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Entrance and Wall Conduction Effects in Parallel Flow Heat Exchangers

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ABSTRACT

Heat exchanger analysis tools such as F correction factor charts and ε-NTU relations assume that the overall heat transfer coefficient is constant across the heat exchanger. The short lengths and the comparatively thick walls in microchannel heat exchangers preclude the existence of thermally fully developed flow over a large portion of the heat exchanger and increase the significance of heat conduction in the wall. In this study, a parallel flow heat exchanger is simulated numerically to determine the impact of entrance effects and conduction heat transfer through the walls on the performance of the heat exchanger, including the number of transfer units and the effectiveness. The impact of wall conduction on the heat transfer is studied for different wall thickness and thermal conductivity. It is shown that entrance and wall effects can be incorporated in the longitudinally averaged number of transfer units and accounting for them significantly changes the heat exchanger size for to a given effectiveness.

KEYWORDS

Microchannel, Heat Exchangers, Conjugate, Developing flow, Effectiveness, NTU, Parallel Flow

NOMENCLATURE

a	ducts width, m
A	Heat Transfer Area, m^2
A_c	Duct cross sectional Area, m^2
α_1	lower duct aspect ratio ($=b_1/a$)
α_2	upper duct aspect ratio ($=b_2/a$)
b_1	lower duct height, m
b_2	upper duct height, m
C	$C = \dot{m}c_p$, Fluid capacity, W/K
D_h	hydraulic diameter $D_h = \dfrac{4A_c}{p}$
h	convective heat transfer coefficient, $W/m^2 k$
k	thermal conductivity, w/m k
K_r	$K_r = \dfrac{k_s}{k_1}$
L	duct length, m
N_1	$N_1 = \dfrac{k_1 a}{k_s t}$ nond. hot fluid/solid conductivity
N_2	$N_2 = \dfrac{k_2 a}{k_s t}$ nond. cold fluid/solid conductivity
Nu	Nusselt number
P_i^*	$P_i^* = \dfrac{P_i}{\rho_i u_{1,in}^2}$ nondim. pressure
p	heated perimeter (2a), m
Pr	Prandtl number ($= c_p\mu/k$) nondim
Pe	Peclet number, $Pe_i = \dfrac{\rho_i u_{1\infty} a}{\mu_i}\dfrac{\mu_i c_{p\,i}}{k_i} = \dfrac{u_{1\infty} a}{\alpha_i}$
Re_i	Reynolds number, $\mathrm{Re}_i = \dfrac{\rho_i u_{1\infty} a}{\mu_i}$
T	fluid temperature, K
$T_{1\infty}$	fluid temperature at inlet of the lower duct, K
$T_{2\infty}$	fluid temperature at inlet of the upper duct, K
t	conducting wall thickness, m
T_i^*	$T_i^* = \dfrac{T_i - T_{2,in}}{T_{1,in} - T_{2,in}}$, nondim.
x*	$x^* = \dfrac{x}{a}$ nondim. x coordinates
x^+	Graetz Number, $x^+ = \dfrac{x}{D_h Pe}$, nondim.
y*	$y^* = \dfrac{y}{a}$ Nondim. y coordinates
z*	$z^* = \dfrac{z}{a}$ Nondim. z coordinates

U overall heat transfer coefficient, W/m^2 k

u*$_i$ $u_i^* = \dfrac{u_i}{u_{1,in}}$ nondim. x-component of velocity

v*$_i$ $v_i^* = \dfrac{v_i}{u_{1,in}}$. nondim. y-component of velocity

w*$_i$ $w_i^* = \dfrac{w_i}{u_{1,in}}$. nondim. z-component of velocity

Greek Symbols

ρ density, kg/m^3

α thermal diffusivity, m^2/s

μ dynamic viscosity, kg/m s

ν Kinematics viscosity, m^2/s

Δ* $\Delta^* = \dfrac{T_{m,H} - T_{m,C}}{T_{1,in} - T_{2,in}}$, Nondim.

Subscripts and Superscripts

*	nondimensional quantity
1	lower duct (hot fluid)
2	upper duct (cold fluid)
m	mean
H	Hot
C	Cold
s	refers to solid
in	refers to inlet

1. INTRODUCTION

Full numerical solution of heat exchangers are computationally prohibitive because the flow and temperature fields must be determined simultaneously in at least two fluids and the solid separating the two. The heat exchanger analysis and design are, therefore, based on solutions obtained for flows in individual ducts subject to isothermal or uniform heat flux boundary conditions and experimental measurements. The F correction factor charts, or the ε-NTU relations, widely used for heat exchanger analysis, are based on the assumption that the overall heat transfer coefficient is constant.

In microchannel heat exchangers the area of the solid's cross section perpendicular to the direction of the flow may be as large as the cross sectional area available for the flow. For these heat exchangers, thermally fully developed flow may not exist over a large portion of the heat exchanger due to the comparatively thick walls through which heat is conducted, and the short length of the heat exchanger. These necessitate examination of the detailed flow fields to determine the validity of the assumptions typically made, including constant overall heat transfer coefficient.

Yin and Bau (1992) studied flow between infinite parallel plates and circular pipes to study the effect of axial conduction on the performance of micro-channel 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 axial conduction plays an important role at the entrance region, were local values Nusselt Number are higher than when axial conduction is ignored for uniform wall temperature. 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 found that the channel size and wall material have strong influence on the heat transfer capability of a micro heat exchanger.

The effect of solid conduction on the heat exchanger performance has been studied extensively. 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 influence of axial heat conduction in the channel walls.

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 that the temperature of the fluid to be uniform at any flow cross section.

Ravigururajan et. al. (1996) built an experimental set up to study the thermal performance characteristics of single-phase flow in parallel micro heat exchanger, Refrigerant-134 was used as an experimental fluid. He attributed the increase in heat transfer coefficient to the thinning of the boundary layer in the narrow channels, which lowers thermal resistance. Davis and Gill (1969) included heat conduction in the wall for Poiseuille-Couette flow, where heat is transferred through a heating zone in a stationary wall of finite thickness, and the heat flux is uniformly specified at the outer surface of the heating zone. They concluded that the interface temperature distributions and local Nusselt number distributions were affected by the flow conditions, the wall thickness, the ratio of thermal conductivity of the wall to the fluid, and the width of the duct (aspect ratio). Their experimental observations also pointed out that wall conduction was significant in the heat transfer phenomena.

Neti and Eichhorn (1983) used the finite difference method to solve for hydrodynamically and thermally developing flow in a square duct. They neglected the axial momentum and energy diffusion and assumed that the pressure gradient varied linearly in the axial direction. They also neglected the wall conduction, presenting results for the centerline velocities, pressure drop and Nusslet number for the case of Pr = 6.0 for an isothermal duct.

Aparaecido and Cotta (1990) studied the thermally developing flow inside rectangular ducts analytically by extending the generalized integral transform technique to solve the energy equation for a wide range of aspect ratios. Their results for local and average Nusselt number in the entrance region are used for validation of numerical solutions.

Fakheri et. al. (1994) numerically investigated the hydrodynamically and thermally developing flows in rectangular ducts with conducting walls. Based on their numerical solution, they came up with the following correlation $T_m = 1 - e^{-9.8x^{0.819}a^{0.524}}$ which approximates the mean fluid temperature in the entrance region of an isothermal rectangular duct for $x^+ > 0.01$ to within 10 %. They showed the significance of conduction on mean fluid temperature and Nusselt numbers. Chandurptha, et. al. (1977) numerically solved for laminar heat transfer of a Newtonian and non-Newtonian fluid in the thermal entrance region of a square duct for different boundary conditions. Shah and London (1978) compiled a comprehensive solution for laminar forced convection flow heat transfer in wide range of duct sizes and shapes.

Al-Bakhit and Fakheri (2004) considered the problem of simultaneously developing flows in parallel rectangular ducts including conduction through the solid boundaries. In that study, the authors examined the impact of the inclusion of the wall conduction and entrance effects on the performance of the heat exchanger. The wall was assumed to be thin, so that the wall temperature variation in the z direction can be neglected, but the heat conduction in the z direction was included (thin fin assumption). It was shown that there is significant change in the overall heat transfer coefficient in the developing region and that the three-dimensional heat transfer in the heat exchanger wall must be included in the analysis. The present study extends this work to thick wall, encountered in microchannels, where the z variation of the temperature in the wall separating the two channels is included.

2. ANALYSIS

In this study, the flow field in a parallel flow heat exchanger is numerically simulated to determine the impact of different flow parameters on heat transfer and determine the accuracy of the assumption of constant overall heat transfer coefficient. The analysis is for developing flow in ducts of a parallel flow heat exchanger.

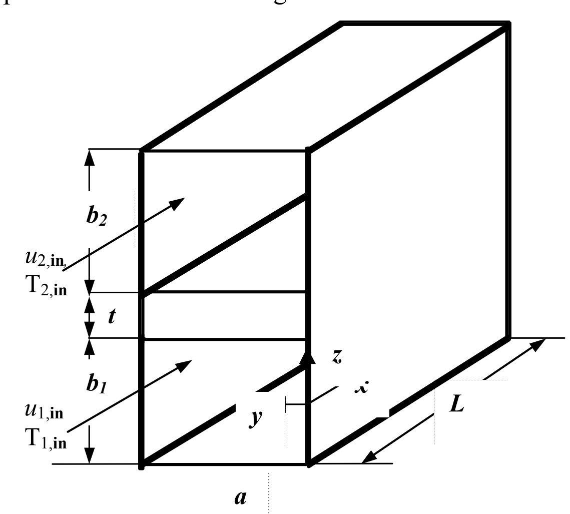


Figure 1 Three-dimensional sketch of the rectangular duct parallel flow heat exchanger

In the configuration shown in Figure 1, the hot fluid enters the lower ducts with a uniform velocity $u_{1,in}$ and uniform temperature $T_{1,in}$ while the cold fluid enters the ducts in the upper row at $u_{2,in}$, and $T_{2,in}$. Heat is transferred from the hot fluid to the cold fluid through the channel walls that have a thickness t. Although the hot and cold fluid ducts may have different aspect ratios, the results presented here are for the cases where both ducts have the same aspect ratio.

The goal of the investigation is to numerically solve the three-dimensional heat transfer for thermally developing laminar flows in the two parallel rectangular channels, and to look into parameters that affect the heat transfer between the two fluids.

For laminar, 3-D, steady, incompressible flow, the nondimensional governing equations become

$$\frac{\partial u_i *}{\partial x *} + \frac{\partial v_i *}{\partial y *} + \frac{\partial w_i *}{\partial z *} = 0 \tag{1}$$

$$u_i^* \frac{\partial u_i^*}{\partial x *} + v_i^* \frac{\partial u_i^*}{\partial y *} + w_i^* \frac{\partial u_i^*}{\partial z *} = -\frac{\partial P_i^*}{\partial x *} + \frac{1}{\mathrm{Re}_i} \nabla^2 u_i^* \tag{2}$$

$$u_i^* \frac{\partial v_i^*}{\partial x *} + v_i^* \frac{\partial v_i^*}{\partial y *} + w_i^* \frac{\partial v_i^*}{\partial z *} = -\frac{\partial P_i^*}{\partial y *} + \frac{1}{\mathrm{Re}_i} \nabla^2 v_i^* \tag{3}$$

$$u_i^* \frac{\partial w_i^*}{\partial x *} + v_i^* \frac{\partial w_i^*}{\partial y *} + w_i^* \frac{\partial w_i^*}{\partial z *} = -\frac{\partial P_i^*}{\partial y *} + \frac{1}{\mathrm{Re}_i} \nabla^2 w_i^* \tag{4}$$

$$u_i^* \frac{\partial T_i^*}{\partial x *} + v_i^* \frac{\partial T_i^*}{\partial y *} + w_i^* \frac{\partial T_i^*}{\partial z *} = \frac{1}{Pe_i} \nabla^2 T_i^* \tag{5}$$

where i=1 and 2 refer to the lower and upper (hot and cold) channels respectively. The three-dimensional steady state conduction equation through the solid is

$$\nabla^2 T_s^* = 0. \tag{6}$$

Nondimensional boundary conditions are

For $z* < b_1 *$

$$x* = 0 \qquad u* = 1, v* = w* = 0, T* = 1 \tag{7}$$

$$x* = L* = \frac{L}{a} \qquad \frac{\partial u *}{\partial x *} = \frac{\partial v *}{\partial x *} = \frac{\partial w *}{\partial x *} = \frac{\partial T *}{\partial x *} = 0 \tag{8}$$

$$y* = 0 \qquad u* = v* = w* = \frac{\partial T *}{\partial y *} = 0 \tag{9}$$

$$y* = \frac{a}{2} \qquad \frac{\partial u *}{\partial y *} = \frac{\partial v *}{\partial y *} = \frac{\partial w *}{\partial y *} = \frac{\partial T *}{\partial y *} = 0 \tag{10}$$

$$z* = 0, \qquad \frac{\partial u *}{\partial z *} = \frac{\partial v *}{\partial z *} = \frac{\partial w *}{\partial z *} = \frac{\partial T *}{\partial z *} = 0 \tag{11}$$

For $b_1 * + t* < z* < b*_1 + b*_2 + t*$

$$x* = 0 \qquad u* = \frac{u_{2,in}}{u_{1,in}}, v* = w* = 0, T* = 0 \tag{12}$$

$$x* = L* = \frac{L}{a} \qquad \frac{\partial u *}{\partial x *} = \frac{\partial v *}{\partial x *} = \frac{\partial w *}{\partial x *} = \frac{\partial T *}{\partial x *} = 0 \tag{13}$$

$$y* = 0 \qquad u* = v* = w* = \frac{\partial T *}{\partial y *} = 0 \tag{14}$$

$$y* = \frac{a}{2} \qquad \frac{\partial u *}{\partial y *} = \frac{\partial v *}{\partial y *} = \frac{\partial w *}{\partial y *} = \frac{\partial T *}{\partial y *} = 0 \tag{15}$$

$$z^* = \frac{b_1^* + b_2^*}{2} + t^*, \quad \frac{\partial u^*}{\partial z^*} = \frac{\partial v^*}{\partial z^*} = \frac{\partial w^*}{\partial z^*} = \frac{\partial T^*}{\partial z^*} = 0 \quad (16)$$

At the solid liquid boundaries

$$z^* = \frac{b_1^*}{2} \qquad \frac{\partial T^*}{\partial z^*} = \frac{k_s}{k_1} \frac{\partial T_s^*}{\partial z^*} \quad (17)$$

$$z^* = \frac{b_1^*}{2} + t^* \qquad \frac{\partial T^*}{\partial z^*} = \frac{k_s}{k_2} \frac{\partial T_s^*}{\partial z^*} \quad (18)$$

Once the temperature distribution is determined, the overall heat transfer coefficient for a parallel flow heat exchanger can be obtained by doing an elemental energy balance at a given cross section to get

$$dq = U(T_H - T_C) 2a dx = C_C dT_C = -C_H dT_H \quad (19)$$

then

$$d(T_H - T_C) = -dq \left[\frac{1}{C_H} + \frac{1}{C_C} \right] \quad (20)$$

which is then simplified to

$$d \ln(T_H^* - T_C^*) = -\frac{1 + C_r}{C_{\min}} U p dx \quad (21)$$

Where p is the circumference of the hot duct through which heat is transferred (2a in this case) and $C_r = \frac{C_{\min}}{C_{\max}}$ is the ratio of the thermal capacities of the two fluids. The above equation can be rearranged into

$$\frac{d \ln(T_H^* - T_C^*)}{dx} = -(1 + C_r) \frac{NTU_x}{x} \quad (22)$$

$$\frac{d \ln(T_H^* - T_C^*)}{dx^+} = -(1 + C_r) \frac{NTU_{x^+}}{x^+} \quad (23)$$

Solving for the local value of the Number of Transfer Units, NTU_{x^+}

$$NTU_{x^+} = -\frac{x^+}{1 + C_r} \frac{d \ln(T_H^* - T_C^*)}{dx^+} \quad (24)$$

The longitudinally averaged value of the NTU is defined in terms of the average value of the overall heat transfer coefficient from the entrance of the heat exchanger to any location x from the inlet

$$NTU_{avg,x} = \frac{\overline{U} A}{C_{\min}}$$

Where the average overall heat transfer coefficient is defined as

$$\overline{U} = \frac{1}{x} \int_0^x U dx \quad (25)$$

Integrating Eq.(9)

$$\ln(T_H^* - T_C^*) = -\frac{1 + C_r}{C_{\min}} p \int_0^x U dx = -\frac{1 + C_r}{C_{\min}} p x \frac{1}{x} \int_0^x U dx = -\frac{1 + C_r}{C_{\min}} A \overline{U} \quad (26)$$

Which can be solved for

$$NTU_{avg,x} = -\frac{1}{1 + C_r} \ln(T_H^* - T_C^*) \quad (27)$$

Therefore, if the variation of the mean temperature of the cold and hot fluids are known, the local value of the number of Transfer Units can be determined from Eq. (12) and the average value from the inlet of the heat exchanger to any location x from Eq. (15).

The performance of heat exchangers is characterized by the concept of the heat exchanger effectiveness which is the ratio of the actual heat transfer through the heat exchanger to the maximum possible heat that can be transferred

$$\varepsilon = \frac{q}{q_{\max}} \quad (28)$$

$$\varepsilon = \frac{C_H (T_{H,in} - T_H)}{C_{\min} (T_{H,in} - T_H)} = \frac{C_C (T_C - T_{C,in})}{C_{\min} (T_{H,in} - T_H)} \quad (29)$$

since

$$C_H (T_{H,in} - T_H) = C_C (T_C - T_{C,in}) \quad (30)$$

Then Eqs. (18) and (15) can be rearranged

$$\frac{C_c}{C_H} T_C^* + T_H^* = 1 \quad (31)$$

$$T_H^* - T_C^* = e^{-(1 + C_r) NTU_{avg,x}} \quad (32)$$

To find the relation between the Number of Transfer Units and the heat exchanger effectiveness, when the heat transfer coefficient changes along the heat exchanger, we consider two cases,

a. C_{min}=C_c

Then from Eq. (17) (33)

$$\varepsilon = \frac{T_{C,out} - T_{C,in}}{T_{H,in} - T_{C,in}} = T_C^* \quad (34)$$

and from Eq. (19)

$$C_r T_C^* + T_H^* = 1 \quad (35)$$

Substituting these back into Eq. (20) and simplifying results in

$$\varepsilon = \frac{1 - e^{-(1 + C_r) NTU_{avg,x}}}{1 + C_r} \quad (36)$$

b. C_{min}=C_h

Then from Eq. (17) (37)

$$\varepsilon = \frac{T_{H,in} - T_H}{T_{H,in} - T_{C,in}} = 1 - T_H^* \tag{38}$$

and from Eq. (19)

$$\frac{1}{C_r} T_C^* + T_H^* = 1 \tag{39}$$

Substituting these back into Eq. (20) and simplifying results in

$$\varepsilon = \frac{1 - e^{-(1+C_r)NTU_{avg,x}}}{1 + C_r} \tag{40}$$

As is shown, Eq. (28) is the same as Eq. (24) and interestingly they both are the same as the expression given for the relation between ε and NTU for a parallel flow heat exchanger when the overall heat transfer coefficient is assumed constant. This shows that the same relation between effectiveness and Number of Transfer Units holds when the NTU used in the relationship is the longitudinally averaged value of the Number of Transfer Units for the length of the heat exchanger. This number, of $NTU_{avg,x}$ given by Eq. (15), accounts for the developing effects and heat conduction through the walls.

3. RESULTS

The governing equations are solved numerically using a hybrid approach, where the momentum equation is solved using Fluent 6.1 and the velocity distribution is then inputted into a code developed for solving the energy equation for the conjugate heat transfer using the finite difference method. The details of the numerical solution are given in (Al-Bakhit, H. and Fakheri, A., 2004; Al-Bakhit, H., 2004).

The numerical grid varied with the flow conditions, but was typically composed of a grid about 100 nodes in the x direction, about 50 nodes in the y and z directions in each channel, and depending on the wall conductivity 10 to 150 nodes in the z direction in the wall separating the two ducts. Extensive verifications with the available results were made (Al-Bakhit, H. and Fakheri, A., 2004) to ensure the accuracy of the solutions. The nondimensional velocity field was calculated for various aspect ratios and Reynolds numbers. For different aspect ratios, the calculated values of the centerline velocity in the fully developed region were found to be within 0.2%. of the published results of by Shah and London (1978).

The variation of the local value of the mean temperatures of hot and cold fluid for different Reynolds number is shown in Figure 2. The results are for a balanced flow heat exchanger with square channels, made of low thermally conducting walls (K_r=1). As can be seen, except for the very low Peclet number of 7 (blue line), the mean temperature variation is independent of the Reynolds number, when the results are expressed in terms of Graetz number. Also, as shown before, the mean temperature of the cold fluid is the same as the heat exchanger effectiveness, and therefore, for a given wall conductivity and Graetz number, the effectiveness is the same regardless of the value of the Reynolds number. Figure 3 shows the temperature variation of the cold and hot fluids along the heat exchanger for a balanced flow ($C_r = 1$). The results are for different aspect ratio ducts when the wall thermal conductivity to fluid conductivity is high (K_r =100).

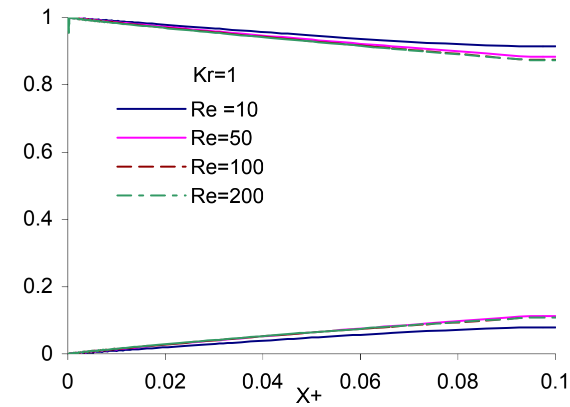


Figure 2 Longitudinal variation of the hot and cold fluid mean temperature

For these cases, the temperature variation in the duct wall can be neglected and the wall will be at a uniform temperature, and in the limit of an infinitely long heat exchanger from Eq. (20)

$$T_C^* = T_H^* = T_w^* \tag{41}$$

then from Eq. (19)

$$T_w^* = \frac{1}{1 + \frac{C_c}{C_H}} \tag{42}$$

which for a balanced flow heat exchanger, having thin walls and/or walls made of high thermal conductivity material $T_w^* = 0.5$.

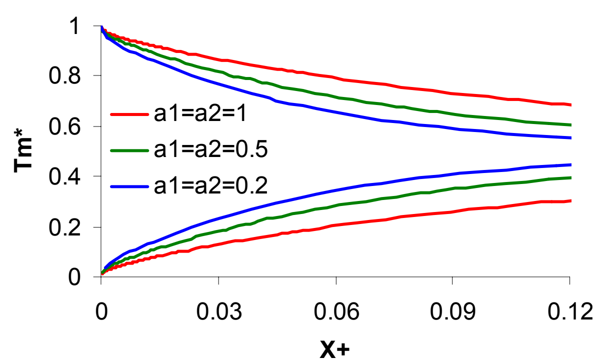
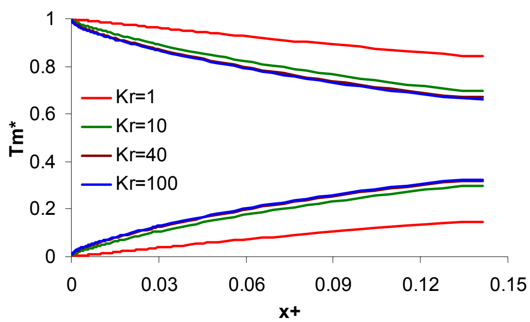


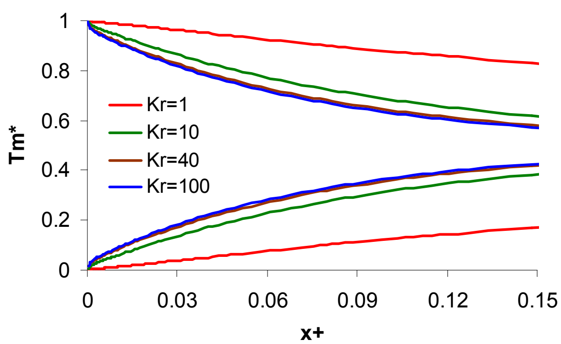
Figure 3 Longitudinal variations of the mean temperatures of hot and cold fluids for K_r =100

Figure 3 shows the longitudinal variation of the mean temperature for the different aspect ratios for a heat exchanger having thin walls and/or very high thermal conductivity, where the temperature variation in the wall is small. As can be seen the temperatures change monotonically and asymptote to the value of 0.5 for a very long heat exchanger. It can also be seen that that for a given Reynolds number, the heat exchanger effectiveness increases significantly with decreasing the aspect ratio.

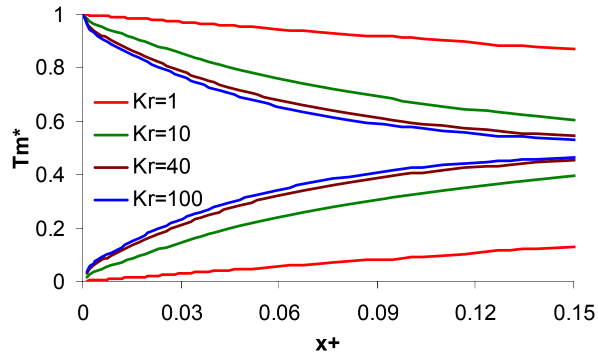
For the case of liquids flowing in low thermal conductivity ducts such as ceramic heat exchangers and or when the wall thickness to the duct width ratio is of the order of one, as in the cases of microchannel heat exchanger then conduction in the solid must be taken into consideration.



α=1.



α=0.5



α=0.2.

Figure 4 Longitudinal variations of the mean temperatures for rectangular ducts

Figures 4 show the longitudinal variation of the mean temperature for the different aspect ratios and different wall thermal conductivities. In all the cases shown, the heat exchanger is a balanced flow one. The impact of thermal conductivity of the solid wall on the heat exchanger performance has been investigated by changing K_r which is the ratio of the solid to fluid conductivity. For a given aspect ratio, more heat is transferred as the wall thermal conductivity is increased up to a certain limit (K_r= 40), after which increasing the thermal conductivity will not enhance the heat transfer. Beyond this value, 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 fluids conditions and the heat exchanger geometry

As shown before, the variation of the heat transfer coefficient at the entrance region of the heat exchanger or the impact of conduction heat transfer through the walls can be studied by how these effects impact the Number of Transfer Units. Figure 5 shows the variation of the average NTU along the heat exchanger for a wide range of Peclet numbers and thermal conductivities. As can be seen from Figure 5, the solution is only a function of conductivity ratio and independent of Reynolds number. For a given heat exchanger length, (area) increasing the wall conductivity, increases NTU and therefore the heat exchanger effectiveness, up to a conductivity ratio of about 40, beyond which the wall material properties do not appear to impact the performance of the heat exchanger. Also, for the same material, and heat exchanger size, increasing the Reynolds number, decreases, x^+ and thus NTU and the heat exchanger effectiveness.

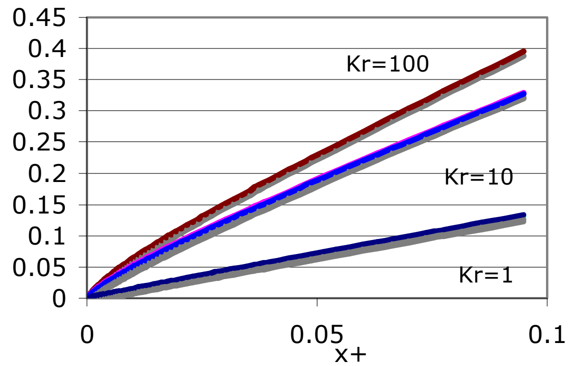


Figure 5 Variation of the Average NTU along balanced parallel flow heat exchanger (α=1)

4. CONCLUSION

In this study, a parallel flow heat exchanger is simulated numerically to determine the impact of entrance effects and conduction heat transfer through the walls on the performance of the heat exchanger, including the number of transfer units and the effectiveness. The impact of wall conduction on the heat transfer is studied for different wall thickness and thermal conductivity. It is shown that all such impacts are reflected in the average number of transfer units. It is shown that there is significant change in the heat exchanger effectiveness in the developing region and that the three-dimensional heat transfer in the heat exchanger walls must be included in the analysis. Accounting for the variation in the overall heat transfer coefficient will significantly reduce the heat exchanger length required to reach the maximum possible effectiveness. For conventional heat exchangers, the thin fin assumption is a reasonable approximation, and therefore the performance of the heat exchanger is primarily dependent on the flow in the ducts, i.e. the fluid properties, mass flow rate, and aspect ratios.

For microchannel heat exchangers, the conduction in the wall is important, and the thin fin approximation will not be satisfied for practical microchannel heat exchangers. This requires the solution of the three dimensional equations in both solid and fluid. 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 $K_r > 40$.

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