot{Q}_T$  are higher than  $\dot{Q}_{slow}$  obtained from the bulk temperature of incompressible flow because of the additional heat transfer. The difference between  $\dot{Q}_h$  and  $\dot{Q}_T$  increases with the increase of the stagnation pressure because of the Joule-Thomson effect of a gas.

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