Hydraulic and thermal design of a gas microchannel heat exchanger

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Abstract. In this paper investigations on the design of a gas flow microchannel heat exchanger are described in terms of hydrodynamic and thermal aspects. The optimal choice for thermal conductivity of the solid material is discussed by analysis of its influences on the thermal performance of a micro heat exchanger. Two numerical models are built by means of a commercial CFD code (Fluent). The simulation results provide the distribution of mass flow rate, inlet pressure and pressure loss, outlet pressure and pressure loss, subjected to various feeding pressure values. Based on the thermal and hydrodynamic analysis, a micro heat exchanger made of polymer (PEEK) is designed and manufactured for flow and heat transfer measurements in air flows. Sensors are integrated into the micro heat exchanger in order to measure the local pressure and temperature in an accurate way. Finally, combined with numerical simulation, an operating range is suggested for the present micro heat exchanger in order to guarantee uniform flow distribution and best thermal and hydraulic performances.

1. Introduction

With the fast development of micro fabrication technologies in the recent years, many conventional devices in which heat transfer and chemical reactions take place, can be greatly miniaturized with precise control of their dimension [1]. Among these micro-structured devices, the micro heat exchangers are of increasing interest, for their improved heat exchange capacities due to very large surface-to-volume ratio [2].

The applied research on micro channel heat exchangers using liquid (mostly deionized water) as working media can be considered extensive, even though some of the results are not in agreement. Based on experiments with cross-flow micro heat exchangers, Harris et al. [3] found that the volumetric heat transfer in these micro-sized devices is improved at least five times compared with their conventional-sized counterparts. Improved heat exchanger devices of different microstructure designs and manufactured in metal have been described by Brandner et al. [4, 5]. Alm et al. [6] reported larger heat transfer coefficients and larger pressure losses in ceramic microchannel heat exchangers available for both counter flows and cross flows. Cao et al. [7] proposed experimental correlation for the prediction of the average value of Nusselt number and pressure drop of microchannel heat exchangers. On the contrary, the investigation of Garcia-Hernando [8] showed no heat transfer enhancement or higher pressure drop through micro heat exchangers due to the small scale of the microchannels. More recently, Mathew and Hegab [9] provided a universal correlation to estimate the heat transfer performance of microchannel heat exchangers subjected to external heat flux, regardless of the cross-sectional geometry of the microchannels.

Compared with research on liquid flow micro heat exchangers, experimental investigations on these devices objected to gas flows is scarce. The previous works available in literature for gas micro heat exchangers are merely reported by Bier at al. [10], Meschke et al. [11] and Koyama and Asako [12]. The detailed parameters of the gas micro heat exchangers analyzed in their works and their test conditions are listed in Table 1.
The first work on gas micro heat exchangers made of metal was reported by Bier et al. [10] for cross flows of helium, nitrogen and argon. The researchers found that in laminar regime the overall heat transfer coefficients to be greatly reduced for low mass flow rates. In addition they proposed a homogeneous model to account for a dominant influence of longitudinal heat conduction in the solid wall. As the thermal conduction in metal (copper and stainless steel) overwhelms that of the fluid, actually in such a case the micro heat exchanger works as a temperature mixer for cold and hot flows. This is probably the reason why under this model the thermal performance of heat exchangers does not differ between cross flow, co-current flow and counter-current flow, which is quite different from the typical features of conventional-sized heat exchangers. The heat transfer rate in this work was reported highly dependent on the arrangement of flow. Accordingly it was suggested by the authors that, in order to obtain larger heat transfer rates, material of lower thermal conductivity should be used both for microchannel plates and for the shell of the micro heat exchanger.

The investigation by Meschke et al. [11] on a ceramic micro heat exchangers for gases also stressed lower values of heat transfer coefficients than the theoretical predictions based on the conventional theory. According to the authors, the reduced heat transfer coefficients were subjected to the strong axial heat conduction in the solid wall. In addition, the heat loss due to conduction from the ceramic core to the steel housing was found to become significant on the thermal performance of the micro heat exchanger. The influence of thermal conduction in the solid on the overall thermal performance was that large that the impact of flow arrangement (cross, co-current, counter-current) becomes comparably insignificant.

Similar conclusion was drawn by Koyama and Asako [12] from experiments with a stainless steel micro heat exchanger. In this work no significant difference in the total thermal performance was found between co-current and counter-current flows. The investigators finally used an isothermal model with T boundary condition to explain their experimental data.

Summarizing the above experimental research of gas flow micro heat exchangers, it is worthy to note that all the authors did not carefully consider the possible effect of material choice on the thermal performance of the micro heat exchangers in the design phase of these devices.

Another issue raised on the research of micro heat exchangers is the flow distribution among the channels, which is linked to the hydraulic performance of these devices. Most of the micro heat exchangers described in literature are based on parallel microchannels built on a plate; different arrangements of the plates in the stacking process enable different geometries for the flow configuration of the micro heat exchangers. One of the common features of such plates is that the flows through all the parallel microchannels share the same feeding source; the source flow requires a distribution chamber in order to spontaneously feed all the microchannels as uniformly as possible. In the work of Bier et al. [10] and Meschke et al. [11] this issue of flow distribution was not addressed. Koyama and Asako [12] found the experimentally measured pressure drop to be several times higher than prediction, and they regarded non-uniform distribution of mass flow as one reason to this disagreement. However, the exact impact of flow maldistribution on additional pressure drop was not estimated in the work of Koyama and Asako [12].

The flow distribution is very important for micro heat exchangers, because it does not only influence the hydraulic behaviour of the micro heat exchanger, but also changes the thermal behaviour of the device. The optimal design for uniform distribution of flow was studied by Commenge et al. [13] who proposed an approximate pressure drop model. Chein and Chen [14] compared the velocity and temperature distribution along 11 parallel microchannels by changing the position of distribution/collecting ports. Tonomura et al. [15] developed an automatic optimization algorithm which is able to tailor iteratively the shape of contributing/collecting manifold until satisfying mass flow distribution. Balaji and Lakshminarayanan [16] proposed a new design of the distribution chamber in order to obtain uniform flow distribution among the parallel microchannels. In their design the flow enters the distribution chamber from one port while the

| Authors            | Channel width [µm] | Channel height [µm] | Materials     | Conductivity [WK⁻¹m⁻¹] | Partition wall thickness [µm] | Gas species | Flow arrangement |
|--------------------|--------------------|---------------------|---------------|------------------------|-----------------------------|-------------|------------------|
| Bier et al. [10]   | 100                | 78                  | copper        | 401                    | 22                          | He, N₂, Ar  | cross flow       |
| Bier et al. [10]   | 90                 | 75                  | stainless steel| 14.7                   | 25                          | He, N₂, Ar  | cross flow       |
| Meschke et al. [11]| 1500               | 1500                | ceramic SiC    | 125                    | 2000                        | N₂          | cross flow       |
| Koyama and Asako   | 300                | 200                 | stainless steel| 15                     | 300                         | air         | counter-current  |

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exhausted flow is collected by two ports. An analytical model for optimization of flow distribution in a manifold has been reported by Renault et al. [17]. The model is able to calculate the flow distribution faster than CFD simulation without losing too much accuracy.

The methods described in the works available in the open literature provide convenient instruction for the design of micro heat exchangers. However, up to now it is difficult to find a systematic way to optimize the flow distribution for multi-plate microchannel heat exchangers, mainly due to the limited possibilities to balance all the parameters in the design. These parameters include dimension of micro channels (width, height and length), shape of distributing/collecting manifolds, curvature of the corners, length of connections between the heat exchanger core and manifolds, position of the feeding/exhaust ports, direction of the inflow/outflow, etc. Furthermore, in practical tests and applications of micro heat exchangers the parameters mentioned above may be subjected to different types of restriction coupled with higher priorities (assembling, working condition, vulnerability to possible damages, pricing etc.). Therefore, experimental investigators should carefully look for compromises in the design phase, and it is recommended that the influence, such as solid wall conduction and flow distribution, are accurately estimated or even simulated with CFD methods in order to interpret experimental data under specific conditions. This applies more truly to micro heat exchangers using gas as working fluid, because up to now the experimental research in this field is weak if compared with results of numerical works and/or with the experimental works on liquid micro heat exchangers.

In this study the choice of the best material for a gas micro heat exchanger was quantitatively evaluated in order to optimize its dynamic and thermal performances. With an appropriate design some scaling effect, like the axial thermal conduction along the solid walls of the micro heat exchanger, can be greatly suppressed, while the overall heat transfer efficiency can be improved. The uniformity of mass flow and pressure distribution was obtained for different flow conditions by means of a commercial CFD code. The local pressure losses at both distribution and collecting chambers are numerically evaluated for different configurations of the manifolds.

2. Experiment

Figure 1 shows schematics of the stacked core for a double-layered microchannel heat exchanger and a multi-layered one. The arrangement of the plates enables co-current and counter-current gas flows within large ranges of pressure and mass flow rate. The cores are stacked in such a way that the layer of hot flow and the layer of cold flow are alternated. The core is housed in a shell made of polymer. The low heat conductivity of polymer (compared with metal) ensures low heat losses. In addition, for the double-layered micro heat exchanger the thin plate between the hot and cold flow can be exchanged. The material and thickness of the plate can be chosen more or less freely based on various purposes.

![Figure 1](image-url)

**Figure 1.** Schematic of the micro heat exchanger core. (a) double-layered; (b) multi-layered.

The detailed parameters of the micro heat exchanger are listed in Table 2. The microchannels have square cross section with a length of 39.8 mm. The bottom thickness of the thin layer is 100 µm. On each layer 133 parallel microchannels are manufactured. These parallel microchannels are separated by side walls with a thickness of 100 µm. The micro heat exchanger has a surface-to-volume ratio of 20000, larger than the
typical values of the conventional compact heat exchangers (<1000). The entire core is made of PEEK with a thermal conductivity of 0.25 WK\(^{-1}\)m\(^{-1}\). The reason for such a choice will be detailed in the discussion section.

**Table 2. Parameters of the micro heat exchanger.**

| Micro heat exchanger | Value of parameters |
|----------------------|---------------------|
| Channel width \(W\): [µm] | 200 |
| Channel height \(H\): [µm] | 200 |
| Hydraulic diameter \(d_h\): [mm] | 200 |
| Channel length \(L\): [mm] | 39.8 |
| Surface roughness of channel \(\varepsilon\): [µm] | 0.15 |
| Thickness of channel side wall \(s_1\): [µm] | 100 |
| Thickness of partition wall \(s_2\): [µm] | 100 |
| Material of channel wall | PEEK |
| Thermal conductivity of channel wall \(\lambda_w\): [WK\(^{-1}\)m\(^{-1}\)] | 0.25 |
| Material of shell | PEEK |
| Thermal conductivity of shell: [WK\(^{-1}\)m\(^{-1}\)] | 0.25 |
| Number of channels per layer | 133 |
| Number of layers for hot/cold flow | 1 and 10 |
| Ratio of (heat transfer) surface to (flow) volume: [m\(^2\)/m\(^3\)] | 20000 |

A typical layer with 133 parallel microchannels, distribution/collecting chambers and feeding/exhaust ports is shown in Figure 2(a). The layer has a width of 39.8 mm, which is identical to the length of the microchannels. For cross flow the alternative layers can be rotated by 90 degree, while the hot layer and cold layer of the core are still in perfect thermal contact. As the micro heat exchangers are designed to work in large pressure range (up to 40 bar), 22 pillars with a diameter of 0.8 mm are added to the distribution/collecting chamber in order to maintain mechanical stability and avoid possible deformation under high pressure. With the presence of these supporting pillars, different methods on flow distribution in literature are not able to predict the flow distribution. Actually such supporting pillars are quite common in the application of microdevices (such as micro evaporators [18]), if the dimension and thickness of microstructure are greatly reduced while the mechanical stability has to be maintained.

The micro heat exchanger was manufactured using mechanical precision machining [19, 20]. CNC-controlled tooling systems and miniaturized tools made from natural diamond have been used to create the microchannels into polished PEEK platelets. The precise control of the design allows manufacturing of microchannels with an absolute dimension uncertainty of about +/- 1µm in all three dimensions (namely: width, depth and length of the microchannels). The average surface roughness \(\varepsilon\) inside the microchannels was not measured but estimated to be in the range of 150 nm. This was based on experiences with the manufacturing of microstructure devices made of PEEK as well as on measurements undertaken by Hecht et al. [21] and Vittoriosi et al. [22]. The ratio between the average roughness and the hydraulic diameter is about 7*10\(^{-4}\). Due to this low value the estimated relative roughness is most likely small enough to be neglected in the evaluation of frictional losses and convective heat transfer coefficients along the microchannels.

Figure 3(b) shows a magnified area of the distribution chamber. Two K-type temperature sensors and one pressure sensor are delicately planted in the top surface of the chamber. In the previous experimental work on gas micro heat exchangers, fluid temperature and pressure were measured before the feeding ports, or after the exhaust ports [12]. This underestimates the local pressure loss and temperature loss by far. The large disagreements between experimental data and theory were attributed to the pressure or temperature losses [12]. To solve those measurement problems, in the present study pressure and temperature are measured at a position very close to the ends of microchannels.
Figure 2. Typical layer (with 133 microchannels) of the micro heat-exchanger (a); in-situ measurement in distribution/collecting chambers (b) (1- pressure sensor; 2- temperature sensor; 3- pillars. unit: mm).

3. Numerical model
The existing models in literature on uniform flow distribution cannot be applied to the present pillared chambers, not only due to the existence of pillars, but due to the different location and direction of inflow and outflow voids. Therefore, a CFD model was created using the finite volume solver of ANSYS (FLUENT 6.3). The model is able to calculate the exact mass flow and pressure distribution of the present design. The dependence of mass flow and pressure distribution on absolute feeding pressure was examined using this model. In addition, the local pressure drop at both distribution and collecting chambers was computed.

The CFD model adopted the following assumptions:
1. Two-dimensional model;
2. The flow is incompressible;
3. The walls have smooth surfaces;
4. No heat transfer is involved;

The influence of the grid dependence on the result was carefully examined. Three sets of meshes with different coarseness were built, as listed in Table 3. To reduce the computation time, only the distribution chamber with a very short length of the microchannels was meshed, as suggested by Renault et al. [17]. The grid dependence was checked for four different pressure drops under which the mass flow rate was monitored. The finest mesh was taken as a reference. From Table 3 it can be noted that with the fine mesh the relative difference in the mass flow rate is less than 1% for all the pressure drops in computation. Therefore, the model was built with a fine mesh.

Two different CFD models were built. The first one is only for the distribution chamber with a short length of microchannels; the second one is a full simulation of the entire layer, including the full length of the 133 parallel microchannels and the collecting chamber. The numerical results from these two models have been compared.
4. Results and discussion

4.1. Heat conduction in the solid wall

As stated in the introduction section, all the available experimental works on gas flow micro heat exchangers concluded that conventional theory on heat exchangers failed to explain the experimental data due to heat conduction in the solid wall. Instead, an isothermal assumption was used in order to better explain the experimental trends. Even the flow arrangements (cross flow, co-current flow and counter-current flow) make no difference on the convective heat transfer coefficients by using highly conductive material (copper, stainless steel and ceramic), as evidenced by the work of Bier et al. [10], Meschke et al. [11] and Koyama and Asako [12]. However, in the previous works no statement was made on how the thermal conductivity of wall affects the performance of the gas micro heat exchangers.

To estimate the axial heat conduction in the wall, Maranzana et al. [23] proposed a wall axial conduction parameter, the ratio of conductive heat flux in the solid wall to the convective heat flux in the microchannel. For a circular tube with inner diameter $d$ and external diameter $D$, the wall axial conduction parameter can be expressed as:

$$ M = \frac{\phi_w}{\phi_f} = \frac{\lambda_w}{\lambda_f} \frac{D^2 - d^2}{dL} \frac{1}{RePr} $$

(1)

where $\phi_w$ is the heat flux due to thermal conduction in the wall, $\phi_f$ the convective heat flux, $\lambda_w$ the thermal conductivity of the solid wall, $\lambda_f$ the fluid thermal conductivity, $L$ the length of microchannel, $Re$ the Reynolds number and $Pr$ the Prandtl Number of the fluid, respectively.

Generally if $M<0.01$, which means the conductive heat flux is less than 1% of the convective heat flux, the influence of axial heat conduction in the wall can be neglected. When this parameter is larger than 0.01, the conductive heat flux becomes significant in the overall heat transfer. The influence of axial wall heat conduction on heat transfer was investigated in the work of Yang et al. [24] for a single microtube with thick walls.

For multi-layered microchannel heat exchangers, the ratio of wall conductive heat flux to convective heat flux can be calculated by

$$ M = \frac{\phi_w}{\phi_f} = \frac{\lambda_w}{\lambda_f} \frac{(W + s_1/2)(H + s_2/2)}{WH} \frac{2WH}{(W + H)L} \frac{1}{RePr} $$

(2)

where $W$ is the width of the microchannel, $H$ the microchannel height, $s_1$ the thickness of sidewall between parallel microchannels and $s_2$ the thickness of the partition wall separating the cold flow and the adjacent hot flow.

In order to have an insight of the wall thermal conduction influence, the data points measured by Bier at al. [10], Meschke et al. [11] and Koyama and Asako [12] for gas flow micro heat exchangers were depicted in Figure 3, showing the axial conduction parameter as a function of Reynolds number. The value of the axial conduction parameter was calculated based on the exact dimension of the microchannels, thicknesses of the walls, thermal conductivity of the materials, thermal properties of the fluids as well as specific flow conditions from each author. The dashed line in Figure 3 indicates the threshold indicated by Maranzana et al. [23] for the axial conduction parameter. From Figure 3 it is surprising to note that almost all the data points by these investigators fall in the region where axial heat conduction in the wall is of significance. In some cases the axial conductive heat flux in the wall is comparable to or even larger than the convective heat flux. Thus, the temperature profile tends to flatten out throughout the micro heat exchangers and the heat...
transfer was deteriorated. This is probably the reason why these researchers found an isothermal model or a
T boundary condition (uniform wall temperature) more appropriate to explain the experimental trends. If the
tests carried out by Bier et al. for micro heat exchangers made of different materials are compared, it can be
found that for the same gas species (e.g. helium) the axial heat conduction in copper micro heat exchangers
is approximately one order of magnitude higher than that in stainless steel micro heat exchangers. Actually
this is easily reflected in Eq. (2). For the same material (stainless steel), the axial conduction parameter is
larger in the test of Koyama and Asako [12] than that in the investigation by Bier at al. [10]. This is because
of the much thicker wall used in the micro heat exchanger of the former. By observing these data points, it
can be found that the axial conduction parameter has a decreasing trend with the increase of Reynolds
number. Therefore, for micro heat exchangers made of more conductive material, larger Reynolds numbers
or higher mass flow rates are suggested in order to avoid large wall axial heat conduction.

![Figure 3. Wall conduction parameter in the previous and present work (S.S.-stainless steel).](image)

Polymer (PEEK) was selected for the micro heat exchanger in the present work. With the information of
microchannel size and thermal properties listed in Table 2, the magnitude of the axial conduction parameter
is shown in Figure 4 for the Reynolds number range proposed for the experiments of the current study. In the
design of the present work, the significance of axial heat conduction in the wall becomes very slight
compared with convective heat transfer. Therefore the present design should be able to use the thermal
analysis model proposed for conventional heat exchangers. Also it is expected that with the present design
the dependence of the thermal performance on flow arrangements (co-current, counter-current and cross
flow) can be distinguished. Experimental data from these heat exchangers will be reported as soon as
possible in another paper.

Although all the experimental research on gas flow micro heat exchangers reported strong axial
conduction in the wall, similar problem seems not encountered in the study ([3, 9]) on liquid flow through
these devices. In order to know the possible influence of fluid types on the wall axial conduction, the data
points by Bier at al. [10], Meschke et al. [11] and Koyama and Asako [12] are again depicted in Figure 4.
The difference to Figure 4 is that the working gases are changed into water. In Figure 4 the data series (a)
corresponds to the Reynolds number in the work of Bier at al. [10]; while the data series (b) refers to the
Reynolds number from Koyama and Asako [12]. It can be seen from Figure 4 that most of the data points fall
below the threshold when the working fluid is changed from gas to liquid, except for the copper micro heat
exchanger working under very low Reynolds numbers. Therefore, in the research of liquid micro heat
exchangers the influence of axial conduction in the wall is widely reduced. This can directly be noted from
the ratio of axial conduction parameter for gas (i.e. air) to that for liquid (i.e. water), as shown in Eq. (3).

\[
\frac{M_{air}}{M_{water}} = \frac{\dot{\lambda}_{water} Pr_{water}}{\dot{\lambda}_{air} Pr_{air}}
\]  

(3)

Supposing the same micro heat exchanger working at the same Reynolds number, the ratio in Eq. (3)
generates a value of about 220. This draws great attention to axial conduction in the wall when the working
fluid is changed from liquid to gas.
Figure 4. Wall conduction parameter by changing the working gases in previous work into water.

The overall efficiency of heat exchangers, which can be calculated by Eq. (4) ([25]), may also be influenced by the wall thermal conductivity.

\[ \varepsilon = \frac{q c_{p,c} \left( T_{c,\text{out}} - T_{c,\text{in}} \right)}{q_{c_{p,h}} \left( T_{h,\text{in}} - T_{h,\text{out}} \right)} = \frac{q_{c_{p,h}} \left( T_{h,\text{in}} - T_{h,\text{out}} \right)}{q_{c_{p,h}} \left( T_{h,\text{in}} - T_{c,\text{in}} \right)} \] (4)

where \( q \) is the gas mass flow rate on the hot (\( h \)) and cold (\( c \)) side of the heat exchanger, \( c_{p} \) is the fluid specific heat and \( T \) is the temperature at the inlet (\( \text{in} \)) and at the outlet (\( \text{out} \)) of the device.

The optimal thermal conductivity of material for micro heat exchangers was numerically studied by Stief et al. [26] in 1999. Figure 5 shows their results for the overall efficiency of a counter-current heat exchanger as a function of the solid thermal conductivity. If the thermal conductivity approaches 0, the solid wall functions as thermal isolation material between hot and cold flows. Therefore the heat transfer is blocked by the wall, and the heat exchanger efficiency is close to 0. If the wall is made from a very good heat conductive material, the strong axial conduction in the wall reduces the heat exchanger efficiency as well. Due to this reason thermal conductivity in the order of 1 W/mK offers best efficiency for a gas-gas heat exchanger. For PEEK with a thermal conductivity of 0.25 W/mK, the micro heat exchanger presented in this work is expected to have relatively high overall efficiency (see Figure 5).

Figure 5. Heat exchanger efficiency in dependence on the wall thermal conductivity (from Stief et al. [26]).

4.2. Flow distribution in the distribution chamber
Numerical simulation may provide useful information on local distribution of flow in the design phase of micro heat exchangers. The main concern on numerical means is that it requires a lot of efforts on the
geometry modeling and computation time to simulate a full layer, which is composed of several hundreds of microchannels, distribution chamber as well as collecting chamber. A common treatment for simplification is to simulate how the flow distributes from the feeding port to the inlets of microchannels based on some assumed pressure difference (local pressure loss). This provides a quick solution to know the general flow distribution, and sometimes used by engineers and technicians to modify the designs of micro devices. However, this method is not generally verified in order to know whether it is able or not to predict the real flow distribution, if the flow conditions far downstream the distribution chamber are not considered. In this work, two numerical models are built: the first, in which a single distribution chamber is modeled and the second in which a full layer is taken into account. The numerical results of these two models will be compared, in order to check the real need of an extensive computation of the full-layer model.

For the numerical model of a single distribution chamber, the feeding pressure was fixed at four different absolute values, namely 110 kPa, 160 kPa, 250 kPa and 500 kPa. The absolute value of the exhaust pressure is set at 100 kPa, which is atmospheric pressure. The mass flow rates through every microchannel are the unknown variables to be calculated. The outlet pressure of the single chamber is assumed to be 99% of the feeding pressure, which results in an assumption of 1% pressure drop.

Figure 6 shows the air flow velocity distribution in the distribution chamber under feeding pressure of 160 kPa. The disturbance of the pillar array is distinct in the velocity field. However, the wakes of the disturbance downstream the pillar fade out in a short flow path, and the velocity redistributes immediately before the flow enters the parallel microchannels. In addition, there is a region with relatively high flow velocity in the distributing chamber where there is absence of pillars in the flow path.

![Figure 6. Flow velocity field in the distributing chamber at feeding pressure of 160 kPa.](image)

To have an insight into the mass flow distribution through each microchannel, the mass flow rate through every microchannel under different feeding pressure values was calculated. The result is shown in Figure 7 for a dimensionless mass flow rate.

\[
q_{ave} = \frac{1}{n} \sum_{i=1}^{n} q_i, \quad n = 133
\]

where \(q_i\) is the mass flow rate of microchannel \(i\).

Figure 7 indicates quite ununiform mass flow distribution for all the parallel microchannels. For microchannels indexed from 115 to 132, the mass flow rate is more than twice as high as the average one. In addition, it is worthy to note that the distribution of the mass flow rate has very slight dependence on the feeding pressure. It is yet unknown whether the trend displayed here is representative for a full layer. Therefore, another CFD model with a full layer was built in order to have a more detailed insight of the real flow situation.
4.3. Flow distribution and pressure drop of a full layer

The numerical results from the simulation of a full layer (shown in Figure 2) are discussed in this section for feeding pressure set at 110 kPa, 160 kPa, 250 kPa and 500 kPa. The flow is vented from the exhaust port at atmospheric pressure, namely 100 kPa. With this model the mass flow rate and pressure drop through all the microchannels will be self-balanced.

The velocity field in the distribution chamber and the microchannels is given in Figure 8 for a feeding pressure of 160 kPa. As the velocity profile in the microchannels does not change after the hydraulic entry length, only a part of the microchannel length is shown in Figure 8. The same will be applied thereafter for the downstream part of the layer. From Figure 8 it can be noted that the velocity in the distribution chamber is generally higher than that in the microchannels. This is because the total flow area normal to the streamline in the distribution chamber is smaller than the total flow area normal to the streamline of all the parallel microchannels. This is a common feature of most microchannel devices using distribution connections between macro piping lines and microchannels. The velocity contour of the downstream part of the layer is shown in Figure 9. By observing the velocity distribution, it can be observed that the flow velocity through the microchannels closest to the feeding port (or with short flow path) is unexpectedly the lowest. This is related to the pressure distribution throughout the whole layer.

Figure 7 Mass flow rate of each microchannel under different feeding pressure.

Figure 8. Flow velocity field in the distribution chamber and microchannels at feeding pressure 160 kPa (length of microchannel partly displayed, velocity>100 m/s in white region).
The pressure contour for the upstream part of the layer is shown in Figure 10, for the downstream part in Figure 11. The pressure at the inlets of the microchannels close to the feeding port is very high. This is caused by the high pressure at the outlets of these microchannels, as it is shown in Figure 11. The pressure drop from the microchannel outlets to the exhaust port is larger for microchannels further away from the exhaust port. This affects the pressure development inside the microchannels all the way from outlet to inlet. Accordingly the pressure distribution at the inlet region of the microchannels will be influenced. This is quite different from the simulation of a single distributing chamber.

Figure 12 shows the distribution of the mass flow rate through all the parallel microchannels for different feeding pressure values. The microchannels furthest from the feeding port show the highest mass flow rate.
The mass flow rate decreases in the microchannels approaching the feeding port, with small fluctuations in between due to disturbances created by the pillars. Actually from Figure 12 it can be seen that the existence of the pillars seems to even the mass flow distribution by creating an average flow velocity. In the region far from the pillars, the mass flow rate gradient is steeper. In addition, by increasing the feeding pressure, the uniformity of the mass flow rate distribution becomes worse.

**Figure 12.** Mass flow rate of each microchannel under different feeding pressure.

By comparison between Figure 12 and Figure 7, the simulation of the full layer generates exhibits a mass flow distribution completely different from the simulation of a single distribution chamber. Therefore, without full simulation the numerical results might be misleading.

The pressure at the inlet and outlet of each microchannel is compared for all the parallel microchannels in a dimensionless way. Similarly to the average mass flow rate, the mean pressure averaged over all the 133 microchannels is calculated by

\[
p_{\text{ave}} = \frac{1}{n} \sum_{i=1}^{n} p_i, \quad n = 133
\]

where \(p_i\) is the pressure in microchannel \(i\).

In Figure 14 the pressure distribution at the inlet of the parallel microchannels is depicted. Higher feeding pressure generates a more uniform distribution of the inlet pressure. For the microchannels in the middle area of the layer, the inlet pressure is quite close to the average one. This provides some hints for the measurement of the average pressure at inlet. According to these simulation result, it seems to be better to place the pressure sensor in the middle part of the total width of all the microchannels.

**Figure 13.** Pressure distribution at inlet of each microchannel.
Besides pressure distribution, the pressure loss at the distribution chamber is calculated for all the microchannels as a subject of different feeding pressure values, as shown in Figure 14. Generally, with higher pressure at the feeding port a higher fraction of pressure is lost due to the flow distribution and the supporting pillars. Therefore, if the pressure measured at the feeding port or piping line (which is the easiest way to measure) is considered as the inlet pressure of the microchannels, a large uncertainty may be faced especially at high pressure values or larger mass flow rates.

For a fixed feeding pressure, a negligible pressure loss was found for microchannels very close to the feeding port, indexed from around 110 to 133 for the present micro heat exchanger. For these microchannels, the flow path from the feeding port is very short and free of obstructing pillars. For several microchannels furthest away from the feeding port, the pressure loss is also relatively small due to the fact that there are no obstacles in form of pillars within the flow path.

![Figure 14](image-url)

**Figure 14.** Pressure drop in the distribution chamber (from feeding port to inlet of microchannel).

The pressure distribution at the outlets of the parallel microchannels is shown in Figure 16, calculated for different feeding pressure values. Compared with that at the microchannel inlets, the distribution at the outlets is relatively linear. The further the microchannel is away from the exhaust port, the larger the outlet pressure becomes. Similarly, higher feeding pressure values increases the non-uniformity of outlet pressure distribution. The outlet pressure of the microchannel located at the middle of the whole panel is most representative of the average pressure.

![Figure 15](image-url)

**Figure 15.** Pressure distribution at outlet of each microchannel.
Figure 16 shows the pressure loss from microchannel outlet to the exhaust port. In the pillar-free region with the shortest flow path, the pressure loss is very small. An exception is the feeding pressure of 500 kPa. At feeding pressure higher than 250 kPa, the microchannel outlet pressure can go above 1.5 times of atmospheric pressure. Thus, simply regarding the outlet pressure to be ambient might lead to large uncertainties.

![Figure 16](image.png)

Figure 16. Percentage of pressure drop in the collecting chamber (from outlet to exhaust port).

The statistical data of the numerical simulation on the whole layer are listed in Table 4, in terms of mass flow distribution, microchannel inlet pressure distribution and outlet pressure distribution. The number of microchannels showing a mass flow rate within a band of ±20% of the average value is counted. The same has been performed for the pressure at inlet and outlet. It can be noted that at low feeding pressure the distribution of these parameters is very uniform. Uniformity is decreased with increasing inlet pressure. For a feeding pressure of 500 kPa and above, more than half of all the microchannels show magnitudes for these parameters falling outside the range. Therefore, for the present micro heat exchanger, it is suggest that the feeding pressure should be kept lower than 250 kPa in order to maintain uniform flow distribution.

### Table 4. Uniformity of mass flow rate and pressure distribution in terms of microchannel number.

| $p_{in}$ [kPa] | $p_{out}$ [kPa] | Mass flow rate with $q_i < (1\pm 20\%) q_{ave}$ | Num. of channels with $p_{in,i} < (1\pm 20\%) p_{in,ave}$ | Num. of channels with $p_{out,i} < (1\pm 20\%) p_{out,ave}$ |
|---------------|----------------|---------------------------------------------|---------------------------------|---------------------------------|
| 110           | 100            | 102/133                                     | 133/133                         | 133/133                         |
| 160           | 100            | 87/133                                      | 133/133                         | 115/133                         |
| 250           | 100            | 83/133                                      | 81/133                          | 43/133                          |
| 500           | 100            | 79/133                                      | 40/133                          | 23/133                          |

5. Conclusions

In this paper the design of a gas flow microchannel heat exchanger was studied from both hydrodynamic and thermal aspects. The main conclusions are drawn as follows:

- For microchannel heat exchangers used with gas flows, the thermal conductivity of the solid material is extremely important. With highly conductive material, the overall heat transfer efficiency is very low and the dependency of thermal performance on flow arrangements is lost.
- A gas flow micro heat exchanger made of polymer is designed and manufactured. Local measurement of pressure and temperature is achieved by integrating sensors very close to the microchannels.
- Simulation of a full layer shows quite different distribution of mass flow and pressure compared with that of a single chamber. This means a mere simulation of the distribution chamber is not sufficient
to gain real insights into the flow distribution, and thus may be misleading in the design of micro heat exchangers.

- The full numerical simulation uncovers the distribution of mass flow rate, inlet and outlet pressure in dependence of the feeding pressure. Based on the results, the present micro heat exchanger is found to have relatively uniform flow distribution if working at a feeding pressure of 250 kPa or lower.
- The pressure drop (losses) in the pillared distribution/collection chamber is obtained for different feeding pressure. The distribution of pressure loss in terms of microchannel index is investigated for both distribution chamber and collecting chamber; this analysis gives useful hints for the optimal hydraulic design of a gas micro heat exchanger.

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7. Nomenclature

| Symbol | Definition                        | Unit          |
|--------|-----------------------------------|---------------|
| $c_p$  | Gas specific heat                | (J/kgK)       |
| $d$    | Inner diameter                    | (m)           |
| $D$    | External diameter                 | (m)           |
| $d_h$  | Hydraulic diameter                | (m)           |
| $H$    | Height of microchannel            | (µm)          |
| $L$    | Length of microchannel            | (µm)          |
| $M$    | Axial heat conduction parameter   | (-)           |
| $p$    | Pressure                           | (Pa)          |
| $Pr$   | Prandtl number                     | (-)           |
| $q$    | Mass flow rate                     | (kg/s)        |
| $Re$   | Reynolds number                    | (-)           |
| $s_1$  | Thickness of channel sidewall     | (µm)          |
| $s_2$  | Thickness of partition wall       | (µm)          |
| $T$    | Temperature                        | (K)           |
| $W$    | Width of microchannel             | (µm)          |

**Greek symbols**

- $\varepsilon$ Absolute roughness (m)
- $\phi$ Heat flux (W/m²)
- $\lambda$ Thermal conductivity (W/mK)
- $\mu$ Dynamic viscosity (kg/ms)

**Subscript**

- c cold flow
- f fluid
- h hot flow
- in inlet value
- out outlet value
- w wall

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