



Fluid Dynamics and Transport Phenomena

Computational analysis of the flow of pseudoplastic power-law fluids in a microchannel plate



Saeed Mortazavi

Department of Chemical Engineering, Shahrood Branch, Islamic Azad University, Shahrood, Iran

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ABSTRACT

The flow of pseudoplastic power-law fluids with different flow indexes at a microchannel plate was studied using computational fluid dynamic simulation. The velocity distribution along the microchannel plate and especially in the microchannel slits, flow pattern along the outlet arc and the pressure drop through the whole of microchannel plate were investigated at different power-law flow indexes. The results showed that the velocity profile in the microchannel slits for low flow index fluids was similar to the plug flow and had uniform pattern. Also the power-law fluids with lower flow indexes had lower stagnation zones near the outlet of the microchannel plate. The pressure drop through the microchannel plate showed huge differences between the fluids. The most interesting result was that the pressure drops for power-law fluids were very smaller than that of Newtonian fluids. In addition, the heat transfer of the fluids through the microchannel with different channel numbers in a wide range of Reynolds number was investigated. For power-law fluid with flow index ($n = 0.4$), the Nusselt number increases continuously as the number of channels increases. The results highlight the potential use of using pseudoplastic fluids in the microheat exchangers which can lower the pressure drop and increase the heat transfer efficiency.

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1. Introduction

Rapid growth of microsystems technologies in recent years leads to use of these systems for various applications. These applications vary from chemical to biological fields such as microheat exchangers [1,2], electrodes [3], microelectromechanical systems [4], fuel cells, microreactors [5,6], molecular biology [7–9], DNA chips [10], selective extraction [11], multiphase emulsions preparation [12], hollow sphere production [13] and many other technologies. High surface-volume ratio and small scale dimensions lead to microdevices advantage over conventional macrosystems. For optimum design of microsystems, it is necessary to recognize the transport phenomena in the microscale. Unfortunately, the small dimensions of these devices cause some problems when high flow rates are needed. At large input the flow velocity is very high which results to huge pressure drop and non-uniformity in flow distribution and reduces the microchannels performance. Therefore, many numerical and experimental studies were performed to investigate the flow behavior and heat transfer in the microchannels.

Some experiments were performed to study the velocity distribution through the microchannels. Adeosun *et al.* [14] characterized experimentally and numerically the mixing performance of a T-junction microchannel. Residence time distribution (RTD) was used to indirectly assess the mixing behavior in the microchannel. The results showed good agreement between normalized RTD results (obtained via

convection–diffusion model) and normalized experimental RTDs. Jones *et al.* [15] studied the flow distribution in a 76-channel heat sink microchannel using an infrared microparticle image velocimetry (IR μ PIV) technique. It was observed that the misdistribution for low Reynolds number was very low.

A few studies investigated the pressure drops along the microchannels. Zivkovic *et al.* [16] developed a correlation equation for pressure drop along a rectangular orifice within a similar shaped channel under creeping flow situations. Their correlation implies that the pressure drop is appropriate to the average velocity through the orifice. However, this approximation was valid only for orifice aspect ratio greater than 30.

There are some works on the power-law fluids flow in the channels. Srinivas [17] investigated a 2D electro-osmotic flow of a power-law fluid within circular and elliptic microchannels. Their results showed that the elliptic microchannels decrease the flow rate as compared to the circular ones. Zhu *et al.* [18] studied the periodical flow of the power-law fluids through a rectangular microchannel while electro-viscous affected the flow pattern. It was concluded that the velocity field depends mostly on the flow index parameter (n). Stocks *et al.* [19] optimized heat transfer ability of non-Newtonian fluid flow in a simple and complex microchannel. A shear thickening non-Newtonian fluid (dilatants) with flow index of $n = 1.5$ greatly reduced the thermal conductivity and thus heat transfer efficiency. Bijjam *et al.* [20] studied a 2-D flow of power-law fluids over a circular cylinder confined in a channel for a varying range of power-law indices and Reynolds numbers. They elucidated that for highly shear thinning fluids ($n \leq 0.6$), the total

E-mail address: s.mortazavi@iau-shahrood.ac.ir.

drag coefficient increases with increasing value of Reynolds number. However, for power-law fluids ($n > 0.6$) the drag coefficient decreases on increasing the Reynolds number. Mohebbi *et al.* [21] studied the laminar power-law fluid flow (with power-law flow indices of $n = 0.7$ and $n = 1.3$) between two parallel plates for different Reynolds numbers. Their results showed that the heat transfer rate increases with an increase in the power-law index. Hayat *et al.* [22] presented the peristaltic motion of power-law fluids in an asymmetric channel with convective conditions at the channel walls. Their thermal studies revealed that increase in the power-law fluid parameter decreases the temperature of fluid. Shojaeian *et al.* [23] performed convective heat transfer analysis to reveal the effect of some variables such as power-law flow indexes on the heat transfer characteristics of fully developed power-law fluid flows in tubes under uniform heat flux. They concluded that Nusselt number decreases with increasing power-law index. Yigit *et al.* [24] investigated numerically the effects of aspect ratio on the laminar Rayleigh–Bénard convection of the power-law fluids in rectangular enclosures with differentially heated horizontal walls. They found that the mean Nusselt number does not exhibit a monotonic increase with decreasing power-law index for aspect ratios smaller than 2 due to change in the flow pattern. Baranwal and Chhabra [25] studied the effects of the power-law index on the streamlines and Nusselt numbers along the surface of the hot and cold cylinders. They showed that heat transfer is enhanced in shear thinning fluids. Akbar and Butt [26] investigated peristaltic mechanisms in a two dimensional nonuniform channel filled with Herschel–Bulkley fluids. They observed that the flow velocity decreases with an increase in power-law index. Dhiman and Kumar [27] studied the flow of non-Newtonian power-law fluids over a triangular cylinder confined in a planar channel. They reported that for a fixed value of the Reynolds number, friction component of drag coefficient and pressure component of drag coefficient decreases with decrease in power-law index. Sun and Zhang [28] studied numerically a non-Newtonian nanofluid in a microplanar sudden expansion. They indicated that there exists a huge overestimate on the system pressure drop if Newtonian model is taken instead of non-Newtonian model. Nikfarjam and Sohankar [29] investigated power-law fluid flow from a single and two tandem square cylinders in the laminar fluid flow regimes. They showed that the pressure and total drag coefficients of the upstream cylinder decrease slightly and then increase with the power-law index. Tian *et al.* [30] prepared two shear thinning fluids from xanthan gum (XG) and propylene oxide solutions under alkaline condition. Their measurements showed that these two fluids can reduce the pressure drop of straight pipe. Rao *et al.* [31] described the flow and heat transfer from a square cylinder immersed in power-law fluids. They found that the effect of power-law index on both the pressure drag and heat transfer coefficients diminishes with the increasing in the Reynolds number. Bharti *et al.* [32] investigated the pressure drop of incompressible power-law fluids through a cylindrical microchannel at low Reynolds number. They showed that the pressure drop increases with increasing shear-thinning behavior. Borzenko *et al.* [33] simulated numerically the axisymmetric power-law fluid flow in a pipe with a sudden contraction. They showed that the local resistance coefficient increases with increase in the power-law index. Dosunmu and Shah [34] investigated the laminar flow behavior of non-Newtonian fluids using Carreau and modified power-law models and derived a new flow rate–pressure drop equation. Babaie *et al.* [35] presented an empirical correlation between friction pressure and generalized Reynolds number for annular flow of power-law fluids. Li *et al.* [36] investigated the three dimensional laminar flow and forced convective heat transfer of carboxy methyl cellulose aqueous solutions as shear-thinning power-law fluids in protruded microchannel. They proposed a new correlation which considers the effect of the power-law index on the relative friction factor and Nusselt number.

The effect of microchannels structure on the flow pattern was also investigated. Mei *et al.* investigated the flow distribution in an A-type (with one inlet and two outlets) microchannel reactor for hydrogen

production [37]. Their results showed that the standard deviation of the velocity flow field in the A-type microreactor was about a third of that Z-type microreactor. Balaji *et al.* designed a novel microchannel plate configuration to obtain uniform flow distribution at high velocities [38]. Their results showed that the microchannels with one inlet and two outlet ports in line with the microchannels lead to more uniform flow distribution than other configuration. However this work restricted to Newtonian fluids.

Most of the above studies were focused on the flow behavior or heat transfer coefficients of microchannels and more generally restricted to Newtonian fluids. The pressure drop optimization through microchannels was less considered. On the other hand non-Newtonian fluids and especially pseudoplastic fluids have extraordinary potential for lowering pressure drop of the flows and thus reduction of energy consumption as their viscosity reducing with increase in shear rate. Therefore, the main object of the present study is numerically study the pressure drop of power-law fluids through a microchannel plate. The configuration of this microchannel was optimized previously with Balaji *et al.* [38]. In addition, the flow distributions through the whole of the microchannel, flow pattern in each slit and along outlets ports were investigated. Also, the heat transfer of the power-law fluids through the microchannel with various configurations and in a wide range of Reynolds number was investigated. The power-law fluids with various flow indexes (n) and different operating conditions were considered here.

2. Governing Equations

The geometry of the microchannel is similar to that described by Balaji *et al.* [38]. Briefly, a rectangular microchannel plate with dimensions of $8.6 \text{ mm} \times 7 \text{ mm}$ and with different channel numbers was considered here. The widths of plates between channels were set to 0.2 mm . Fig. 1 shows the 2-D view of microchannel geometry with delineated centerline, flow directions, channel passages, inlet and outlets arcs. The inlet and outlet ports had circular shape which showed that have better performance than other geometrical shapes. One inlet and two outlets in the figure are linked to one microplate and the other three voids are related to another microplate (as a possible counter-current microheat exchanger) which were considered here as solid bodies.

Three main continuity, momentum and energy equations in 2-D Cartesian coordinates were solved for the entire geometry:

$$\nabla \cdot \mathbf{U} = 0 \quad (1)$$

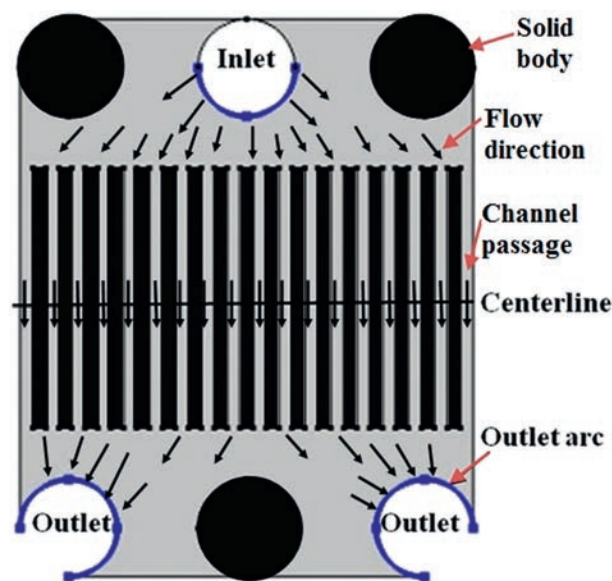


Fig. 1. Microchannel geometry with defined centerline, inlet and outlet arcs.

$$\rho \frac{\partial \mathbf{U}}{\partial t} + \rho (\mathbf{U} \cdot \nabla) \mathbf{U} = \nabla \cdot (\boldsymbol{\tau} - p\mathbf{I}) \quad (2)$$

$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p \mathbf{U} \cdot \nabla T = \nabla \cdot (k' \nabla T) \quad (3)$$

where $\mathbf{U}$, $\mathbf{I}$, p , T , ρ , C_p and k' are the velocity vector, identity tensor, pressure, temperature, fluid density ($1000 \text{ kg} \cdot \text{m}^{-3}$), heat capacity ($4182 \text{ J} \cdot \text{kg}^{-1} \cdot ^\circ\text{C}$) and heat conductivity coefficient ($0.6 \text{ W} \cdot \text{m}^{-1} \cdot ^\circ\text{C}$), respectively. $\boldsymbol{\tau}$ is the stress tensor for the power-law fluids:

$$\boldsymbol{\tau} = K(\dot{\gamma})^n \quad (4)$$

where K , n and $\dot{\gamma}$ are consistent coefficient, flow index and the flow shear rate, respectively. The fluid was supposed to be incompressible and thus its density was remained constant. The inlet velocity (u_0) was set to constant values. High velocity values were chosen to assess the flow distribution through the microchannel, while the laminar flow regime situation was satisfied. The pressure at the outlet was zero. No slip boundary conditions (zero velocity) at the microchannel walls were specified. The inlet fluid and wall temperatures were set to 290 K and 320 K, respectively.

The Darcy friction factor and Nusselt number are defined as follows:

$$f = \frac{2\Delta p D}{L \rho u_0^2} \quad (5)$$

$$Nu = \frac{hD}{k} \quad (6)$$

where D , L and h are the characteristic length of geometry (the width of the slits), the microchannel length and the heat transfer coefficient, respectively. The modified Reynolds number for power-law fluids is [39]:

$$Re' = \frac{8D^n u^{2-n} \rho}{K} \left(\frac{0.5n}{3n+1} \right)^n \quad (7)$$

Since the main differences between constitutive equations of power-law and Newtonian fluids are the non-unity flow index of power-law fluids, this study focused mainly on this parameter. Therefore the flow indexes varied from 0.4 to 1, because the power-law fluids were considered as pseudoplastic (except for $n = 1$ which was related to Newtonian fluid). The values of K were chosen such that the modified Reynolds number was below 1000. The choice of low values of flow indexes or consistent

coefficient leads to very high Reynolds numbers (at constant velocities) and changes the flow regime from laminar to turbulent one.

Non-uniform triangular mesh elements were used for the flow domains. Mesh resolution was fine and intensified close to the wall and near the outlet arcs. To assess the effect of grid size parameter on the solution accuracy, the mesh resolution increased until the error between two refinement steps was lower than 1%. Fig. 2 shows the effect of grid size on the pressure and average velocity in the middle slit. When grid size was smaller than 0.05 mm, the solutions were considered as independent of grid size. Similar procedures were performed for other fluids. The convergence criteria for residuals were set to 10^{-4} as it is an accepted value for most numerical simulations [40]. The simulations were performed with 110866 elements and 176448 degree of freedom using finite element method.

In addition, for investigation of the validity and certitude of the numerical procedure, a primary simulation was performed based on the work of Balaj *et al.* with their microchannel configuration and fluids characteristics [38]. The average velocity at the middle slit was chosen as criterion variable and the Reynolds number was varied from 5 to 200. Fig. 3 shows the comparison between our results and their data. As can be seen there is good agreement between the two results.

3. Results and Discussion

3.1. Velocity distribution along the microchannel plate

Fig. 4 shows the surface plot of velocity in the microchannel plate. The velocity is higher at the inlet ports and decreases through the microchannel. Although the flow velocity near the walls is very low. It seems that the flow near these regions is very prone to be stagnant. The main difference between the velocity profiles of different power-law fluids is at the out of slits. For low power-law index, the flow streams don't admix near the slits in contrast with Newtonian fluid where the flow is more homogeneous. The viscosity dependence of power-law fluids to shear rate causes their velocity profiles are more sensible to sudden expansion [28].

Fig. 5 shows the velocity distribution along the microchannel slits of power-law fluids with different flow indexes (n). The profiles relate to the centerline halfway of the microchannel plate (which was described in Fig. 1). The horizontal axis is the distance from the left edge of the microchannel plate ($x = 0$ relates to the left side and $x = 3.5 \text{ mm}$ located at the central point of the whole microchannel plate). The velocities between the microchannel slits are zero since there is no flow there.

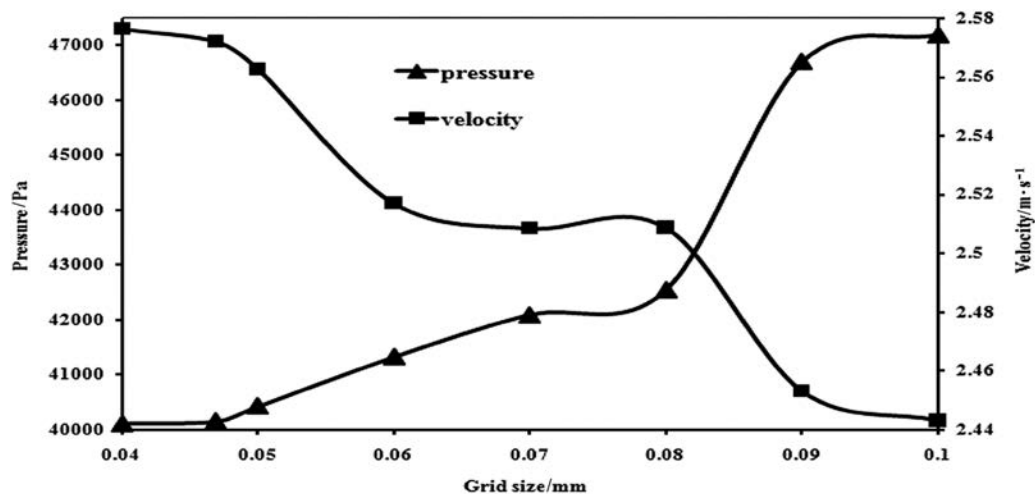


Fig. 2. Effect of maximum grid size on the pressure and average velocity in the middle slit ($U_0 = 1 \text{ m} \cdot \text{s}^{-1}$, $K = 1 \text{ Pa} \cdot \text{s}^n$, $n = 0.6$).

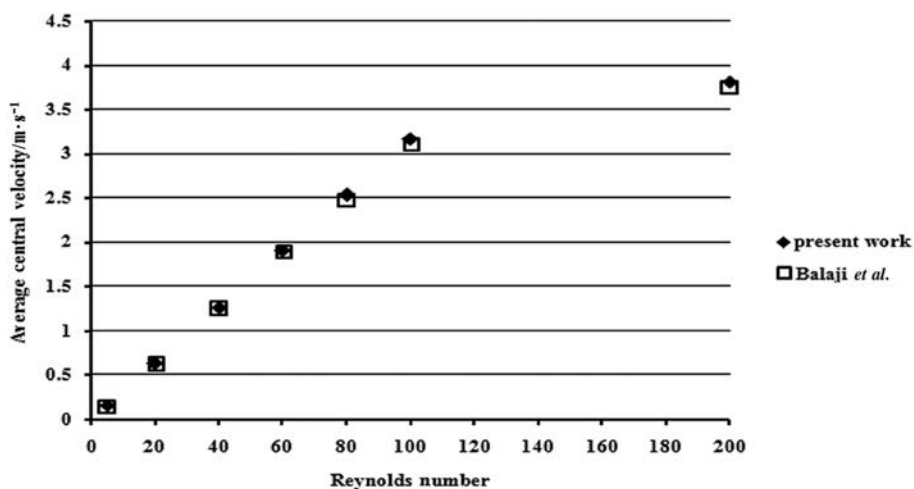


Fig. 3. Comparison between the present work and Balaji *et al.* results [38]. The average velocity at the middle slit was chosen as criterion variable ($\rho = 1 \text{ kg} \cdot \text{m}^{-3}$, $\mu = 10^{-5} \text{ Pa} \cdot \text{s}^{-1}$).

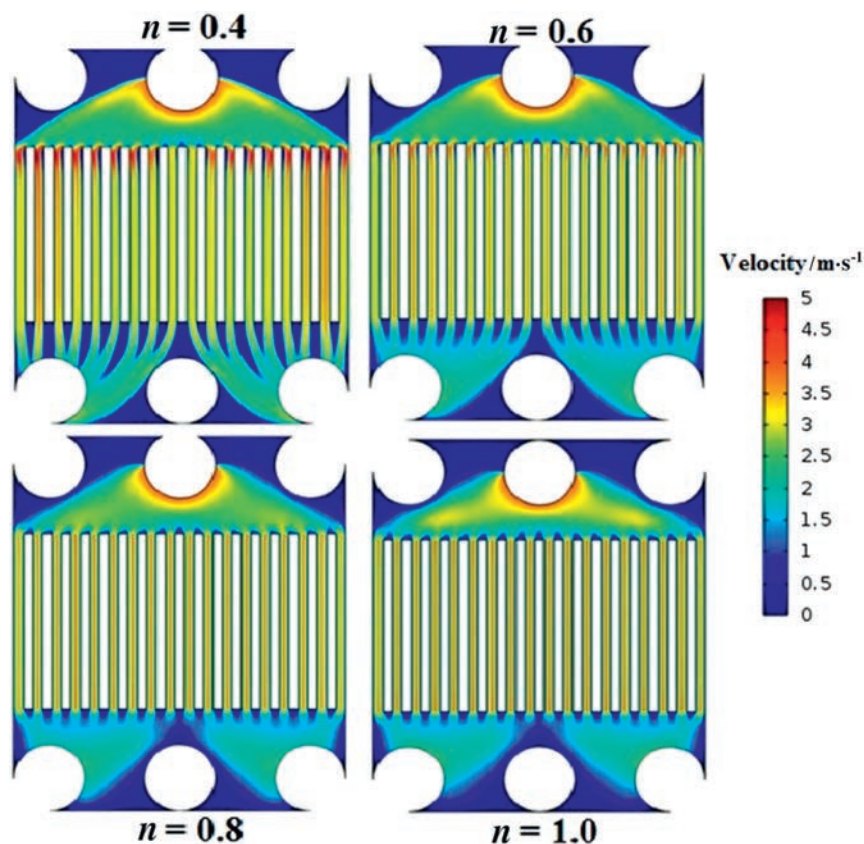


Fig. 4. Surface plot of velocity in the microchannel plate of power-law fluids with different flow indexes ($u_0 = 4 \text{ m} \cdot \text{s}^{-1}$).

As can be seen, the velocity increases with an increment of flow index. That is, the average velocity for $n = 1$ is higher than that of $n = 0.4$. At higher flow indexes, the velocity profiles of different microchannel slits are more consistent with each other than lower flow indexes. For instance, the velocity profiles related to $n = 0.4$ are non-uniform along the microchannel slits in contrast with $n = 1$. Except the last left slit, the profiles are more and less similar for $n = 1$. The sensitivity of viscosity to shear rate for power-law fluids, is higher for lower flow indexes. Thus the velocity profiles are more variable for these fluids than those for Newtonian fluid.

Table 1 shows the difference between maximum and minimum velocities along the microchannel slits. The difference between maximum and minimum velocities is higher for lower flow indexes. However, these differences are not very much. For example, when the inlet velocity is $4 \text{ m} \cdot \text{s}^{-1}$, the velocity difference (Δv) for $n = 0.6$ and $n = 1$ are about 0.23 (5.7%) and $0.12 \text{ m} \cdot \text{s}^{-1}$ (3.1%), respectively. On the other hand, the pressure drops of pseudoplastic fluids are very smaller than Newtonian ones and additionally the velocity profile in the slits at low flow index is more smooth and plug like despite parabolic velocity profile for $n = 1$ (these will be discussed in the next sections).

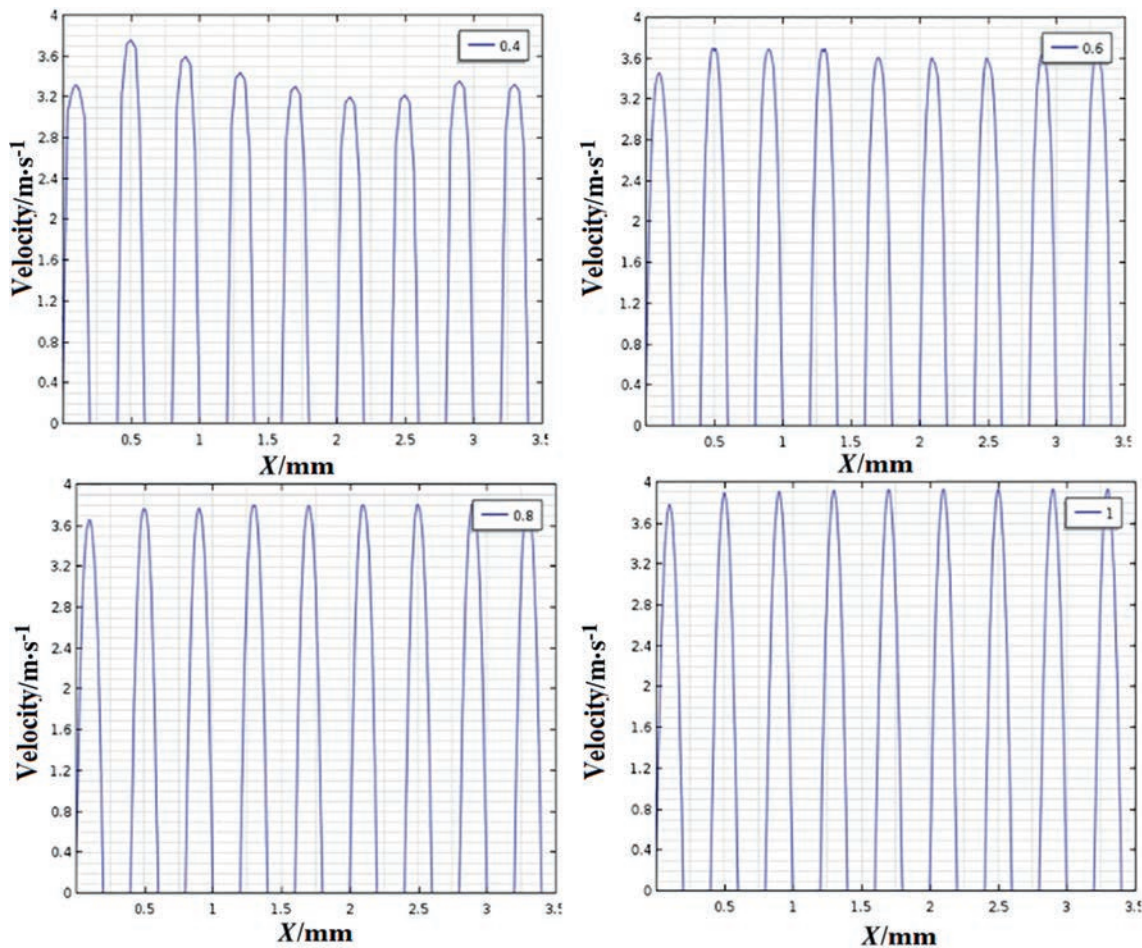


Fig. 5. Velocity distribution along the microchannel plate slits for different flow indexes ($u_0 = 4 \text{ m} \cdot \text{s}^{-1}$).

Table 1
Characteristics of the velocity distribution of power-law fluids with different flow indexes at the centerline of the microchannel. The data are based on maximum velocities

Inlet velocity/ $\text{m} \cdot \text{s}^{-1}$	Flow index (n)	Average velocity/ $\text{m} \cdot \text{s}^{-1}$	Standard deviation/ $\text{m} \cdot \text{s}^{-1}$	Maximum velocity/ $\text{m} \cdot \text{s}^{-1}$	Minimum velocity/ $\text{m} \cdot \text{s}^{-1}$	Velocity difference/ $\text{m} \cdot \text{s}^{-1}$
1	0.4	0.854041	0.024451	0.873568	0.793395	0.080174
	0.6	0.901831	0.018493	0.914612	0.85695	0.057662
	0.8	0.942649	0.013818	0.95111	0.908678	0.042432
	1.0	0.975818	0.010287	0.982065	0.950203	0.031862
4	0.4	3.388336	0.180082	3.74286	3.174173	0.568687
	0.6	3.604192	0.067749	3.689303	3.461991	0.227313
	0.8	3.770938	0.048181	3.806057	3.651073	0.154984
	1.0	3.903284	0.039708	3.927534	3.803886	0.123648

3.2. Velocity profile in one microchannel slit

Since the flow rate at the middle slit is higher than others, its velocity profile is more susceptible to be non-uniform. Fig. 6 compares the velocity profiles through the central microchannel slit for different flow indexes. As can be seen, the profile relates to $n = 1$, have high gradient. The slope of curves declines as n decreases such that the profile relates to $n = 0.4$ is approximately like the plug flow. Similar results were reported previously. For example, Zhao *et al.* simulation showed that the velocity profile of power-law fluids becomes more plug-like as the fluid flow index decreases [41]. Using momentum and continuity equations the velocity in a narrow slit (v_z) can be expressed by [42]:

$$v_z = \left(\frac{(p_0 - p_L)B}{KL} \right)^{\frac{1}{n}} \frac{B}{\left(\frac{1}{n} \right) + 1} \left(1 - \left(\frac{x}{B} \right)^{\left(\frac{1}{n} \right) + 1} \right) \quad (8)$$

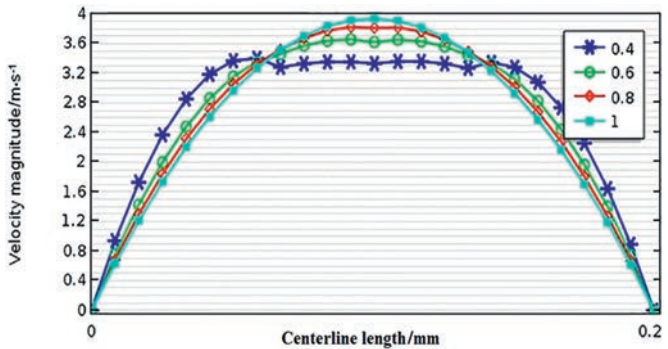


Fig. 6. Comparison of velocity profiles in one slit (middle slit) for different flow indexes ($u_0 = 4 \text{ m} \cdot \text{s}^{-1}$).

where p_0 , p_L , B , x and L are pressure at entrance and exit point, half of slit width, distance from center of slit and slit length, respectively. The derivative of v_z with respect to x is proportional to $(x/B)^{1/n}$. The (x/B) is lesser than one and thus for larger $1/n$ (smaller n) the velocity profile slope tends to zero. This means that the residence time distribution in one slit is narrow for small n and the fluid particles in different position (x) have the same velocity and exit from the slit at the same time.

Table 2 summarizes the distribution characteristics of the velocity profiles which confirm the above observations. The standard deviation increases with increase in the flow index. Also the maximum to average velocity ratio is higher for Newtonian fluids than power-law fluids. The difference is more distinctive at higher flow rates. It is noteworthy that in the Balaji *et al.* work the main attention was to average velocity in different microchannels [38], while the velocity distribution through each slit of the microchannel is as important as average velocity. This can be very important as any non-uniformity can decrease the performance of the system.

Table 2

Characteristics of the velocity profiles of power-law fluids for different flow indexes through the middle slit of the microchannel

Inlet velocity/ $\text{m}\cdot\text{s}^{-1}$	Flow index (n)	Average velocity/ $\text{m}\cdot\text{s}^{-1}$	Maximum velocity/ $\text{m}\cdot\text{s}^{-1}$	Standard deviation/ $\text{m}\cdot\text{s}^{-1}$
1.0	0.4	0.694228	0.883065	0.245048
	0.6	0.683562	0.925855	0.265785
	0.8	0.673383	0.954816	0.279582
	1.0	0.667004	0.983945	0.290207
4.0	0.4	2.752983	3.398615	0.913227
	0.6	2.775374	3.655442	0.970875
	0.8	2.695916	3.822647	1.11932
	1.0	2.725515	3.935056	1.160613

3.3. Velocity distribution at the outlets

Fig. 7 shows the velocity profile along the arc length of the left microchannel outlet. Due to microchannel symmetry form, the velocity profile for the right outlet is the same as the left one and thus not shown here. It is interesting to note that even for $n = 1$, the velocity profile at the middle zone of the outlet arc length is approximately plug-like and the velocity is approximately constant. However, it seems that the length of this zone (plateau length) is longer for low flow index ($n = 0.4$). This can be important because the velocity out of the plateau zone decreases rapidly and at the corners of the outlet tends to zero value. The zero velocities lead to stagnation points and reduce the efficiency of the microchannel plate. These dead zones are the limitations of this geometry as mentioned by Ballaj *et al.* [38]. Therefore fluids with a low flow index can be used to lessen these zones and overcome the drawback of this geometry.

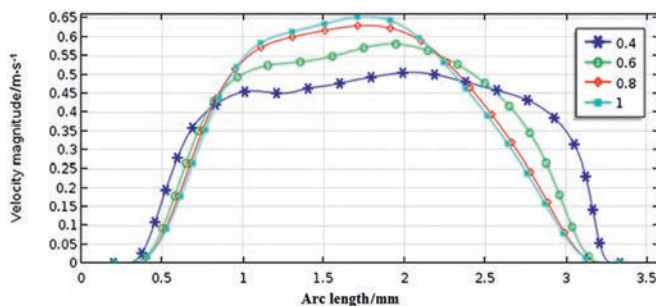


Fig. 7. Velocity profiles along arc length of the left outlet for different flow indexes ($u_0 = 1 \text{ m}\cdot\text{s}^{-1}$).

Table 3 shows the plateau length, maximum and standard deviation for these velocity profiles. The velocity profile with lower n has smaller maximum velocity. The maximum velocity for $n = 1$ is about 20% larger than that of $n = 0.4$. However, the velocity distribution is more uniform for lower n . The variance of velocity distribution is higher for larger n which is the same as the previous section. It is interesting that the average velocities over the outlet arc for different flow indexes are the same. It is expected, because the problem was solved for the steady state situation. Due to continuity and incompressibility assumption of the fluid, the outlet mass flows must be equal to the inlet one and thus the average velocities at the outlets will be the same.

Table 3

Maximum, standard deviation and plateau length of velocity profiles along arc length of the left outlet

Inlet velocity/ $\text{m}\cdot\text{s}^{-1}$	Flow index (n)	Maximum velocity/ $\text{m}\cdot\text{s}^{-1}$	Standard deviation/ $\text{m}\cdot\text{s}^{-1}$	Plateau length/mm
1.0	0.4	0.50	0.20	1.7
	0.6	0.58	0.22	1.2
	0.8	0.63	0.23	1.1
	1.0	0.65	0.23	0.9
4.0	0.4	2.58	0.78	2.6
	0.6	1.93	0.81	1.9
	0.8	2.31	0.87	0.9
	1.0	2.54	0.91	0.8

3.4. Pressure drop

Microdevices with high flow output cause to high pressure drop. Thus, a higher pressure should be applied at the inlet port and this leads to higher energy consumption and in addition more expensive equipments are needed. For example, it is well known that the pumping power is proportion to the pressure difference between the inlet and outlet of the devices [43]. Therefore the reduction of pressure difference can decrease energy consumption.

In this section the pressure drop through the inlet and outlet zones was studied. **Table 4** shows the pressure drop at different inlet velocities. It is very interesting that when the inlet velocity is $4 \text{ m}\cdot\text{s}^{-1}$, the pressure drop for $n = 0.4$ (3.5 kPa) is about 1000 times smaller than that of Newtonian fluid (3294.5 kPa). This huge difference can actually save a large amount of energy. Even at lower inlet velocity ($1 \text{ m}\cdot\text{s}^{-1}$), the pressure drop for $n = 1$ is more than 300 times larger than that of $n = 0.4$. Thus with increasing throughput in the microchannel, the difference gets larger. For more investigation, the pressure drop can be evaluated from Eq. (5) by supposing that the most pressure drop occurs along the slits of the microchannel plate:

$$\Delta p = \left(\frac{v_y}{\frac{nB}{1+n} \left(1 - \left(\frac{x}{B} \right)^{\frac{n+1}{n}} \right)} \right)^n \frac{KL}{B} \quad (9)$$

Table 4

Pressure drop through the inlet and outlet zones

Inlet velocity/ $\text{m}\cdot\text{s}^{-1}$	Flow index (n)	Pressure difference/kPa
1.0	0.4	1.2
	0.6	9.1
	0.8	61.6
	1.0	411.5
4.0	0.4	3.5
	0.6	40.2
	0.8	371.6
	1.0	3294.5

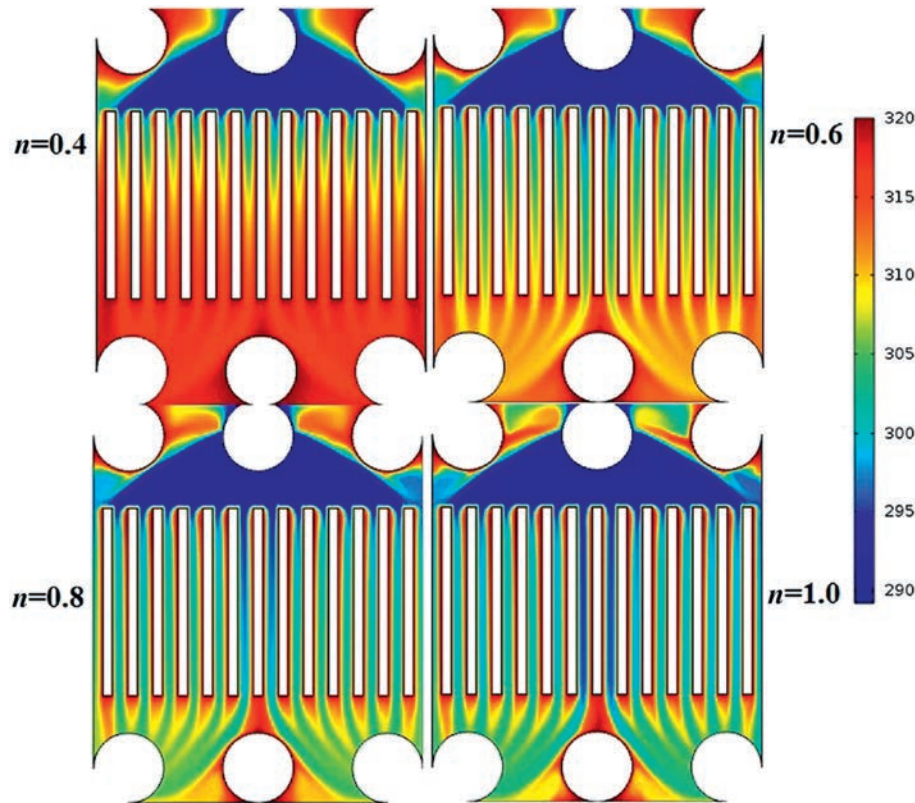


Fig. 8. Temperature profile through the microchannel plate for power-law fluids with different flow indexes ($Re' = 1000$).

From the above equation, it is clear that at the same flow velocity (V_y) and position (x), the ΔP decreases rapidly as flow index decreases. Inversely, if the same ΔP applied to the microchannel, the flow rate is proportion to $\Delta P^{1/n}$ which means that for pseudoplastic fluids the flow rate will be very larger than Newtonian fluids at the same ΔP . As a result, it is possible to achieve high flow rates with small ΔP . Addition of small amount of polymers to low molecular weight solvents (e.g. water) can change their rheological properties from Newtonian behavior to power-law fluids [44]. These modified fluids can be replaced by common fluids in microdevices to decrease energy consumption. Consequently, it is possible to use this microchannel plate configuration using power-law fluids even under high velocity conditions, without much energy consumption or large non-uniformity in the flow distribution.

3.5. Heat transfer

Fig. 8 shows the temperature profiles through the microchannel for different power-law flow indexes at Reynolds number of 1000. The outlet temperature increases with the decrease of the flow index. According to Eq. (7), at constant Reynolds number the inlet velocity decreases with the decrease of the flow index. When the flow rate decreases, the fluid has more time span to contact with the hot walls. Therefore, its final temperature increases significantly.

Fig. 9 displays the effect of Reynolds number on the friction factor and Nusselt number for different flow indexes. It seems that the friction factor decreases with increment of Reynolds number. It is well known that the friction factor is inversely proportional to Reynolds number. Though,

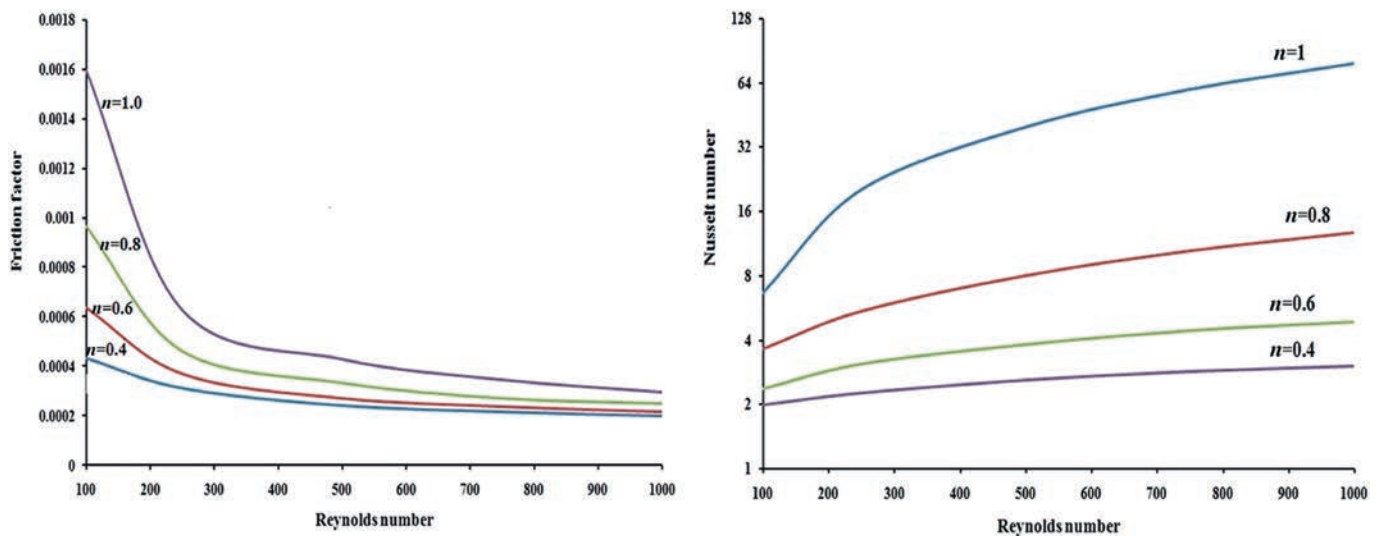


Fig. 9. Effect of Reynolds number on the friction factor and Nusselt number for fluids with different flow indexes.

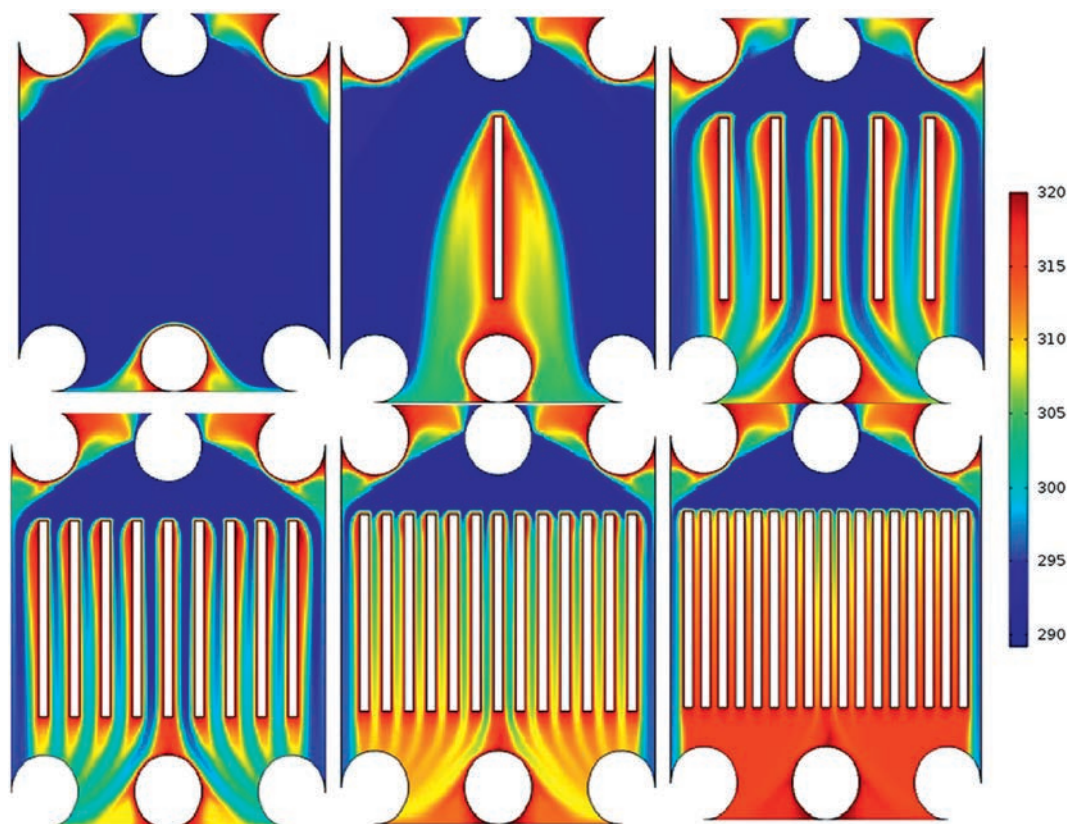


Fig. 10. Temperature profile of the microchannel with different channel numbers ($n = 0.8$, $Re' = 1000$).

this reduction is more significant at low Reynolds number. Also, the friction factor and Nusselt number have higher values for fluids with higher flow index due to their higher pressure drop through the microchannel. Although the difference between the fluids is not significant at high Reynolds number. These findings are in consistence with the other results in the literature. For example, Rao *et al.* [31] showed that the pressure drag coefficient decreases with the decrease of power-law flow index and with increase in the Reynolds number, the effect of power-law index becomes smaller. The Nusselt number increases with increment of Reynolds number and power-law index at constant Reynolds number. As mentioned above, at constant Reynolds number, the velocity decreases with decrease of power-law index. Therefore the Nusselt number was lower for power-law fluids than that of Newtonian fluid.

Fig. 10 shows the temperature profile of the microchannel with different channel numbers. It is obvious that the outlet temperature increases with increase of the channel numbers. Higher channel number increases the surface area and raises the heat transfer and the fluid final temperature. Fig. 11 shows the effect of channel number on the friction factor and Nusselt number for fluids with different power-law indexes. The friction factor and Nusselt number have higher values for Newtonian fluid than pseudoplastic fluids. The effect of channel number on the friction factor and Nusselt number is contradictory. Firstly, both friction factor and Nusselt number increase rapidly, when the channel number increases. However, the friction factor decreases rapidly and past through a minimum and again rises slowly. In the case of Nusselt number, the trend is more complicated and depends on the flow index.

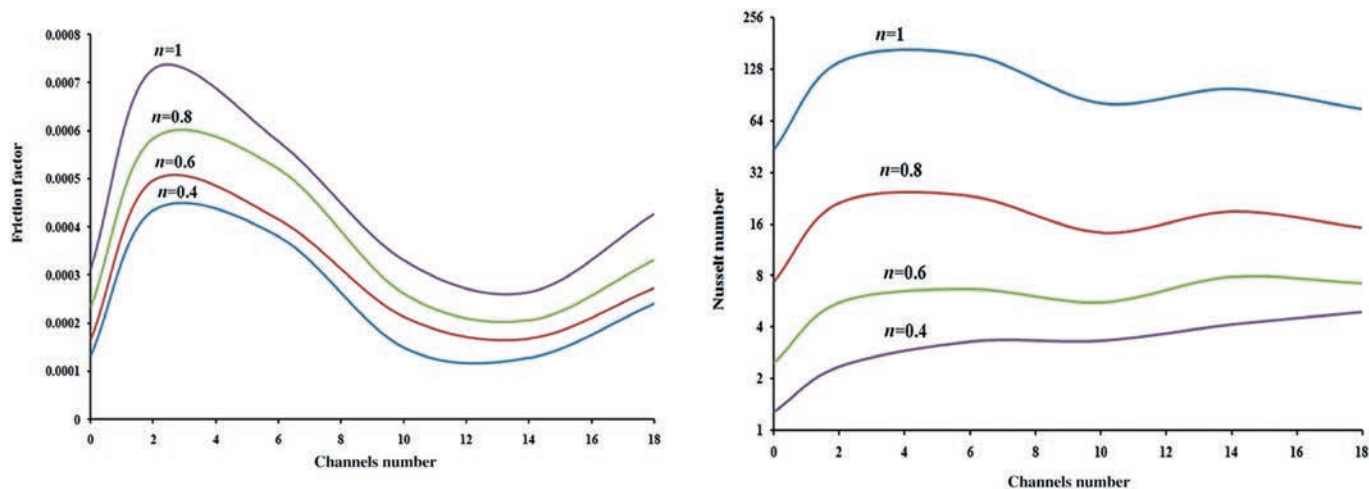


Fig. 11. Effect of channel number on the friction factor and Nusselt number for fluids with different power-law indexes.

For example, the Nusselt number for Newtonian fluid increases at low channel number and then decreases. But for fluid with low power-law index ($n = 0.4$) the Nusselt number increases monotonically with increase in the channel number. This shows that the power-law fluids with low fluid index have higher heat transfer efficiency.

4. Conclusions

The flow distribution, pressure drop and heat transfer of power-law fluids with different flow-indexes at a microchannel plate have been studied under different operating conditions. The velocity profile along various microchannel slits was more similar for Newtonian fluid. However the non-uniformity for pseudoplastic fluids was not significant. On the other hand, the study of velocity distribution for low flow index fluids in one microchannel slit was similar to the plug flow. The velocity profile for $n = 1$ was parabolic and was very non-uniform which can lead to different residence time for particles with different position. Also the velocity distribution through the outlet was studied. The results showed that the fluids with low flow index ($n = 0.4$) had more uniform velocity profiles than that of Newtonian fluid. In addition, the power-law fluids with lower n have smaller stagnation zones. The most interesting results were the pressure difference between the inlet and outlet ports. The pressure drop varied over a wide range for different flow-indexes. This variation can reach orders of magnitudes at high flow rates such that the pressure drop for $n = 0.4$ was very smaller than that of $n = 1$. Also, the effect of Reynolds number and channel number on the friction factor and Nusselt number for power-law fluids with different flow indexes were investigated. The friction factor decreases with increase in Reynolds number in spite of Nusselt number. The effects of channel number were contradictory. At low channel number both friction factor and Nusselt number increase. But the friction factor decreases with increment of channel number. The Nusselt number of Newtonian fluid decreases with increase in channel number. However, for power-law fluid with flow index $n = 0.4$, the Nusselt number increases continuously as the number of channels increases. These results show the use of pseudoplastic power-law fluids in the microchannels such as microheat exchanger which can lower the pressure drop and increase the heat transfer efficiency.

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