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Computational Studies of Heat Transfer Enhancement in a Circular Wavy Micro Channel

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Using computational analysis, convective heat transfer augmentation of laminar flow through wavy micro channel was studied. A 3-D geometry was used for simulation. The simulation was performed using non dimensional governing equations of a laminar flow at steady state. The flow is developing by both thermally and hydrodynamically while the walls of the channel are set aside at a steady temperature. The computations were prepared with Reynolds number within the range from 100 to 800 using air (Pr=0.7) as the working fluid. The simulation was carried out by using five different wave ratio (Y= P/D) to reach the optimal geometry with the maximum performance evaluation criterion. The results portrayed that the efficiency of heat transfer in wavy shaped micro channel was improved than a typical circular micro channel. The penalty due to the drop of pressure also increased in case of wavy micro channel whereas the channel having smaller wave ratio has more friction penalty. Commonly, the heat transfer act of wavy micro channel has an augmented heat transfer efficiency because of the thermal boundary layer disorder and block of longitudinal heat transfer.
Keywords: Computation, 3-D flow, wavy, micro channel, enhancement, heat transfer, overall enhancement factor

1. Introduction

Heat transfer behavior in duct using swirl generator Bhattacharyya et al., (2016, 2016, 2017) encountered in many industrial applications such as solar thermal heaters and heat exchangers. The castigation of fluidics in the micro range has been established as a major field of research due to its various tenders in various forms. Recently, newer technologies in the field of fluidics in the micro range have solicitations ranging from analgesics and biomedical industries (such as drug design, delivery & detection, diagnostic devices etc.) to electronics industry (as heat exchangers for integrated cooling of electronic circuits). Widespread applications like print head of an inkjet printer or nib of a fountain pen etc. can even be visualized in in daily lives. Cells containing fuels, pumps and gas turbines are some other regions of applications. The incessant augmentation in the functionality and reduction in scope of electronics in the micro range has released the requisites for new ways for operative removal or dissipation of high fluxes of heat in order to enhance its performance. The controlling of heat of micro-electronic devices becomes a stimulating problem due to the small dimensions of the devices required for the removal of heat and very stern working temperature conditions for optimal performance of electronic devices. An amount of different practices for removal of heat such as intruding jets and heat pipes have been applied to achieve optimum cooling. Usage of mixers in the micro range on the channel wall helps to accomplish effective removal of fluxes of heat. The chief goals of this study are minimizing maximum device temperature gradients that are possible by configuring the wall of circular shape micro-mixers. The commencement of study of channels in the micro range ensued by Tuckerman and Pease (1981) published a work that apportioned with the beneficial effects of using channels which have less values of diameter for dissipation of heat purposes of very large scale integrated circuits. It was revealed that the hydraulic diameter of the channel varies inversely to the coefficient of transfer of heat. Some of the outcomes achieved by means of experiment for flow of fluid in micro-channel using gas smidgens its source to Wu and Little (1983). In this experiment the friction factors in the laminar region were more than expected, and they found that 350 to 900 was the transition regime. A research was carried out by Pfahler et al., (1991) on flow of fluid in micro channels. It was set up that for large flow channels the interpretations obtained by conducting

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tests were in coarse agreement with the conventional model whereas for small channels the level of difference augmented. In addition to the previous work, Pfahler et al., (1990) extracted measurements, through experimentation of friction factor or apparent viscosity of silicon oil and isopropyl alcohol by virtue of its flow in micro channels. They detected that for channels of grander dimensions the outcomes of research exhibited an insignificant discrepancy with the probable values of conventional theory. Nevertheless, the superficial viscosity began to drop from the theoretical value for a constant pressure drop, as the channel dimensions shrank, though markedly contradictory behaviours were noticed between the non-polar silicon oil and the polar isopropyl alcohol. Investigation by Choi et al., (1991) on the friction factor, coefficient of Nusselt number and the substantial paraphernalia of roughness of surface of inner wall for flow in laminar and turbulent regime in micro channels. An assured incongruence amid conventional flow model and consequences of experimental analysis for tubes of micro scale sizes was observed by Peng and Peterson (1994, 1994, 1995, 1995, and 1996). They discovered a dependency of friction factor on hydraulic diameter and aspect ratio of the channel in which flow of fluid takes place after commencing the analysis of an assortment of micro channels of diverse hydraulic diameters ranging from 133 μm to 367 μm. Investigational examinations were carried out by some researchers for single-phase forced convection in micro channels of rectangular cross-section and significant depth. Testing for two configurations were efficaciously executed a multiple channel system and a single channel system. In this particular case of the multiple channel system the channels were 251 μm in width and the channel walls were 119 μm in thickness. In those schemes the dimensions of the channels approximated to 1000 μm in depth and a projected area of 2.5 cm×2.5 cm is demarcated. The fluid flow was in the form of de-ionized water, of Reynolds number varying from 173 to 12900. The convective heat transfer that was found through investigational means coordinated realistically well with conventional developing channel flow theory. A theoretical as well as experimental work by G. Hetsroni et al., (2005), on single-phase heat transfer in micro channels, shed light on several planes of flow in micro channels as pressure drop, evolution of flow of fluid from laminar to turbulent regime etc. Heat transfer relating to trifling Knudsen number was considered as the problem. Analysis of data of heat transfer in micro-channels with hydraulic diameter range from 60 μm to 2000 μm was accomplished. The substantial effects from different dimensional variations, axial heat flux due to thermal conduction through the fluid used in the experimental setup and channel walls, as well as the dissipation of energy were discussed. It was found by them that the effect of dissipation of energy on transfer of heat in micro-channels is insignificant under typical condition of flow.

The literature review suggests that the transfer of heat by asset of flow of fluid in micro channel with different roughness geometry was studies in the past. Very litter channel modification work found in literature. A wavy micro channel is attempted in this study. In this paper, the laminar flow numerical heat transfer and pressure drop results of circular wavy micro channel is presented.

2. Computational Domain and Boundary Conditions

The full length wavy micro-channel as shown in Figure 1, of inlet diameter (D) and pitch, (P) is engaged. Air velocity was familiarized at inlet of the channel and a pressure outlet condition was smeared at exit. Air inlet temperature of 200K was set in the direction of fluid flow. The temperature of wall was kept constant at 250 K throughout the test. At mean bulk temperature, the thermal and the physical properties of air were taken to be invariant. The channel walls were presumed to be impermeable and no-slip condition was instigated.

3. Mathematical Model

The basic form of continuity, energy and momentum equations for a three dimensional, incompressible, steady state flow and laminar forced convection of transfer of heat minus viscous dissipation are as follows:

Continuity Equation

$$\frac{\partial \mathrm{u}}{\partial \mathrm{x}}+\frac{\partial \mathrm{v}}{\partial \mathrm{y}}+\frac{\partial \mathrm{w}}{\partial \mathrm{z}}=0 \tag{1}$$

Momentum Equation

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial x}+\ w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\ \frac{\partial^2 u}{\partial z^2}\right) \tag{2}$$

$$\rho\left(u\frac{\partial v}{\partial y}+v\frac{\partial v}{\partial y}+\ w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\ \frac{\partial^2 v}{\partial z^2}\right) \tag{3}$$

Energy Equation

$$u\frac{\partial T}{\partial y} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) \quad (4)$$

4. Numerical Solutions

Numerous grid sizes were verified as a part of grid independence study. A mesh containing 188,777 elements and 80,982 nodes were cast-off for the present computation after a severe grid independence check. Theoretical equations for laminar model in a 3-D geometry of diameter 0.5 mm and length 50 mm, were elucidated numerically for heating of the working fluid which was air (Pr = 0.7). The 3-D geometry-data file was generated in ANSYS Design Modeler 16.2.

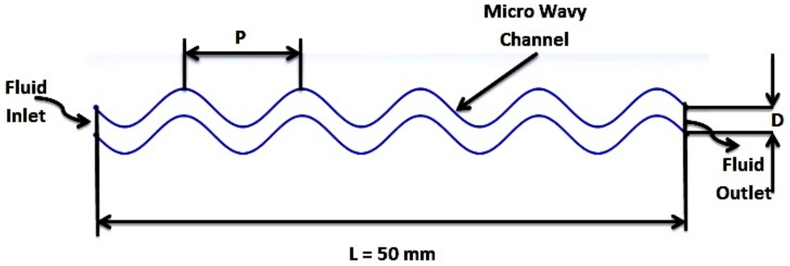

Figure 1: Computational Domain

The equations for discretization of control volume were deduced from these basic equations by employing the hybrid scheme. The procedure of numerical calculations of the flow field is the SIMPLE algorithm which is an acronym for Semi-Implicit Method for Pressure-Linked Equations.

ANSYS Fluent 16.2 is used for the development of the three-dimensional system of grid as shown in Figure 2. Making an allowance for flow of air in the channel with transfer of heat, the mathematical model applied is composed of the conservation equations of mass, momentum and energy for incompressible flow in two dimensions with the following assumptions:

- The flow is three-dimensional, laminar and stationary
- The properties of the air, which are thermo-physical, are assumed to be invariant.
- The thermal conductivity of the walls is supposed to be constant.

The Reynolds number of flow of air in the duct is found out from the following expression:

$$Re = \frac{\rho v D}{\mu}; \quad (5)$$

The convective heat transfer coefficient is then used to obtain Nusselt number, Nu, from the following expression:

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

The friction factor is found out from the measured values of drop in pressure (ΔP), across the length of test section.

$$f = \frac{2\Delta P D}{\rho L V^2}; \quad (7)$$

To assess the increment of heat transfer under a pumping power of given magnitude, the performance evaluation criteria is expressed by Bhattacharyya et al. (2016) below as:

$$\eta = \left(\frac{Nu}{Nu_0}\right) / \left(\frac{f}{f_0}\right)^{0.33}; \quad (8)$$

Where, Nu_0 and Nu are Nusselt numbers for the smooth channel and the augmented channel correspondingly; f and f_0 are friction coefficients for augmented channel and smooth channel correspondingly.

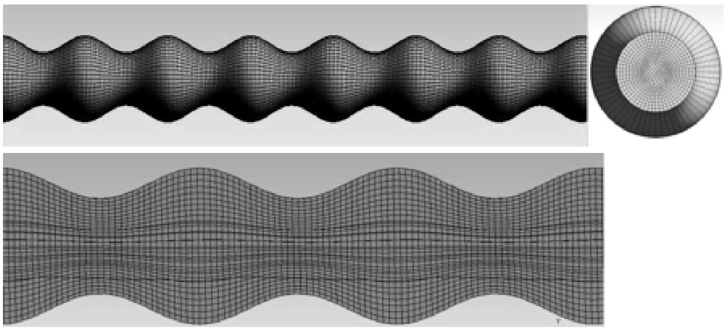

Figure 2. Meshing of computational domain

5. Results and discussion

Correlations delivered by Shah and London (1985) and Darcy (1985) were used to contradict and validate the results of the simulation. The data obtained from CFD simulation of the plain micro-channel tends to the forecasted results from the recommended correlations with little margin of error Shah and London (1985) and Darcy (1985) with data range of +6.3% to +8.2% and +1.3% to +1.5% for the Nusselt number, Nu, and friction factor, f, respectively as revealed in Figure 3 and Figure 4. As predicted by Shah and London (1985) and also visible from Figure 5 the Nusselt number does show a definite trend – increasing with rise in Reynolds number. The addition of waviness in the channel does aid to effectively increase the rate of heat transfer by disrupting the boundary layer development and also by creating local turbulence and swirl flow. This phenomenon accounts for the increase in Nusselt number that is clearly seen in Figure 5. It is also evident from the figure that there is a significant increase in enhancement which is noticed for Reynolds number 500 and above. The results show that with decrease in wave ratio there in an enhancement in transfer of heat. According to the figure, the geometry with y=2.0 gives the best enhancement. The friction factor features acquired as a result of the effects exhibited by the wavy micro channel is shown in Figure 6. The figure symbolizes the association between the Reynolds number and the friction factor at various wave ratio of the wavy micro channel used in the present simulation. It could be well recognized from Figure 6 that the friction factor was in the equivalent trend, both for the channel with wavy nature and also with the plain micro channel. The friction factor of the wavy micro channel is inversely proportional to Reynolds number. At a certain Reynolds number, the micro channel with wavy nature resulted in greater friction factors over those of the plain micro channel.

Additional substantiation that provisions this concept of transfer of heat augmentation inside a micro-channel by generating obstructions/indentations is presented in Figure 7. This indicates the ratio of outlet temperatures for the wavy micro channels to the outlet temperatures of a plain micro-channel. From Figure 7 the same implication can be made – with decreasing nature of wave ratio in wavy micro channel there in an increase in heat transfer which makes it easier to carry more hot fluid through the outlet. The geometry with Y=4.0 is best in achieving this effect.

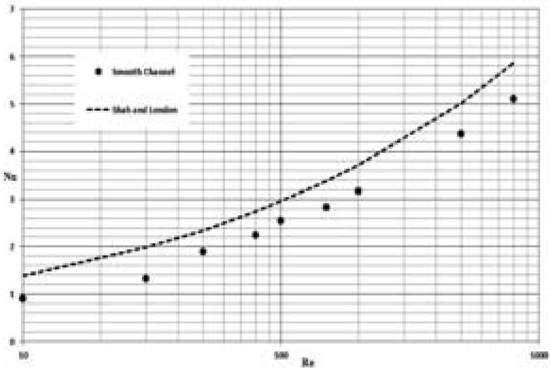

Figure 3: Comparison of the numerical results with the correlation data of the Nusselt number (Nu) of the plain channel

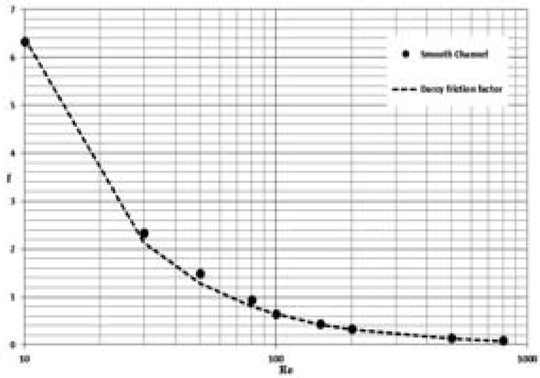

Figure 4: Comparison of the numerical results with the correlation data of the friction factor (f) of the plain channel

Fluid having Reynolds number more than or equal to 300, a noteworthy increase in the outlet temperature is observed. This was because of the blockage of flow, greater contact area of the surface, the phenomenon caused by the swirl flow as well as the dynamic pressure dissipation of the fluid due to the viscosity loss nearby the tube wall. Moreover, the pressure drop had a greater probability to befall by the interaction of the inertia forces with pressure forces in the boundary layer.

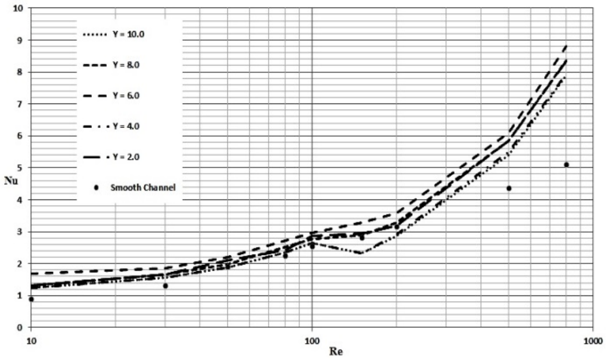

Figure 5: Variation of Nusselt number with Reynolds of wavy micro channel with different wave ratio.

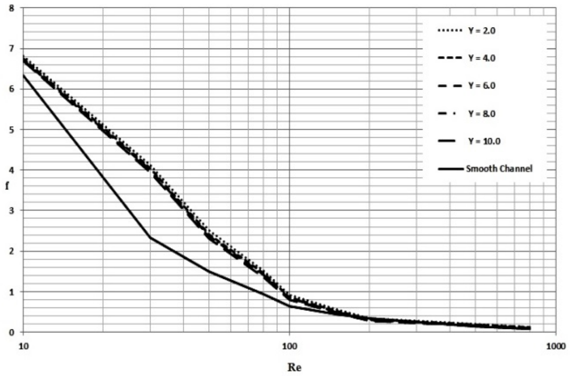

Figure 6: Variation of friction factor with Reynolds of wavy micro channel with different wave ratio.

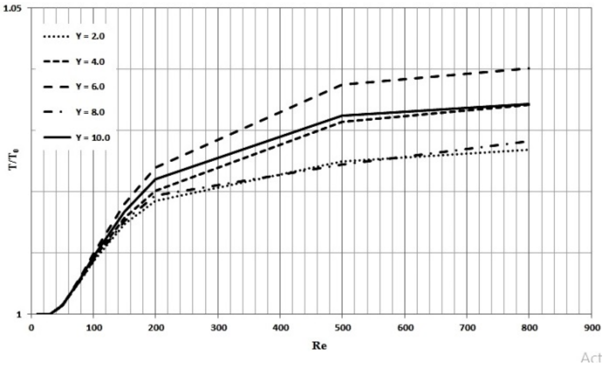

Figure 7: Variation of T/T_0 with Reynolds of wavy micro channel with different wave ratio

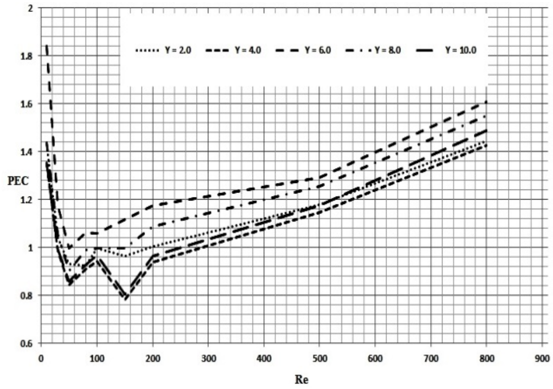

Figure 8: Variation of enhancement efficiency with Reynolds number of wavy micro channel with different wave ratio

On Figure 8, we plotted the curve of the global thermo-hydraulic performance parameter PEC. The thermal performance is the ratio of the dimensionless Nusselt number and the dimensionless friction factor and this ratio shows the amount of the energy is saved. As it is common in heat transfer research and literature the thermal performance is shown in the figure. By doing all the computation test on wavy micro channel and plain micro channel it was found efficient from the energy point of view and enhancement efficiency was found to be almost greater than the unity in high Reynolds number range (above Re 250). The enhancement efficiency above unity indicated that the effect of heat transfer enhancement due to the turbulator was more dominant than the effect of rising friction and vice versa. From the Figure 8, one can see that the channel with wave ratio Y=6.0 provides higher thermal enhancement efficiency through out entire Reynolds number range.

6. Conclusion

In the present study effect of wavy micro channel on the heat transfer coefficient and friction factor is determined numerically. Investigations have been carried out over a range of Reynolds number (Re 10–800). The following conclusions may be drawn from the study is that the high Nusselt number and friction factor are observed for Y=2.0 and 4.0. In general, it can be said that the enhancement of heat transfer for any arrangement of wavy micro channel is caused due to the increased turbulence and the vortex generated due to the swirl flow and secondary flow of air produced along the wave increases the heat transfer from the heated plate to the moving air.

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