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# PERFORMANCE OF CROSS FLOW MICROCHANNEL HEAT EXCHANGERS SUBJECTED TO VISCOUS DISSIPATION

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## ABSTRACT

This paper analyzes the effect of viscous dissipation on the thermal performance of balanced flow cross flow microchannel heat exchangers. The cross flow microchannel heat exchanger analyzed in this paper is one that is subjected to axial heat conduction. Governing equations are developed for each of the fluids and the wall separating the fluids. The equations are solved simultaneously using the numerical technique of finite difference method to obtain the temperature profile. The effectiveness of each fluid is determined using the temperature profile thus obtained. The effectiveness and the temperature of the fluids are found to depend on NTU, axial heat conduction parameters and the viscous dissipation parameter. In the presence of axial heat conduction the effectiveness of the fluid decreases for a specific NTU. In addition, the effectiveness of the fluids decreases with increase in axial heat conduction parameters at a particular NTU. The effectiveness of the hot fluid in the presence of viscous heat dissipation alone decreased at a particular NTU. On the other hand the effectiveness of the cold fluid for the same amount of viscous heating improved at a specific NTU. The combined effect of axial heat conduction and viscous dissipation on the hot fluid is to decrease its effectiveness. With regard to the cold fluid effectiveness it can either increase or decrease due to the combined effect of axial heat conduction parameter and viscous dissipation.

## INTRODUCTION

Microchannel heat exchangers (MCHX) are two-fluid heat transfer devices employing microchannels. MCHXs are used in several fields of engineering including mechanical, chemical, and biomedical engineering. Employing channels with hydraulic diameter smaller than 1 mm have certain advantages such as enhanced heat transfer coefficient, increased heat transfer surface area per unit volume and these features in turn help reduce the overall size of the device which is especially useful for applications where space is limited. All types of materials are currently used for fabricating MCHXs. The materials used till date includes glass, silicon, alumina, silicon carbide, copper, and PDMS [1]. The material of choice for fabricating a MCHX depends on operating parameters like temperature, pressure and chemical inertness. The fabrication method is almost always dictated by the material used. Commonly used fabrication methods include chemical etching, sand blasting, mechanical micromachining, molding and stereolithography [2, 3]. MCHXs can be operated in three different flow configurations; 1) parallel flow, 2) counter flow and 3) cross flow. In parallel flow configurations both the cold and hot fluid flow in the same direction while in the counter flow arrangement the direction of flow of the hot fluid is opposite to that of the cold fluid. In cross flow arrangement the hot fluid flows in a direction perpendicular to that of the cold fluid. Each of these configurations has advantages over the other and the selection of the flow configuration depends on the application. One of the earliest applications of MCHX has been in microminiature refrigerators which employed a counter flow MCHX ($MCHX_{CF}$) [4]. It was used for exchanging heat between high and low pressure streams [5]. A MCHX in parallel flow configuration ($MCHX_{PF}$) has been proved to be more beneficial for use as a chemical reactor for micro fuel cells due to the absence of axial heat conduction [6]. Cross flow MCHX ($MCHX_{CrF}$) has been used by Wilhite et al. [7, 8] in conjunction with a chemical reactor for cooling the products of chemical reaction. Pressure drop and fouling are the two main concerns when microchannels are used in heat transfer devices. Pressure drop is inversely proportional to hydraulic diameter of a channel and thus with reduction in the channel diameter pressure drop associated with a specific flow rate increases. One way to overcome this without affecting the throughput would be stack layers of substrates with multiple microchannels in each substrate. Fouling is another issue commonly encountered in microfluidic devices especially the ones used for carrying out chemical reactions. Fouling of channels will also contribute to increase in pressure drop. Fouling will lead to increased thermal resistance and reduced throughput. Currently there is a thrust towards developing techniques to mitigate the effect of fouling in microchannels. One approach that has been suggested is the use of ultrasound to remove the scales from the channel walls [9]. Another approach is to choose the material of the substrate in such a way that it will be able to partially/fully resist any deposition [10]. Techniques to increase the shear stress on the microchannel walls are also considered as a technique to mitigate fouling. To this extend researchers have suggested the use of pulsatile flow in microchannels as well as use of zig-zag microchannels instead of straight channels to increase the velocity/shear stress near the wall [11]. Irrespective of these two drawbacks MCHXs have been commercialized by firms such as Chart Industries and Heatric®.

MCHXs are designed using conventional ε-NTU relationships [12]. These relationships do not account for the effect of axial heat conduction, viscous dissipation, flow maldistribution and external heat transfer on the effectiveness of MCHXs. In this paper the effect of viscous dissipation on the effectiveness of the MCHX is studied. Viscous dissipation is the conversion of pumping power into thermal energy. Pumping power in microchannel is much higher than that in macroscale channels for the same specific flow rate because of the high pressure drop. Thus it is important to consider the effect of viscous dissipation while designing MCHXs and subsequently studied in this paper. Among the three different flow configurations a $MCHX_{CF}$ has the best performance while a $MCHX_{PF}$ has the lowest performance of a particular NTU. Even though a $MCHX_{CF}$ has better performance than a $MCHX_{CrF}$ it has an advantage with respect to location of the manifolds. As the fluids flow perpendicular to each other the manifolds need not be arranged on top of each other like in the case of counter flow or parallel flow MCHX. This arrangement of manifolds eases the fabrication MCHXs which are almost always planar in nature. Therefore in this paper the effect of viscous dissipation in $MCHX_{CrF}$ subjected to axial heat conduction is analyzed.

## NOMENCLATURE

| | |
|---|---|
| A | : area ($m^2$) |
| C | : heat capacity (J/K) |
| $C_p$ | : specific heat (J/kg K) |
| dZ | : differential nondimensional length |
| h | : heat transfer coefficient ($W/m^2K$) |
| k | : thermal conductivity |
| Kn | : Knudsen number |
| L | : length of the microchannel (m) |
| Ma | : Mach number |
| NTU | : number of transfer units |
| T | : temperature (K) |
| x | : axial coordinate along $L_x$ (m) |
| y | : axial coordinate along $L_y$ (m) |
| Z | : nondimensional axial coordinate |
| λ | : axial heat conduction parameter |
| θ | : nondimensional temperature |
| ψ | : viscous dissipation parameter |

**Subscripts:**

| | |
|---|---|
| c | : cold fluid |
| cr | : cross section |
| h | : hot fluid |
| w | : wall |
| x | : x-coordinate |
| y | : y-coordinate |

## LITERATURE REVIEW

In this section a few of the articles analyzing the effect of axial heat conduction and viscous dissipation on the thermal performance of MCHXs have been reviewed.

Chiou [13] carried out theoretical studies on the thermal performance of cross flow heat exchangers subjected to axial heat transfer. The effect of axial heat conduction is to bring about equal reduction in the effectiveness of both the fluids. The degradation in the effectiveness of the fluids increased with increase in NTU for a balanced flow cross flow heat exchanger. On the other hand for unbalanced flow heat exchanger operating under similar conditions the reduction in effectiveness initially increased with increase in NTU but with further increase in NTU this reduction in effectiveness decreased.

Mathew and Hegab [14] studied the effect of external heat transfer and internal heat generation (viscous dissipation) on the thermal performance of a $MCHX_{PF}$ under balanced flow conditions. External heat transfer refers to the heat transfer between the fluids of the $MCHX_{PF}$ and the ambient. Depending on the ambient temperature the external heat transfer can be either from each of the fluids to the ambient or from the ambient to the fluids. If the external heat transfer is from the hot fluid to the ambient then its effectiveness increases otherwise it decreases. Similar situation exists with the cold fluid. Moreover, if the ambient temperature is below the inlet temperature of the cold fluid then the effectiveness of the hot fluid increases while that of the cold fluid decreases. On the other hand if the ambient temperature is greater than the inlet temperature of the hot fluid then the hot and cold fluid effectiveness decreases and increases, respectively. Regarding the individual effect of internal heat generation, it always decreases and increases the effectiveness of the hot and cold fluid, respectively.

Mathew and Hegab [15] recently analyzed the combined effect of axial heat conduction and external heat transfer in a balanced flow $MCHX_{CF}$. The fluids are subjected to a constant heat flux external thermal source and this is the source of external heat transfer. Axial heat conduction in a $MCHX_{CF}$ will decrease the effectiveness of both the fluids for all values of NTU. The effect of external heat transfer will act to reduce the effectiveness of the hot fluid while it increases the effectiveness of the cold fluid under all circumstances. The combined effect of axial heat conduction and external heat transfer has a negative effect on the effectiveness of the hot fluid if external heat flux is directed into the hot fluid. On the other hand the effectiveness of the cold fluid can either increase or decrease when axial heat conduction is present along with external heat flux subjected into the cold fluid. This is because axial heat conduction has a negative effect on the cold fluid effectiveness while external heat transfer has a positive effect. Thus depending on which of these two effects is greater the cold fluid effectiveness can either increase or decrease. The trends observed with respect to the effectiveness of the fluids would reverse if the external heat flux is directed out of the fluids in the presence of axial heat conduction.

From the few articles surveyed here it is clear that currently no model is available in literature that deals with axial heat conduction and viscous dissipation in a $MCHX_{CrF}$. This paper provides such a model and uses the model to analyze such a $MCHX_{CrF}$.

## THEORETICAL MODEL

Figures 1 is a schematic of the $MCHX_{CrF}$ analyzed in this paper. The width of the $MCHX_{CrF}$ in both the directions is assumed same for this study. The development of the thermal model is simplified by making certain assumptions. These assumptions are provided below.

1) The $MCHX_{CrF}$ under study is an unmixed-unmixed type.
2) The $MCHX_{CrF}$ operates under steady state condition
3) Temperature is uniform across every cross section of the microchannels.
4) Heat transfer between the fluids is continuous in the region between each inlet and outlet.
5) The length of the $MCHX_{CrF}$ in the x-direction is same as that in the y-direction.
6) Effects such as flow maldistribution and external heat transfer are considered negligible.
7) No-slip flow condition is assumed on all the walls of the $MCHX_{CrF}$. For gases, Kn should be smaller than $10^{-3}$ in order for the no-slip flow condition to be valid.
8) If gases are used in both set of microchannels then they are assumed to be incompressible (Ma < 0.3).

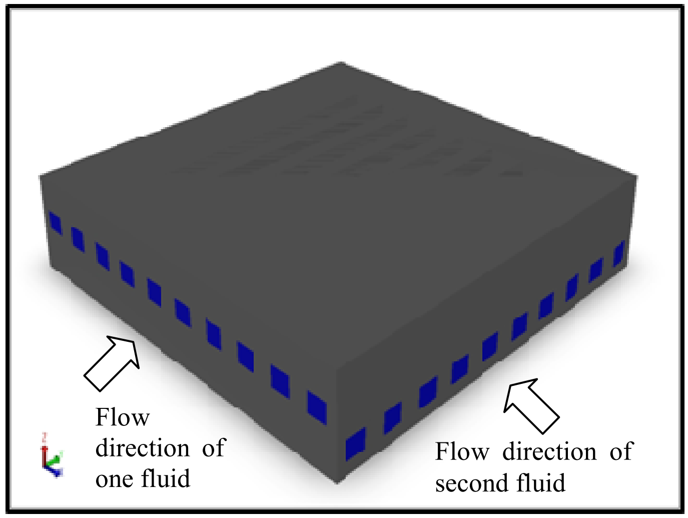

Figure 1: Schematic of a $MCHX_{CrF}$.

Based on the above mentioned assumptions the governing equations for both the fluids and the wall are developed. The governing equations are based on the principles of continuum mechanics and thus it can be applied to even macroscale heat exchangers in which viscous dissipation is a concern. However, the effect of viscous dissipation is most relevant in a $MCHX_{CrF}$. The governing equation of the hot fluid is provided in Eq. (1) while that of the cold fluid is shown in Eq. (2).

$$\frac{\partial T_h}{\partial x} + \frac{hp_s}{C_h}(T_h - T_w) = -\frac{\nu}{C_h}\frac{dP_h}{dx} \tag{1}$$

$$\frac{\partial T_c}{\partial y} - \frac{hp_s}{C_c}(T_w - T_c) = -\frac{\nu}{C_c}\frac{dP_c}{dy} \tag{2}$$

$$kA_{cr}\frac{\partial^2 T_w}{\partial x^2} + kA_{cr}\frac{\partial^2 T_w}{\partial y^2} + hp_s(T_h - T_w) - hp_s(T_w - T_c) = 0 \tag{3}$$

The terms of these equations are converted into nondimensional form by using the following terms.

$$\theta = \frac{T - T_{c,i}}{T_{h,i} - T_{c,i}}, \quad 2NTU = \frac{hA}{C_{\min}}, \quad Z_x = \frac{x}{L_x}, \quad Z_y = \frac{y}{L_y}$$

$$\lambda_{Z_x} = \frac{kA_{cr}}{L_x C_{\min}}, \lambda_{Z_y} = \frac{kA_{cr}}{L_y C_{\min}},$$

$$\psi = \frac{-\nu\dfrac{dP_h}{dx}}{C_{\min}(T_{h,i} - T_{c,i})} = \frac{-\nu\dfrac{dP_c}{dy}}{C_{\min}(T_{h,i} - T_{c,i})},$$

The viscous dissipation parameter ψ is the ratio of the pumping power to the maximum heat transfer possible in a $MCHX_{CF}$. Higher this value, higher the pumping power with respect to the maximum heat transfer possible. The longitudinal heat conduction parameter is the ratio of the maximum calorific thermal resistance in the $MCHX_{CrF}$ to the conduction thermal resistance between the inlet and outlet section of each fluid. The governing equations in nondimensional form can be rewritten in the following form. If the $MCHX_{CrF}$ is operated under balanced flow conditions then the heat capacity of the hot fluid and cold fluid is equal.

$$\frac{\partial \theta_h}{\partial Z_x} + 2NTU\left(\theta_h - \theta_w\right) = \psi \tag{4}$$

$$\frac{\partial \theta_c}{\partial Z_y} + 2NTU\left(\theta_w - \theta_c\right) = \psi \tag{5}$$

$$\lambda_{Z_x}\frac{\partial^2 \theta_w}{\partial Z_y^2} + \lambda_{Z_y}\frac{\partial^2 \theta_w}{\partial Z_y^2} + 2NTU\left(\theta_h - \theta_w\right) - 2NTU\left(\theta_w - \theta_c\right) = 0 \tag{6}$$

The first term in both equations correspond to the axial variation of fluid temperature. The second term accounts for the heat transfer between the fluids. The term on the right hand side Eq. (4) and Eq. (5) is the term that accounts for viscous dissipation in the microchannels. The first and second term on the left hand side of Eq. (6) represent the variation in heat conduction through the wall. The third term accounts for the heat transfer between the hot fluid and wall while the fourth term represents that between the wall and the cold fluid. The boundary conditions are provided in Eq. (7) - Eq. (10). The inlet temperature of the fluids is the boundary condition. The specific way in which temperature is nondimensionalized aids in reducing the hot fluid inlet temperature to unity and the cold fluid inlet temperature to zero. The sections of the wall at the inlets and outlets of the fluids are assumed to be insulated and this is mathematically represented as in Eq. (9) and Eq. (10).

$$\theta_h\big|_{Z_x=0} = 1 \tag{7}$$

$$\theta_c\big|_{Z_y=0} = 0 \tag{8}$$

$$\left.\frac{\partial \theta_w}{\partial Z_x}\right|_{Z_x=0} = \left.\frac{\partial \theta_w}{\partial Z_x}\right|_{Z_x=1} = 0 \tag{9}$$

$$\left.\frac{\partial \theta_w}{\partial Z_y}\right|_{Z_y=0} = \left.\frac{\partial \theta_w}{\partial Z_y}\right|_{Z_y=1} = 0 \tag{10}$$

These equations are solved using the finite difference method [16] The convective term of Eq (4) and (5) is discretized using first order upwind scheme while the diffusive term in Eq. (6) is discretized using the second order central difference scheme [16]. The discretized form Eqs. (4) - (6) is shown in Eqs. (11) and (13). The boundary conditions presented in Eqs. (9) and (10) are discretized using second order central difference scheme [16]. The error criterion used for absolute convergence criterion of the temperature of both the fluids and wall is maintained at $10^{-7}$. Grid independency is checked by changing the inter-node distance from 0.05 to 0.004 with an intermediate grid size of 0.01. The average of the outlet temperatures of the fluids at each of these grid sizes is compared to check the dependency of the results on the grid size.

$$\theta_h\big|_{i,j} = \left(\frac{1}{NTU + \frac{1}{\Delta h_{Z_x}}}\right)\left(\frac{\theta_h\big|_{i-1,j}}{h_{Z_x}} + 2NTU\,\theta_w\big|_{i,j} + \psi\right) \tag{11}$$

$$\theta_c\big|_{i,j} = \left(\frac{1}{NTU + \frac{1}{\Delta h_{Z_y}}}\right)\left(\frac{\theta_c\big|_{i-1,j}}{h_{Z_y}} + 2NTU\,\theta_c\big|_{i,j} + \psi\right) \tag{12}$$

$$\theta_w\big|_{i,j} = \left(\frac{1}{4NTU + \frac{\lambda_{Z_x}}{\Delta h_{Z_x}^2} + \frac{\lambda_{Z_y}}{\Delta h_{Z_y}^2}}\right)\left(\frac{\lambda_{Z_x}\theta_w\big|_{i-1,j}}{\Delta h_{Z_x}^2} + \frac{\lambda_{Z_x}\theta_w\big|_{i+1,j}}{\Delta h_{Z_x}^2} + \frac{\lambda_{Z_y}\theta_w\big|_{i,j-1}}{\Delta h_{Z_y}^2} + \frac{\lambda_{Z_y}\theta_w\big|_{i,j+1}}{\Delta h_{Z_y}^2} + 2NTU\,\theta_h\big|_{i,j} + 2NTU\,\theta_c\big|_{i,j}\right) \tag{13}$$

The discretized equation of the hot fluid, i.e. Eq. (11), need not be applied to the node at the inlet section of the hot fluid microchannel as the temperature at this location is known (Eq. (7)). Similarly, Eq. (12) will only be applied to all nodes other than the one at the entrance to the cold fluid microchannel. The temperature at the inlet of the cold fluid microchannel is known a priori as shown in Eq. (8). However Eq. (13) has to be applied even to the nodes at the inlet and outlet sections of each fluid because the boundary conditions of the wall do not specify temperature, rather it specifies zero heat flux through the wall at these locations. Equation (13) can only be applied to the interior nodes of the wall. This equation can be used for the nodes at the boundary after incorporating the specific boundary condition into this equation.

The effectiveness of the $MCHX_{CrF}$ is defined as provided in Eq. (14) and Eq. (15). In a $MCHX_{CrF}$ without viscous dissipation the effectiveness of both the fluids will be the same. However, in this case the hot fluid effectiveness will be different from that of the cold fluid due to the presence of an additional heat source and both need to be analyzed. The effectiveness of the fluids is calculated based on the average of the outlet temperatures of each of the fluids.

$$\varepsilon_h = 1 - \theta_h\big|_{Z_x=1} \tag{14}$$

$$\varepsilon_c = \theta_c\big|_{Z_y=1} \tag{15}$$

The theory provided in this paper can be used for both fully developed as well as developing flow. This is because the flow condition mainly affects the heat transfer coefficient in the microchannels and this parameter is used for determining the operating parameter NTU. Thus by using correlations which are specific for the flow condition it would be possible to determine the appropriate heat transfer coefficient and in turn determine the NTU accurately for the actual flow condition in the microchannel.

## RESULTS AND DISCUSSION

The validity of the governing equations and the numerical technique is checked by comparing the predictions of this model with that obtained from the ε-NTU relationships available in literature [10]. Two set of ε-NTU relationships are available in literature: 1) the ε-NTU relationship of a $MCHX_{CrF}$ free of viscous heating and axial heat conduction, 2) the ε-NTU relationship of a $MCHX_{CrF}$ free of viscous heating but subjected to axial heat conduction [10]. For comparing the ε-NTU relationship obtained from the present model with that of a $MCHX_{CrF}$ free of viscous heating and axial heat conduction the parameters ψ and λ are set to zero. The comparison between the results are compared in Fig. 2.

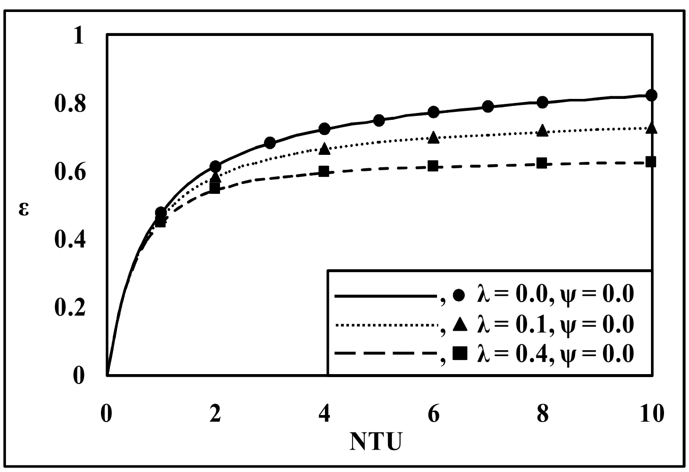


Figure 2: Comparison between ε-NTU of a balanced flow $MCHX_{CrF}$ determined using the present model with that available in literature (●: Conventional ε-NTU relationship; ▲, ■: ε-NTU relationship that includes axial heat conduction) [10].

The line represents the ε-NTU relationship based on the results obtained from the present model while that from the conventional ε-NTU relationship is represented by data points. For the second comparison, the viscous dissipation parameter, ψ, is set to zero while the axial heat conduction parameter is set to a finite value. Comparisons between the present model and that available in literature are done for λ = 0.1, and 0.4. The curve represents the results obtained from the model developed in this paper while that from literature is shown by data points. From Fig. 1 it can be seen that the results of the model developed in this paper matches exactly with that available in literature [10]. This matching of results validates the model as well as the discretization schemes used in this paper.

Figure 3 represents the effect of viscous heat dissipation on the hot fluid effectiveness of a $MCHX_{CrF}$ subjected to axial heat conduction. The viscous dissipation parameter is varied between 0.0 and 0.3 with increment of 0.1. When ψ = 0.1 means that the total heat due to viscous dissipation is 10% of the maximum heat transfer possible in a heat exchanger. The solid line in this figure represents the variation of effectiveness with NTU of a $MCHX_{CrF}$ free of axial heat conduction and viscous dissipation.

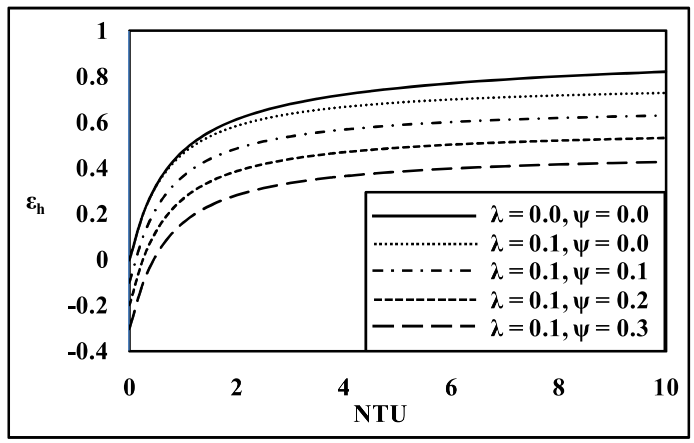


Figure 3: ε-NTU relationship of hot fluid with constant axial heat conduction parameters and varying viscous dissipation parameter.

It can be seen from this figure that in the presence of just axial heat conduction the effectiveness of the hot fluid decreased at every NTU. The decrease in ε for a finite value of λ is because of the redistribution of wall temperature due to heat conduction through the wall. In the presence of viscous dissipation and axial heat conduction the effectiveness of the hot fluid for every NTU decreased below that in a $MCHX_{CrF}$ subjected to just axial heat conduction. This is because viscous heat dissipation which is equivalent to an internal heat source generates heat inside the microchannels. The total amount of heat generated due to viscous heat dissipation between the inlet and outlet of the microchannel is equal to the pumping power. Thus the amount of heat generated per unit length of the each microchannel is mathematically equal to the total pumping power per unit length. The heat generated due to viscous dissipation inside each hot fluid microchannel increases its local temperature and thereby reduce its effectiveness.

The effectiveness of the hot fluid in a $MCHX_{CrF}$ with axial heat conduction increased with increase in NTU even in the presence of viscous dissipation. This is because of the increase in heat transfer between the hot fluid and the wall that is associated with increase in NTU as well as due to the face that the heat generated due to viscous dissipation is maintained constant. NTU can be increased with either decrease in heat capacity of the fluids or by increasing the thermal conductance of the $MCHX_{CrF}$. Either of these options if implemented will increase the heat transferred between the hot fluid and the wall. Decrease in heat capacity of the fluids will increase the residence time. This increase in residence time will lead to increased heat transfer between the hot fluid and the wall. On the other hand if the thermal conductance of the $MCHX_{CrF}$ is increased to increase NTU then thermal resistance to heat transfer will reduce and will lead to increased heat transfer between the hot fluid and the wall.

When only axial heat conduction is present the effectiveness of the hot fluid is zero when NTU is zero. This is because when NTU is zero there is no heat transfer between the hot fluid and wall thereby eliminating the possibility of heat conduction through the wall that could lead to degradation of effectiveness. No heat exchanger will ever be operated at this NTU but is discussed here in order to make the complete the theoretical analysis performed in this paper. The hot fluid effectiveness of a $MCHX_{CrF}$ subjected to axial heat conduction is numerically equal to the viscous dissipation parameter when NTU is set to zero. Even though there is no heat transfer between the fluids when NTU is zero there is heat being generated inside the hot fluid microchannel due to the phenomenon of viscous dissipation. The heat generated inside the microchannel can neither be partially or completely removed due to the absence of heat transfer. This heat due to viscous dissipation which remains in the hot fluid raises its temperature thereby reducing its effectiveness. Therefore in absence of heat transfer between the hot fluid and the wall the total nondimensional heat gained by the hot fluid is numerically equal to the viscous dissipation parameter. Thus the effectiveness of the hot fluid is equal to the negative of the viscous dissipation parameter.

Figure 4 represents the effectiveness of the cold fluid due to viscous dissipation in a $MCHX_{CrF}$ subjected to axial heat conduction. The axial heat conduction parameter is maintained at 0.1 while the viscous dissipation parameter is varied from 0.0 to 0.1 to 0.2 to 0.3. The effectiveness of the cold fluid of a $MCHX_{CrF}$ subjected to just axial heat conduction is lower in comparison with a $MCHX_{CrF}$ free of axial heat conduction and viscous dissipation. However, when viscous dissipation is present in addition to axial heat conduction then the effectiveness of the cold fluid increased above that of a $MCHX_{CrF}$ free of axial heat conduction and viscous dissipation. This is because heat generated due to viscous dissipation has a positive effect on the effectiveness of the cold fluid. The degradation in effectiveness due to axial heat conduction is more than compensated due to viscous dissipation. Moreover, with increase in viscous dissipation parameter the effectiveness of cold fluid increased for a specific NTU and axial heat conduction parameters. In addition the effectiveness of the cold fluid increased with increase in NTU for specific values of viscous dissipation parameter and axial heat conduction parameters. The effectiveness of the cold fluid is numerically equal to the viscous dissipation parameter when NTU is zero. The reasons for these are similar to that provided for the trends observed with respect to the hot fluid.

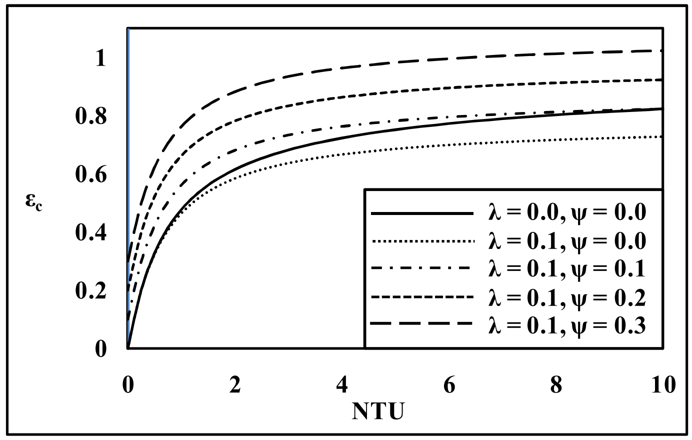


Figure 4: ε-NTU relationship of cold fluid with constant axial heat conduction parameters and varying viscous dissipation parameter.

Figure 5 represents the effect of axial heat conduction on hot fluid effectiveness of a $MCHX_{CrF}$ that is subjected to viscous dissipation. In the presence of viscous dissipation alone the effectiveness of the hot fluid decreases. The reason for this is similar to that stated earlier, i.e. viscous dissipation generates heat inside the microchannel which causes the fluid temperature to be higher than that in a $MCHX_{CrF}$ free of viscous dissipation. If axial heat conduction also exists in this $MCHX_{CrF}$ then effectiveness of the hot fluid further degrades. This is because the heat conduction through the wall redistributes the wall temperature leading to lower heat transfer between the hot fluid and the wall. With increase in axial heat conduction parameters the effectiveness of the hot fluid decreases for a specific NTU when the viscous dissipation parameter is kept constant.

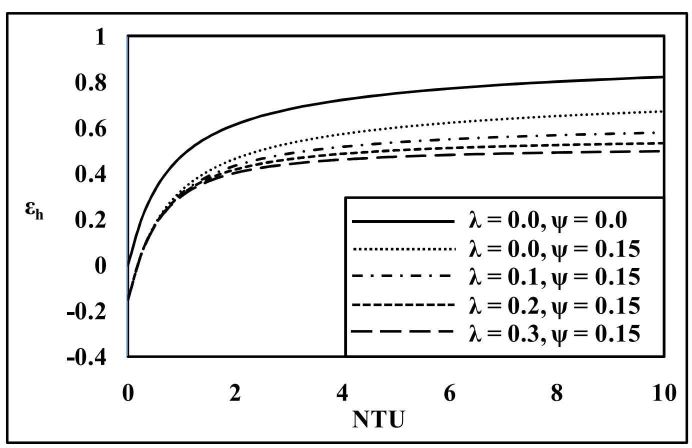


Figure 5: ε-NTU relationship of hot fluid with varying axial heat conduction parameters and constant viscous dissipation parameter.

With increase in NTU the effectiveness of the hot fluid of a $MCHX_{CrF}$ subjected to viscous dissipation increases for all values of axial heat conduction parameters. This is because of the improvement in heat transfer between the hot fluid and the wall. Similar to the earlier case, the increase in NTU will bring about increase in heat transfer between the fluids due to either increase in the residence time or reduction in the thermal resistance between the fluids.

The effectiveness of the hot fluid is numerically equal to the viscous dissipation parameter when NTU is zero even in the absence of axial heat conduction parameter. The reason for this same as that explained for the previous case. Moreoever, when axial heat conduction is introduced there is no change in the effectiveness of the hot fluid at NTU of zero. This is because when NTU is zero axial heat conduction is non-existent because there is no heat transfer between the hot fluid and the wall. Thus only viscous dissipation is present which tends to decrease the hot fluid effectiveness.

Figure 6 contains the effectiveness of the cold fluid of a $MCHX_{CrF}$ subjected to viscous dissipation with varying degree of axial heat conduction. The viscous dissipation parameter is kept at 0.15 and the axial heat conduction parameters are varied from 0.1 to 0.2 to 0.3. The effectiveness of the fluids in the presence of viscous dissipation alone increased above that in a $MCHX_{CrF}$ free of both the effects. The heat generated inside the cold fluid microchannel due to viscous dissipation increases its temperature more than that would occur in a $MCHX_{CrF}$ free of these two phenomena and thus the observed increase in its effectiveness.

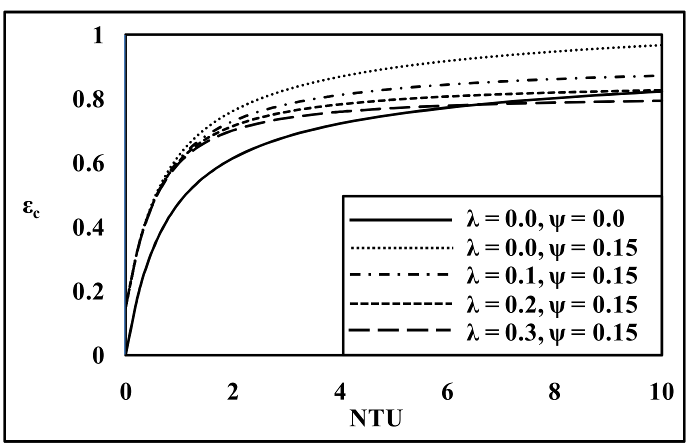


Figure 6: ε-NTU relationship of cold fluid with varying axial heat conduction parameters and constant viscous dissipation parameter.

When axial heat conduction is introduced, the effectiveness of the cold fluid starts to drop below that in a $MCHX_{CrF}$ just with viscous dissipation. The degree of drop depends on the nullifying effect of axial heat conduction on the amount of the heat that is gained by the cold fluid due to viscous heating. With increase in axial heat conduction parameters the effectiveness of the cold fluid drops for a specific NTU and $\psi$. This is because with increase in axial heat conduction parameters the amount of heat transferred from the wall to the cold fluid decreases thereby reducing the heat gained by the cold fluid. The cold fluid effectiveness is numerically equal to $\psi$ when NTU is zero when axial heat conduction parameters are set to zero. This reason for this has been explained earlier.

## LIMITATIONS OF MODEL

The fifth and sixth assumption puts certain limitations on the model developed in this paper. This limitation is in terms of size of the microchannel especially if gases are used in either of the channels. Based on the fifth assumption the model can only be used if the no-slip flow condition is satisfied on the walls of the microchannel. For this to be valid Kn should be smaller than $10^{-3}$. For air this condition is satisfied only if the channel size is greater than 60 μm. On the other hand for nitrogen the no-slip condition is valid till the channel is reduced to 70 μm. Regarding liquids the no-slip boundary condition is almost always valid irrespective of the hydraulic diameter of the channel as experimentally proved by researchers [17]. The sixth assumption demands that if the fluids used in the $MCHX_{CrF}$ are gases then they need to be incompressible. For microchannels gases can be considered to be incompressible if $Ma < 0.3$ [18]. So from this criterion on Ma it is possible to determine the minimum size of the microchannels that is needed to satisfy this criterion for a specific flow rate and vice versa.

## CONCLUSION

This paper deals with the effect of viscous dissipation in a cross flow microchannel heat exchanger subjected to axial heat conduction. The phenomenon of axial heat conduction degrades the effectiveness of both the fluids by equal amount. The individual effect of viscous dissipation is to increase the effectiveness of the cold fluid while decreasing the effectiveness of the hot fluid. The combined effect of these two phenomena will decreased the effectiveness of the hot fluid. On the other hand the combined effect of these two phenomena can either increase or decrease the effectiveness of the cold fluid based on the net heat gained/lost due in the presence of these two phenomena.

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