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INVESTIGATION OF AXIAL HEAT CONDUCTION FLUX IN RECTANGULAR PARALLEL FLOW MICROCHANNEL HEAT EXCHANGER WITH NO-SLIP FLOW

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ABSTRACT

In this paper the axial heat conduction in a rectangular microchannel heat exchanger is numerically investigated. The flow is taken as laminar, 3D, incompressible, single-phase, steady state flow. The behavior of axial heat conduction in the separating wall for different Reynolds number 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 in FLUENT code 16.0. Various parameters that can have effect on the axial heat conduction were investigated. In the result it is observed that, the axial heat conduction plays an important role at the entrance of the channel, velocities of hot fluid and cold fluid are increasing at the entrance of the channel as increases in Reynolds number.

KEYWORDS: Microchannel heat exchanger, No-slip, Reynolds number, Axial heat conduction flux, Parallel flow.

INTRODUCTION

Micro-channels are found in many systems where extremely efficient heat and mass transfer processes occur. Across the wall of the channel this transfer movement occurs, whereas the flow takes place through the cross-sectional area of the channel. The smaller channel dimension gives higher heat transferal performance although it is attended high pressure drop per unit length. The high flux heat distribution from high-speed microprocessors provided the focus for studies on heat transfer in microchannel. The higher volumetric heat transfer densities postulate advanced manufacturing techniques and lead to more complex multiplex designs. An optimum balance for each application leads to contrary channel dimensions. Heat transfer investigation in microchannel flow has drawn great attention due to its extensive applications in micro heat exchangers and micro fuel cells. Researchers have been dedicated in the investigation using theoretical analysis, numerical and experimental formulation.

NOMENCLATURE

H	Channel height, m	x	Axial coordinate, m/s
B	Channel base, m	x^*	Dimensionless axial distance, m/s
C_c	Heat capacity of cold fluid, W/K	y	Vertical coordinate, m/s
C_h	Heat capacity of hot fluid	z	Horizontal coordinate, m/s
C_p	Specific heat at constant pressure, kg/k	**GREEK SYMBOL**	
D_h	Hydraulic diameter, m	ε	Heat exchanger effectiveness
k	Thermal conductivity, W/mK	ρ	Density, kg/m^3
L	Length of the channel, m	μ	Dynamic viscosity, Ns/m^2
m	Mass flow rate, kg/s	**SUBSCRIPTS**	
p	Pressure, Pa	c	Cold fluid
q	Heat transfer rate, W	h	Hot fluid
q_{max}	Maximum heat transfer rate, W	i	Inlet
T	Temperature, K	max	Maximum value
v_i	Fluid y-component velocity, m/s	min	Minimum value
w_i	Fluid z-component velocity, m/s	o	Outlet
		s	Solid

However if the Peclet number is small in the flow, axial heat conduction is important in the fluid.

Liu and Lee.[1] The rapid development of modern electronics industry has necessitated effective cooling techniques which are capable of dissipating ultra-high heat flux of about $100W/cm^2$ from the highly integrated microelectronics systems to ensure a stable and optimal operation. Moharana et.al[2] studied axial heat conduction effect in microchannel flow using thermocouples with parallel channel design and reported that Non-linear fluid and surface temperature distributions and lower Nu number.

Jian et.al.[3] shows that microchannel heat exchangers are gives very high heat transfer area per unit volume over a conventional heat exchangers, it means the overall heat transfer area per unit volume can be greater than 100MW/m^3k. They show that the performance of heat exchanger is dependent on the mass flow rate and fluid properties. Wang and Shyu.[4] experimentally studied the effect of thermal conductivity of wall material and channel size in microchannel heat exchanger for hot and cold water loop. They concluded that wall material and channel size strongly influence the capability of heat transfer in microchannel heat exchanger. Ravigururajan et.al.[5] experimentally studied the performance characteristics of single phase flow parallel type microchannel heat exchanger using Refrigerant-134 as a fluid in experimental set up and concluded that the heat transfer coefficient increases in the thinning of a boundary layer in the narrow channels, with low thermal resistance.

Kroger[6] solved the governing equation for counter flow heat exchanger (two streams) with considering the effect of axial heat conduction in the channel, he used the one dimensional plate which separating the two fluids and showed the closed form solution for finding the effectiveness of a heat exchanger and also showed that effectiveness of the heat exchanger is largely dependent on the on the axial heat conduction in the separating wall. Yin and Bau.[7] study the effect of axial heat conduction on the performance of microchannel heat exchanger between parallel plates and circular pipe and showed that at the entrance region axial heat conduction plays an important role.

Stief et.al[9] investigated in the microchannel heat exchanger with the effect of solid thermal conductivity and concluded that heat transfer efficiency improve with reduction of wall material thermal conductivity due to influence of axial heat conduction in the separating wall. Neti and Eichhorn[8] used Finite difference method to solve hydro dynamically and thermally developing flow in a square duct. They assumed that in the axial direction pressure gradient linearly varied and neglect the axial momentum and energy diffusion. Venkatarathnam and Narayanan[10] also concluded that heat exchanger performance largely dependent on the heat conduction that takes place through the wall of a heat exchanger used in a miniature Refrigerator.

Al-bakhit and Fakheri[11] numerically investigated the parallel flow microchannel heat exchanger with rectangular ducts. They reported that effectiveness of a heat exchanger will be independent of the thermal conductivity of wall and high conductivity material will not effect the effectiveness of heat exchanger. Al-bakhit and Fakheri[12] investigated in rectangular ducts parallel flow microchannel heat exchanger and showed that the overall heat transfer coefficient is rapidly changed along the length of a heat exchanger in the order of 0.03 for Greatz number. Hasan.[13] made numerical investigation for the study of counter flow microchannel heat exchanger with different geometries and fluids. He found that the existence of axial heat conduction leads to reduction in the effectiveness of counter flow microchannel heat exchanger with square shaped channel.

THEORY AND PROBLEM DISCUSSION

Fig.1 represents the unit cells of microchannel array. The schematic diagram shows the two isosceles right triangular channels and separating solid wall between them which separates the hot and cold fluid and play as a role of interfaces between the fluids.

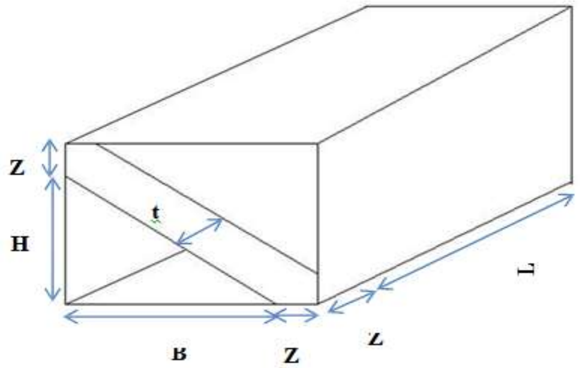

Fig. 1 schematic diagram of parallel flow microchannel heat exchanger.

3cm (L) is the channel length, 195µm (B) is a base of a channel, 195µm (H) is the height of the channel, 70µm (t) is the thickness of the solid wall, Z is projected distance and hydraulic diameter (Dh) 114µm. The cold fluid enters in the upper channel and the hot fluid enters the lower channel with uniform velocity and uniform temperature. The numerical procedures are based on the physical and geometrical assumption of the channel. The following assumptions are taken for the present model:

- The fluids are incompressible (water is used as a working fluid).
- The flow is laminar and steady.
- The fluid is a continuous media, the Knudsen number is small taking as no-slip condition.
- There is no heat generation to/from the ambient.
- Three dimensional flow.
- The pressure gradient is an axial direction only.

Based on the above assumptions, the governing equations and boundary conditions in Cartesian coordinates are written as below:

x- Momentum equation

$$\rho_o\,(u_o\frac{\partial u_o}{\partial x} + v_o\frac{\partial u_o}{\partial y} + w_o\frac{\partial u_o}{\partial z}) = -\frac{\partial p_c}{\partial x} + \mu_o\,(\frac{\partial^2 u_o}{\partial x^2} + v_o\frac{\partial^2 u_o}{\partial y^2} + w_o\frac{\partial^2 u_o}{\partial z^2}) \qquad (1)$$

y- Momentum equation

$$\rho_o(u_o\frac{\partial v_o}{\partial x} + v_o\frac{\partial v_o}{\partial y} + w_o\frac{\partial v_i}{\partial z}) = \mu_o\,(\frac{\partial^2 v_o}{\partial x^2} + v_o\frac{\partial^2 v_o}{\partial y^2} + w_o\frac{\partial^2 v_o}{\partial z^2}) \qquad (2)$$

z- Momentum equation

$$\rho_o\,(u_o\frac{\partial w_o}{\partial x} + v_o\frac{\partial w_o}{\partial y} + w_o\frac{\partial w_o}{\partial z}) = \mu_o\,(\frac{\partial^2 w_o}{\partial x^2} + v_o\frac{\partial^2 w_o}{\partial y^2} + w_o\frac{\partial^2 w_o}{\partial z^2}) \qquad (3)$$

Energy equation for fluid

$$\rho_o\, c_{p_o}(u_o\frac{\partial T_o}{\partial x} + v_o\frac{\partial T_o}{\partial y} + w_o\frac{\partial T_o}{\partial z}) = k_o\,(\frac{\partial^2 T_o}{\partial x^2} + v_o\frac{\partial^2 T_o}{\partial y^2} + w_o\frac{\partial^2 T_o}{\partial z^2}) \qquad (4)$$

Conduction equation for solid

$$\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} = 0 \qquad (5)$$

Continuity equation can be expressed as:

$$\frac{\partial u_o}{\partial x} + \frac{\partial v_o}{\partial y} + \frac{\partial w_o}{\partial z} = 0 \qquad (6)$$

Where o=h and c refers to the hot and cold fluids, respectively. As mentioned in the assumptions the pressure gradients in y and z directions are minimal.

The boundary conditions for lower channel, upper channel and solid wall is adopted from the literature of the Hasan.[13]

- 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; $0 \le z \le D$

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

At y = (D – z) $\tan(45)$, $0 \le z \le D$

$u_h = v_h = w_h = 0$

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

At z = 0, 0 ≤ y ≤ D

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

At y = (D – y) $\tan(45)$, 0 ≤ y ≤ D

$u_h = v_h = w_h = 0$

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

- Upper channel (cold fluid)

At x = 0

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

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 = (D – z) $\tan(45) + 2S$, S ≤ z ≤ (D + S)

$u_c = v_c = w_c = 0$

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

At z = D + S

$u_c = v_c = w_c = 0$

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

- 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 ≤ z ≤ D

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

At z = y $\tan(45)$, S ≤ y ≤ (D + S)

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

At z = 0; D ≤ y ≤ (D + S)

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

At z = (D + S), 0≤ y ≤ S

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

At y = (D + S), 0 ≤ z ≤ S

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

At y = 0, D ≤ z ≤ (D + S)

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

Then, the parameters such as the axial heat conduction q'' in the separating wall, the amount of heat transferred between two fluids and the effectiveness ε which is the ratio of actual heat transfer to the maximum possible heat that can be transferred can be calculated from Ashman and Kandlikar.[14]:

$$\varepsilon = q_{\text{actual}} / q_{\text{max possible}} \qquad (7)$$

$$q_{\text{actual}} = \dot{m}_c c_c (T_{c,out} - T_{c,in}) = \dot{m}_h c_h (T_{h,in} - T_{c,out}) \qquad (8)$$

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

$$C_c = \dot{m}_c c_c \text{ and } C_h = \dot{m}_h c_h \qquad (9)$$

For $C_h < C_c$,

$$q_{\max} = C_h (T_{h,in} - T_{c,in}) \qquad (10)$$

Otherwise

$$q_{\max} = C_c (T_{h,in} - T_{c,in}) \qquad (11)$$

Thus

$$q_{\max} = C_{min} (T_{h,in} - T_{c,in}) \qquad (12)$$

Then the effectiveness is

$$\varepsilon = \frac{C_c(T_{c,out} - T_{c,in})}{C_{min}(T_{h,in} - T_{c,in})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{min}(T_{h,in} - T_{c,in})} \qquad (13)$$

For the axial heat conduction in section x

$$Q_x'' = \left(k_s \frac{T_{s,i} - T_{s,i+1}}{\Delta x} \right)_x \qquad (14)$$

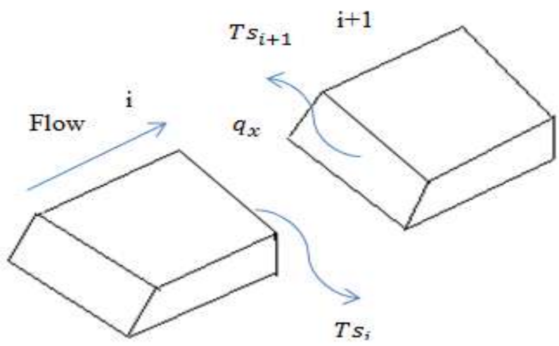

Fig. 2 Schematic of separating wall.

Where, $T_{s,i}$ is the average solid temperature in certain sections and $T_{s,i+1}$at the next section as shown in Fig. 2.

RESULTS AND DISCUSSION

Axial heat conduction on a separating wall in a microchannel heat exchanger is studied for different Reynolds numbers. Water is used as a working substance for both channels in the microchannel heat exchanger. The inlet temperature of cold fluid is 300^{o}K, temperature of hot fluid at inlet 346^{o}K. The processing of developing velocity flow field was simulated in the commercial accessible CFD package of ANSYS FLUENT 16.0.

Comparisons of temperatures for hot and cold fluids with length of the channel as shown in figure 3.

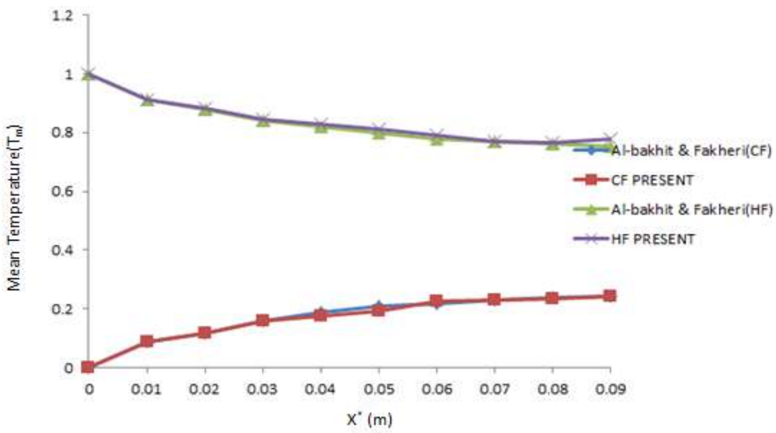

Fig. 3 comparison of mean temperatures for cold and hot fluid along length of present model and data of Al-bakhit and Fakheri.[12]

The present model for the hot, cold fluids is validated with those of Al-bakhit and Fakheri[12] and are found to be in good agreement.

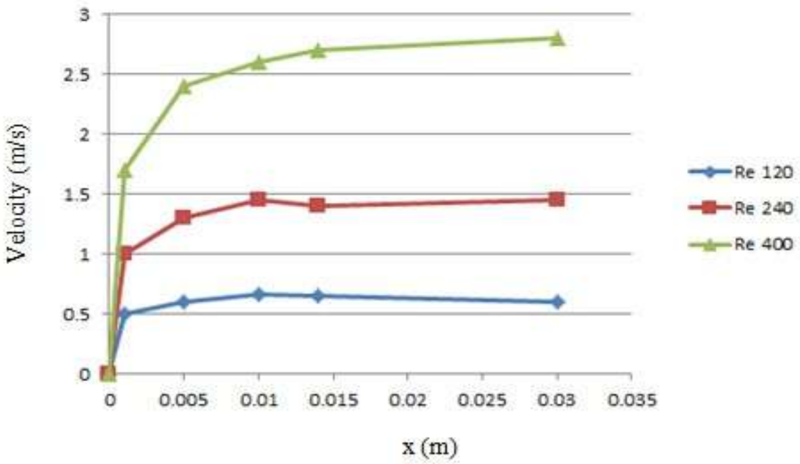

Fig. 4 velocity along the length of the channel for different values of Reynolds number.

The variations of center axial velocities with length of the channel are calculated. It is observed from the obtained result at the entrance of the channel the velocities are increasing as the Reynolds number increasing and reaches its maximum value as shown in figure 4.

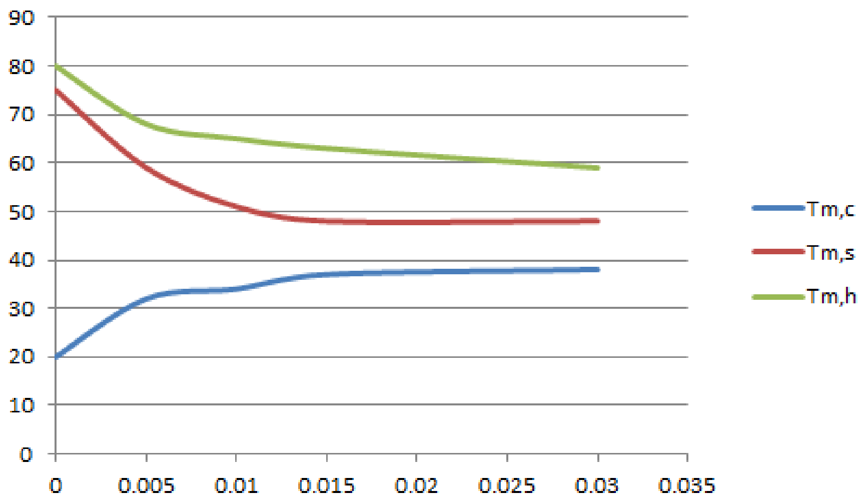

Fig.5 Temperatures of cold fluid, separating wall, hot fluid along the length of the channel.

Figure 5 disputes the heat transfer process in the channel from the result it is noted that the temperature of cold fluid is growing and the temperature of hot fluid is diminishing along the length of the channel. So it can be established that the heat transfer takes place inside a channel from hot fluid to cold fluid and the temperature of solid wall is among them.

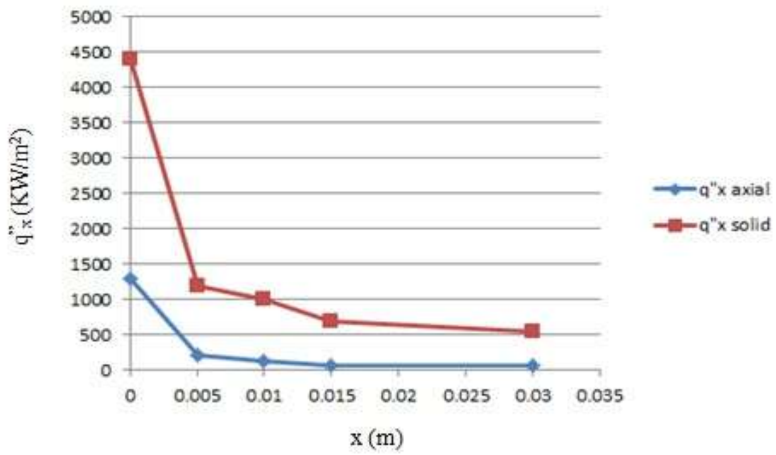

Fig. 6 axial heat conducted and heat conduction between fluids along the length of a channel.

Figure 6 indications the prominence of axial heat conduction in a microchannel heat exchanger and designates the scattering of flux of heat transferred and axial heat conduction flux between two fluids along the length of a channel in the direction of flow. From the outcome it is prominent that, the amount of heat conducted in the axial direction of the solid wall separating the fluids cannot be ignored due to its significant value.

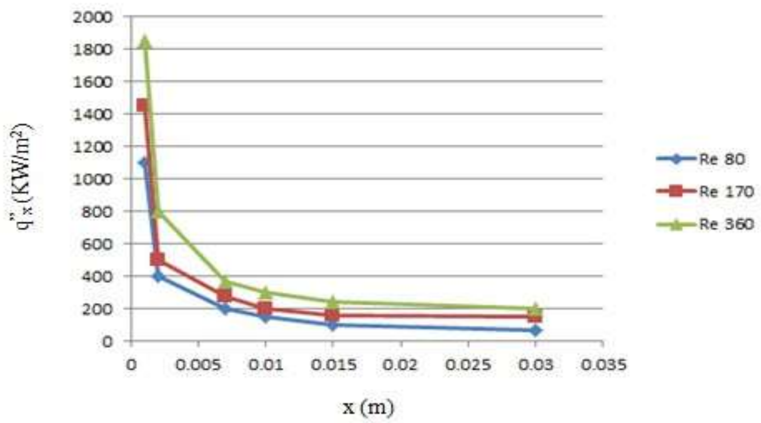

Fig. 7 variation of axial heat conduction for different values of Reynolds along the flow direction.

Figure 7 demonstrate the distribution of longitudinal axial heat conduction flux in the separating wall along the heat exchanger for different Reynolds numbers. The result indicate for all Reynolds number axial heat conduction flux is high at the entrance region, it means the maximum heat transfer occurs at the entrance region and as a result a maximum value of axial heat conduction is obtained in this region. Also it can be seen the axial heat conduction flux improved with growing the value of Reynolds number.

CONCLUSIONS

From the acquired results we resolved that velocities of hot and cold fluids are increasing at the entrance region of the channel and achieves its maximum value. It also be eminent that for different values of Reynolds number the velocities are growing. The heat transfer occurs from hot fluid to the cold fluid due to the separating wall between them and the significant temperature of the separating wall. The increasing amount of heat transfer and increasing effect of Reynolds the part of axial heat conduction increased.

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