Study of the axial heat conduction in parallel flow microchannel heat exchanger

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Abstract In this paper the axial heat conduction in an isosceles right triangular microchannel heat exchanger is numerically investigated, for laminar, 3D, incompressible, single-phase, steady state flow. The behaviour of axial heat conduction in the separating wall under different conditions is studied. The solution was obtained by solving the continuity and Navier–Stokes equations for the hot and cold fluids by using the pressure-correction method to obtain the velocity distribution, and then the energy equations were solved for the two fluids and the separating wall simultaneously to obtain the temperature distribution. The governing equations are discretized using finite-volume and the hybrid differencing scheme with FORTRAN code was used. Various parameters that can have effect on the axial heat conduction were investigated.

The results showed that, the axial heat conduction plays an important role in a parallel flow microchannel heat exchanger and the factors affecting the local and average axial heat conduction are; Reynolds number (Re), thermal conductivity ratio (Kr), hydraulic diameter (Dh), thickness of separating wall (ts) and channel volume. Increasing of Re, Kr and ts leads to an increase in the axial heat conduction while increasing of Dh and channel volume leads to a decrease in the axial heat conduction.

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1. Introduction

The rapid development of modern electronics industry has necessitated effective cooling techniques which are capable of dissipating ultra-high heat flux of about 100 W/cm² from the highly integrated microelectronics systems to ensure a stable and optimal operation (Liu and Lee, 2003). A variety of novel cooling schemes have been proposed and investigated to meet this demand requirement, among which, the microchannel heat exchanger is especially promising for its superior thermal performance. Compared with conventional heat exchangers, the main advantage of the microchannel heat exchangers is their extremely high heat transfer area per unit volume. As a result, the overall heat transfer coefficient per unit volume can be as greater than 100 MW/(m² K), which is much higher than that for conventional heat exchangers (1–2 orders of magnitude) (Jian et al., 2001). In the conventional heat exchanger the solid thickness is comparatively small to the hydraulic diameter; therefore the axial heat conduction may be neglected. This
means that, the performance of heat exchanger is primarily dependent on the flow in the ducts (fluid properties and mass flow rate). But, for a microchannel heat exchanger, the solid thickness is comparatively large to the hydraulic diameter, therefore the axial heat conduction in the separating wall (solid) is considerable and the microchannel heat exchanger effectiveness may decrease due to the effect of axial heat conduction.

Kroeger (1967) solved the governing equations of a two stream counter flow heat exchanger, taking into account the effect of axial heat conduction, he presented a closed form solution for finding the effectiveness of a balanced flow heat exchanger using a simplified one dimensional plate separating the two fluids. He showed that, the effectiveness is largely dependent on the axial heat conduction in the separating wall.

Wang and Shyu (1991) experimentally studied the effect of channel size and wall thermal conductivity in micro heat exchangers for a hot/cold water test loop. They showed that the channel size and wall material have a strong influence on the heat transfer capability of a micro heat exchanger.

Yin and Bau (1992) studied the flow between infinite parallel plates and circular pipes to study the effect of axial heat conduction on the performance of microchannel heat exchangers. They used a fully developed velocity field and analytically solved for the temperature fields in the channel and solid wall. They found that, the axial conduction plays an important role at the entrance region.

Stief et al. (1999) numerically investigated the effect of solid thermal conductivity in micro heat exchangers. They showed that the reduction of conductivity of the wall material can improve the heat transfer efficiency of the exchanger due to the influence of axial heat conduction in the separation wall.

Vekatarathanam and Narayanan (1999) also found that the performance of the heat exchanger is largely dependent on the heat conduction that takes place through the walls of the heat exchangers used in miniature refrigerators. They used a two dimensional energy balance to account for the conduction through wall and convection through the fluid. They assumed the temperature of the fluid to be uniform at any flow cross section.

Al-bakhit and Fakheri (2005) numerically investigated the parallel flow microchannel heat exchanger with rectangular ducts. They showed that using a high conductive material will not have an effect on increasing the heat exchanger effectiveness since the heat exchanger effectiveness will be independent of the wall thermal conductivity for Kn above 40.

Al-bakhit and Fakheri (2006) numerically investigated the parallel flow microchannel heat exchanger with rectangular ducts. They showed that the overall heat transfer coefficient is rapidly changed for x/DpPe (Graetz number) below 0.03, and therefore the assumption of constant overall heat transfer coefficient is not valid if the Graetz number based on the heat exchanger length is of the order of 0.03. Also, the accurate results can be obtained by solving the thermally developing energy equation using fully developed velocity profiles.

Al-Nimr et al. (2009) numerically investigated the hydrodynamics and thermal behaviours of the laminar, 2D, fully developed, slip flow inside an insulated parallel-plate microchannel heat exchanger. They showed that both the velocity slip and the temperature jump at the walls increased with increasing Knudsen number.

Hasan (2009) made numerical investigation to study the counter flow microchannel heat exchanger with different channel geometries and working fluids. He studied the effect of axial heat conduction on the performance of counter flow microchannel heat exchanger with square shaped channel.
and he found that the existence of axial heat conduction leads to a reduction in the effectiveness of this heat exchanger.

2. Mathematical formulation

Simulation of full continuity, momentum and energy equations for different fluid flows arises in various engineering problems. Various different algorithms have been proposed and developed by various researchers. But an approach that is fully robust from the point of view of numerical and modeling accuracy as well as efficiency has yet to be developed.

The configuration shown in Fig. 1 represents a unit cell of microchannel arrays. It includes two isosceles right triangular upper channel at its height and base of channel, and developed by various researchers. But an approach that is fully robust from the point of view of numerical and modeling accuracy as well as efficiency has yet to be developed.

The configuration shown in Fig. 1 represents a unit cell of microchannel arrays. It includes two isosceles right triangular channels with a separating wall between them where, L is channel length, D is height and base of channel, ts is solid thickness and S is projected distance of ts. In this figure, the hot fluid enters the lower channel with a uniform velocity uh,in and uniform temperature Th,in, while the cold fluid enters the upper channel at u,h,in and Tc,in. Heat is transferred from the hot fluid to the cold fluid through the inclined wall of thickness ts. However, the governing equations for the present model are based on the following physical and geometrical assumptions:

- The flow is laminar and steady.
- The Knudsen number is small enough so that, the fluid is a continuous medium (no slip).
- The fluids are incompressible, Newtonian with constant properties; in this case the water is used as working fluid.
- There is no heat transfer to/from the ambient.
- The energy dissipation is negligible.
- The pressure gradient is in axial direction only.
- Three dimensional is the flow and heat transfer.

Based on these assumptions, the governing equations and boundary conditions in Cartesian coordinates are written as below (Hasan, 2009):

**x-Momentum equation:**

$$\rho_i \left( u_i \frac{\partial u_i}{\partial x} + v_i \frac{\partial u_i}{\partial y} + w_i \frac{\partial u_i}{\partial z} \right) = -\frac{\partial p_i}{\partial x} + \mu_i \left( \frac{\partial^2 u_i}{\partial x^2} + \frac{\partial^2 u_i}{\partial y^2} + \frac{\partial^2 u_i}{\partial z^2} \right)$$

(1)

**y-Momentum equation:**

$$\rho_i \left( u_i \frac{\partial v_i}{\partial x} + v_i \frac{\partial v_i}{\partial y} + w_i \frac{\partial v_i}{\partial z} \right) = \mu_i \left( \frac{\partial^2 v_i}{\partial x^2} + \frac{\partial^2 v_i}{\partial y^2} + \frac{\partial^2 v_i}{\partial z^2} \right)$$

(2)

**z-Momentum equation:**

$$\rho_i \left( u_i \frac{\partial w_i}{\partial x} + v_i \frac{\partial w_i}{\partial y} + w_i \frac{\partial w_i}{\partial z} \right) = \mu_i \left( \frac{\partial^2 w_i}{\partial x^2} + \frac{\partial^2 w_i}{\partial y^2} + \frac{\partial^2 w_i}{\partial z^2} \right)$$

(3)

**Continuity equation:**

$$\frac{\partial u_i}{\partial x} + \frac{\partial v_i}{\partial y} + \frac{\partial w_i}{\partial z} = 0$$

(4)

**Energy equation for fluid:**

$$\rho_i C_p \left( u_i \frac{\partial T_i}{\partial x} + v_i \frac{\partial T_i}{\partial y} + w_i \frac{\partial T_i}{\partial z} \right) = k_i \left( \frac{\partial^2 T_i}{\partial x^2} + \frac{\partial^2 T_i}{\partial y^2} + \frac{\partial^2 T_i}{\partial z^2} \right)$$

(5)

**Conduction equation for solid:**

$$\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} = 0$$

(6)

where i = h and c refers to the hot and cold fluids, respectively.

As mentioned in the previous assumptions the pressure gradients in y and z directions are negligible therefore, $\frac{\partial p_i}{\partial y} = \frac{\partial p_i}{\partial z} = 0$.

The boundary conditions are:

- **Lower channel (hot fluid)**

  $$At\ x = 0$$

  $$u_h = u_{h,in}, v_h = w_h = 0, \ T_h = T_{h,in}$$

  $$At\ x = L$$

  $$\frac{\partial u_h}{\partial x} = \frac{\partial v_h}{\partial x} = \frac{\partial w_h}{\partial x} = 0, \ \frac{\partial T_h}{\partial x} = 0$$

  $$At\ y = 0, \ 0 \leq z \leq D$$

  $$u_h = v_h = w_h = 0, \ \frac{\partial T_h}{\partial y} = 0$$

  $$At\ y = (D - z) \tan(45), \ 0 \leq z \leq D$$

  $$u_h = v_h = w_h = 0$$

  $$-k_h \frac{\partial T_h}{\partial n} = -k_s \frac{\partial T_s}{\partial n}, \ T_h = T_s$$

  $$At\ z = 0, \ 0 \leq y \leq D$$

  $$u_h = v_h = w_h = 0, \ \frac{\partial T_h}{\partial z} = 0$$

- **Upper channel (cold fluid)**

  $$At\ x = 0$$

  $$u_c = v_c = w_c = 0, \ T_c = T_{c,in}$$

  $$At\ x = L$$

  $$\frac{\partial u_c}{\partial x} = \frac{\partial v_c}{\partial x} = \frac{\partial w_c}{\partial x} = 0, \ \frac{\partial T_c}{\partial x} = 0$$

  $$At\ y = 0, \ 0 \leq z \leq D$$

  $$u_c = v_c = w_c = 0, \ \frac{\partial T_c}{\partial y} = 0$$

  $$At\ y = (D - z) \tan(45), \ 0 \leq z \leq D$$

  $$u_c = v_c = w_c = 0$$

  $$-k_c \frac{\partial T_c}{\partial n} = -k_s \frac{\partial T_s}{\partial n}, \ T_c = T_s$$

  $$At\ z = 0, \ 0 \leq y \leq D$$

  $$u_c = v_c = w_c = 0, \ \frac{\partial T_c}{\partial z} = 0$$

Figure 1 Schematic of parallel flow microchannel heat exchanger.
Study of the axial heat conduction in parallel flow microchannel heat exchanger

$$At \ z = (D - y) \tan 45, \ 0 \leq y \leq D$$

$$u_h = v_h = w_h = 0$$

$$-k_h \frac{\partial T_h}{\partial n} = -k_s \frac{\partial T_s}{\partial n}, \ T_h = T_s$$

$$At \ y = (D - z) \tan(45) + 2S, \ S \leq z \leq (D + S)$$

$$-k_c \frac{\partial T_c}{\partial n} = -k_s \frac{\partial T_s}{\partial n}, \ T_c = T_s$$

- Upper channel (cold fluid)

$$At \ x = 0$$

$$u_c = u_{c,0} \ v_c = w_c = 0, \ T_c = T_{c,0}$$

$$At \ x = L$$

$$\frac{\partial u_c}{\partial x} = \frac{\partial v_c}{\partial x} = \frac{\partial w_c}{\partial x} = 0, \ \frac{\partial T_c}{\partial x} = 0$$

$$At \ y = (D - z) \tan(45) + 2S, \ S \leq z \leq (D + S)$$

$$u_c = v_c = w_c = 0$$

$$-k_c \frac{\partial T_c}{\partial n} = -k_s \frac{\partial T_s}{\partial n}, \ T_c = T_s$$

$$At \ y = (D + S), \ 0 \leq y \leq S$$

$$\frac{\partial T_c}{\partial z} = 0$$

- Separating wall (solid)

$$At \ x = 0$$

$$\frac{\partial T_s}{\partial x} = 0$$

$$At \ x = L$$

$$\frac{\partial T_s}{\partial x} = 0$$

$$At \ y = (D - z) \tan(45), \ 0 \leq z \leq D$$

$$\frac{\partial T_s}{\partial z} = 0$$

where, \( n \) is the normal to the iso surfaces (isotherm lines). After establishing the governing equations and boundary conditions, the finite-volume method with FORTRAN code will be used to obtain a numerical solution to the problem.

The finite-volume method has become popular in Computational Fluid Dynamics CFD, this is because, first, it ensures that the discretization is conservative, i.e., mass, momentum, and energy are conserved in a discrete sense. Second, the finite-volume method does not require a coordinate transformation in order to be applied for irregular meshes. This flexibility can be used to great advantages in generation grids about arbitrary geometries (Lomax, 1999). Moreover, the finite-volume method combines the simplicity of the finite differences method with the local accuracy of the finite element method; also, at the same dimension of the discretized problem, the accuracy is higher than with finite differences and nearly the same as with finite elements (Petrila and Trif, 2005; Versteeg and Malalasekera, 1995).

After discretization of the governing equations and boundary conditions, a FORTRAN program is developed to perform the numerical solution to the problem. Firstly the distribution of velocity, pressure and temperature for both hot and cold liquids is obtained in addition to the distribution of temperature in solid wall. Then, the parameters such as the axial heat conduction \( q^a \) in the separating wall, the amount of heat transferred between two fluids and the effectiveness \( \varepsilon \)
which is the ratio of actual heat transfer to the maximum possible heat that can be transferred can be calculated from Ashman and Kandlikar (2006):

\[ \varepsilon = \frac{q_{\text{actual}}}{q_{\text{max, possible}}} \]  

(7)

\[ q_{\text{actual}} = \dot{m}_h c_v(T_{c,\text{out}} - T_{c,\text{in}}) = \dot{m}_c c_h(T_{h,\text{in}} - T_{h,\text{out}}) \]  

(8)

For convenience, the flow rates and specific heats are lumped together, and the term product of the capacity rates is

\[ C_v = \dot{m}_h c_v \quad \text{and} \quad C_h = \dot{m}_c c_h \]  

(9)

For \( C_h < C_v \),

\[ q_{\text{max}} = C_h(T_{h,\text{in}} - T_{c,\text{in}}) \]  

(10)

Otherwise

\[ q_{\text{max}} = C_v(T_{h,\text{in}} - T_{c,\text{in}}) \]  

(11)

Thus

\[ q_{\text{max}} = C_{\text{min}}(T_{h,\text{in}} - T_{c,\text{in}}) \]  

(12)

Then the effectiveness is

\[ \varepsilon = \frac{C_v(T_{c,\text{out}} - T_{c,\text{in}})}{C_{\text{min}}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_h(T_{h,\text{in}} - T_{h,\text{out}})}{C_{\text{min}}(T_{h,\text{in}} - T_{c,\text{in}})} \]  

(13)

For the axial heat conduction in section \( x \)

\[ Q_x^* = \left( k_s \left[ T_{i+1} - T_{i+1} \right] / \Delta x \right) \]  

(14)

where \( T_{i+1} \) is the average solid temperature in certain sections and \( T_{i+1} \) at the next section as shown in Fig. 2.

3. Results and discussion

The behaviour of axial heat conduction in the separating wall in a parallel flow microchannel heat exchanger under different conditions has been studied. The dimensions of triangular channels are 2 cm, 200 \( \mu \text{m} \) and 200 \( \mu \text{m} \) in \( x, y \) and \( z \) directions. Both hot and cold fluids are water with properties taken based on the average bulk temperature (viscosity = 0.000598 N s/m\(^2\), density = 981.3 \( \text{kg/m}^3 \), thermal conductivity = 0.643 W/m K, specific heat = 4189 J/(kg K)). The inlet temperatures of hot and cold fluids are 80 °C and 20 °C, respectively. The material of solid wall is variable; therefore the conductivity of material will be studied through the ratio \( k_v = k_s/k_f \). The grid was 100 nodes in \( x \) direction and 50 nodes in each of \( y \) and \( z \) directions.

In order to verify the accuracy of the present numerical model, a comparison is made between the results of present model for rectangular microchannel heat exchanger and that in literature. Fig. 3 represents a comparison between present numerical model and data of Al-bakhit and Fakheri (2006). The figure illustrates the variation of dimensionless mean temperature \( T_m = (T_{c,\text{out}} - T_{c,\text{in}}) / (T_{h,\text{in}} - T_{c,\text{in}}) \) for the hot and cold fluids in a rectangular microchannel heat exchanger with the dimensionless axial distance \( x^+ = \frac{x}{D_h Re Pr} \) at \( Re = 100 \), and \( \frac{\dot{m}_h}{\dot{m}_c} = 100 \). From this figure it can be seen that, there is a good agreement between the numerical results of present model and that for Al-bakhit and Fakheri (2006) and the maximum percentage error was ±2.01%.

Fig. 4 represents the distribution of central axial velocity along the direction of flow for different values of Reynolds number at \( D_h = 117 \mu \text{m} \). From this figure it can be noted that, the velocity increased in the entrance region to reach its maximum value in the fully developed region. Also it can be seen that, the velocity increased with increasing Reynolds number.

Fig. 5 shows the distribution of dimensionless axial velocity in \( y \) and \( z \) directions, respectively for \( Re = 100 \) and \( D_h = 117 \mu \text{m} \) at different locations along the \( x \)-axis (direction of flow). From this figure one can see the parabolic shape of the velocity, also the velocity increased along the flow direction until it reaches its maximum value in the fully developed flow region.

Fig. 6 represents the distribution of temperature for hot fluids, cold fluids and solid wall along the heat exchanger at \( Re = 100 \), and \( D_h = 117 \mu \text{m} \), \( t_s = 50 \mu \text{m} \) and \( k_s = 1 \). This figure indicates that, the temperature of cold fluid increased and

![Figure 2](image2.png)

**Figure 2** Schematic of separating wall.

![Figure 3](image3.png)

**Figure 3** Comparison of the dimensionless mean temperature of the hot and cold fluids for present model and data of Al-bakhit and Fakheri (2006).

![Figure 4](image4.png)

**Figure 4** Distribution of axial central velocity along flow direction for different values of Re.
the temperature of hot fluid decreased along the flow direction due to heat transfer from hot to the cold fluid and the temperature of separating solid wall is in between them.

To show the importance of axial heat conduction in a microchannel heat exchanger. Fig. 7 indicates the distribution of axial heat conduction flux and the flux of heat transferred between two fluids at Re = 100 and $t_s = 50 \mu m$, $K_r = 1$, $D_h = 117 \mu m$ along the heat exchanger (direction of flow). This figure shows that, the amount of heat conducted in the axial direction of the solid wall separating the fluids cannot be ignored due to its considerable value.

Fig. 8 shows the distribution of longitudinal axial heat conduction flux ($q_{0x}^*$) in the separating wall along the heat exchanger for different Reynolds numbers at $D_h = 117 \mu m$, $K_r = 1$ and $t_s = 50 \mu m$. The figure shows that, the value of $q_{0x}^*$ for all Reynolds number is high in the entrance region, due to the effect at entrance region the maximum heat transfer occurs in the entrance region and as a result a maximum value of axial heat conduction is created in this region. Also it can be seen from this figure that, the axial heat conduction flux increased with increasing Re. Due to increasing amount of heat transferred and increasing effect of entrance region with Re, the fraction of axial heat conduction is increased.

Fig. 9 shows the longitudinal distribution of the axial heat conduction flux in the separating wall ($q_{0x}^*$) along the heat exchanger for different hydraulic diameters at Re = 100, $K_r = 1$ and $t_s = 50 \mu m$. The figure shows that, the $q_{0x}^*$ for all
The hydraulic diameters is high in the entrance region as explained before. Also, the $q'_x$ decreased with increasing hydraulic diameter due to decreasing effect of wall thickness compared to the channel dimensions.

Fig. 10 illustrates the longitudinal distribution of the axial heat conduction in the separating wall along the heat exchanger for different values of wall thicknesses $t_s$ at $Re = 100$, $Kr = 1$ and $D_h = 117 \mu m$. From this figure one can see that, the value of axial heat conduction increased with increasing wall thickness due to an increase in the cross-sectional area in which the axial heat conduction transferred also due to increasing conduction resistance for heat transferred to the fluid and as a result increasing the fraction of heat transferred in the longitudinal direction (axial heat conduction). While Fig. 11 shows the decreasing of the axial heat conduction flux with increasing wall thickness due to an increase in the cross sectional area of wall.

Fig. 12 shows the longitudinal distribution of the axial heat conduction flux ($q'_x$) with axial distance $x$ for different thermal conductivity ratios $Kr$ at $Re = 100$, $t_s = 50 \mu m$ and $D_h = 117 \mu m$. For all values of thermal conductivity ratio, the results indicate that the $q'_x$ is high in the entrance region and decreased towards the fully developed region. Also, it can be seen that, the axial heat conduction flux increased with increasing $Kr$ up to $Kr = 10$, after this value the axial conduction becomes zero. This is due to that, the heat transferred from hot fluid to cold fluid increased with increasing $Kr$ up to $Kr = 10$, after $Kr = 10$, which is considered as an optimum value for $Kr$ in this case there is no increase in heat transfer. The numerical results of Al-bakhit and Fakheri (2005) for the parallel flow rectangular microchannel heat exchanger showed that more heat is transferred as the thermal conductivity ratio increased up to $Kr = 40$, after which increasing the thermal conductivity ratio will not enhance the heat transfer. They showed that this is because, the wall will behave as an infinitely conducting wall with negligible temperature gradient, and the heat transfer between the two fluids will be independent of the wall properties, depending only on the fluid conditions and channel geometry. The difference between the numerical results of Al-bakhit and Fakheri (2005) and present results is due to the difference in geometry. However, Fig. 13 may explain this behaviour (Hasan, 2009). As can be seen the dimensionless mean of wall temperature $T_{\omega} = (T_{\omega} - T_{\omega,ref})/(T_{h,ref} - T_{c,ref})$ with $x' = x/D_h$ varied with fluid temperature up to $Kr = 10$. After this value the $T_{\omega}$ is independent of fluid temperature and seen as a straight line. This means that, the gradient $\partial T_{\omega}/\partial x = 0$ and then $q'_x = 0$ after $Kr = 10$. 

**Figure 9** Longitudinal distribution of the axial heat conduction flux with axial distance $x$ for different hydraulic diameters.

**Figure 10** Longitudinal distribution of the axial heat conduction with axial distance $x$ for different wall thicknesses.

**Figure 11** Distribution of the axial heat conduction flux with axial distance $x$ for different wall thicknesses.

**Figure 12** Longitudinal variation of the axial heat conduction flux with axial distance $x$ for different values of $Kr$. 

**Figure 13** Longitudinal distribution of the axial heat conduction with axial distance $x$ for different thermal conductivity.
Fig. 14 shows the variation of the average axial heat conduction flux \( q_{\text{average}} \) with the number of channels for different Reynolds numbers for heat exchanger volume \( 2 \times 10^{-5} \text{ m}^3 \) at \( t_e = 50 \mu\text{m} \) and \( K_e = 1 \). From this figure it can be noted that, for all values of Reynolds number, the \( q_{\text{average}} \) increased with increasing number of channels, due to an increase in the heat...
transfer with increasing number of channels. Also, the figure shows that the $q_{average}$ increased with increasing Re, this is due to an increase in the axial heat conduction with Re.

Fig. 15 illustrates the variation of the microchannel heat exchanger effectiveness $\varepsilon$ with the number of channels for heat exchanger volume $2 \times 10^{-8} \text{ m}^3$ at $K_r = 1$ and $t_s = 50 \mu\text{m}$. For all values of Re, the figure clarifies that, the effectiveness increased with increasing number of channels. This is due to the amount of fluid that decreased when the channel volume decreased and then $\Delta T$ increased. On the other hand, the axial conduction in the solid wall also increased as shown in previous figures but the heat transferred between the two fluids is large enough to overcome the axial heat conduction. Also, the figure shows that, the effectiveness decreased with increasing Re, this is due to an increase in the velocity of fluid with Re and then $\Delta T$ decrease.

The variation of heat exchanger effectiveness to the average axial heat conduction ratio $\varepsilon/q_{average}$ with the number of channels for different values of Re is indicated in Fig. 16. From this figure it can be seen that, for all Reynolds number, the $\varepsilon/q_{average}$ increased with an increase in the number of channels due to increasing effectiveness with the number of channels in spite of increasing the axial heat conduction. Also this ratio decreased with increasing Re due to decreasing effectiveness with increasing Re.

Fig. 17 shows the variation of the average axial heat conduction flux $q_{average}$ with the number of channels for different thermal conductivity ratios for heat exchanger volume $2 \times 10^{-8} \text{ m}^3$ at Re = 200 and $t_s = 50 \mu\text{m}$. For all thermal conductivity ratios, the figure shows that, the $q_{average}$ increased with increasing number of channels due to the large amount of heat transfer that undergoes through the wall in this case up to $K_r = 10$, after this value the $q_{average}$ is equal to zero as explained before. Also, the figure shows that the $q_{average}$ at $K_r = 1$ is lower than for five and 10, due to that, there is more heat transfer in the axial direction at higher $K_r$.

Fig. 18 illustrates the variation of the heat exchanger effectiveness $\varepsilon$ with the number of channels for different values of $K_r$ for heat exchanger volume $2 \times 10^{-8} \text{ m}^3$ at Re = 200 and $t_s = 50 \mu\text{m}$. From this figure one can see that, for all values of $K_r$, the effectiveness increased with increasing $K_r$ due to more heat that is transferred from hot to cold fluid with increasing $K_r$. Also the effectiveness increased with increasing number of channels.

Fig. 19 shows the variation of the heat exchanger effectiveness to average axial heat conduction ratio $\varepsilon/q_{average}$ with the number of channels at different values of $K_r$ for heat exchanger volume $2 \times 10^{-8} \text{ m}^3$ at Re = 200 and $t_s = 50 \mu\text{m}$. From this figure it can be observed that, for all $K_r$ values, the $\varepsilon/q_{average}$ increased with increasing number of channels this is because the effectiveness increased with the number of channels in spite of the increase in axial heat conduction. Also, the $\varepsilon/q_{average}$ at $K_r = 10$ is larger than for one and five because the effectiveness increases with $K_r$.

4. Conclusions

From the results obtained the following conclusions can be drawn:

- The parameters that affect the axial heat conduction in an isosceles right triangular microchannel heat exchanger are: thermal conductivity ratio $K_r$, Reynolds number Re, hydraulic diameter $D_h$, channel volume and wall thickness $t_s$.
- The axial heat conduction increased with increasing $K_r$ up to $K_r = 10$ after this value the axial conduction is equal to zero.
- The axial heat conduction increased with increasing Re, $t_s$ and the number of channels but, decreased with increasing $D_h$ each separately.

References


