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Numerical Investigations of Flow and Heat Transfer in a Wavy Corrugated Channel

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Abstract—*Using corrugated surfaces is a suitable method to increase thermal performance with higher compactness. Corrugations in heat exchangers improve the heat transfer rates 20–30% by increasing heat transfer area and enhancing turbulence at low flowrates. Therefore, most of the compact heat exchangers use corrugated plate between heating and cooling media. Study of heat, mass and momentum transfer at corrugated surfaces is important for the rational design and operation of equipment which employ such surfaces. A two dimensional heat transfer study was investigated in a sinusoidal duct with channel of 20 mm and 40 mm. The 20 mm channel height was carried out for the plate in phase and out of phase arrangements. The commercial code Fluent was used to study the flow features. A low Reynolds number standard k-ε turbulence model was employed to account for turbulence in the flow. Variation of Nusselt number, skin friction coefficient, and pressure drop for different mass flow rate is shown and contours of temperature, velocity and pressure is shown along the channel. For the phase shift plate, the harmonic curves for the local skin-friction coefficient and the local Nusselt number have the same frequency as that of the wavy surface at lower Reynolds numbers. For the plate with same phase the Nusselt number and friction coefficient have higher value at the ridge and lower at the furrow.*

Keywords: *Nusselt number, corrugated channel, heat exchanger*

1. INTRODUCTION

In the recent years, the importance for the development of efficient heat exchangers has grown from the aspect of energy conservation, conversion, recovery, and successful implementation of new energy sources. Compact heat exchangers characterized by high heat transfer surface area to volume ratios have received great attention due to its high heat transfer coefficients compared to other exchanger types. Research has been done to obtain heat exchanger with high effectiveness, low pressure losses, low weight and volume, high reliability and low cost. To obtain these requirements, heat exchangers containing corrugated, wavy and curved flow channels or channels with variable cross sectional areas are used to enhance the heat transfer rate. Both numerical and experimental studies have been reported to investigate the interaction between the flow behaviour and the resulting thermodynamics within heat exchangers in order to optimize and design better units for specific applications.

Heat exchangers are classified on the basis of the transfer process, number of the fluids used, surface compactness, flow arrangements, heat transfer mechanism and construction.

1.1 Corrugation in heat exchangers

Plate exchangers with corrugated walls provide a large surface area to volume ratio and enhanced heat transfer coefficients, while allowing ease of inspection and cleaning. Such types of exchangers, like the herringbone or the chevron type, are being rapidly adapted by food, chemical and refrigeration process industries replacing shell-and-tube exchangers. Corrugated surface geometry is one of the many suitable passive techniques to enhance the heat transfer in heat exchangers. Corrugated plates are used as a turbulence promoter to enhance heat transfer. The induced turbulence is produced by the plate pattern because the fluid flows in narrow streams with many abrupt changes in direction and velocities. This turbulence, created by the shape of the plate pattern, reduces the liquid film resistance to heat transfer more efficiently than turbulence created by high flow rates and pressures in conventional exchangers. When the fluid flows through the corrugated channels, it undergoes breaking and de establishment of thermal boundary layer. The problem of viscous flow in wavy channels was first treated analytically by Burns and Parks [1] in the early seventies, who expressed the stream function as a Fourier series under the assumption of Stokes flow, followed by Sherony and Solbrig [2] in the year 1969, who studied about the heat transfer andfriction factor in a corrugated duct heat exchanger.

Niu and Zhang[13] showed the effects of aspect ratio and bending ratio of ducts on the friction coefficients and heat transfer coefficients and uniform wall temperature boundary condition. Naphon [26] studied the effects of geometry configuration of wavy plates, wavy plate arrangements, and air flow rates on the temperature and flow developments. Islamoglu and Parmaksizoglu [17,18] found in their investigation that increase of channel height gave rise to a substantial increaseboth in the fully developed Nusselt number and the friction factor but performance considering flow area goodness factor slightly decreased.

1.2 Different parameters affecting the heat transfer in a corrugated channel

The parameters influencing the heat transfer and flow characteristics in a corrugated channel can be categorised in two categories viz. geometrical and flow parameters. Designs of heat exchangers are developed by determining the increase in heat transfer rate by changing these parameters. An optimal design should also have minimum pressure drop across the channel. The geometrical parameters affecting the performances of heat exchanger includes chevron angle, geometrical grooved shapes, corrugation angle, phase shift of wave angles, aspect ratio and pitch to height ratio. The flow propertiesinclude the mass flow rate, Reynold number and Prandtl number of the fluid.

2. NUMERICAL STUDY ON CORRUGATED CHANNELS

Computational fluid dynamics is a branch of fluid dynamics which employs numerical methods for solving the equations of fluid flow which are impossible to solve analytically due to complex nature. In this present work we analyse the flow and heat transfer in a sinusoidal plate which is shown in the Fig. above. The plate has a pitch of 75 mm. The plate has a wavy height of 20 mm. ANSYS Finite Element Software with standard (k-ε) turbulence model was used to solve the problem. Second-order upwind scheme and structured uniform grid system are used to discretize the main governing equations. The fluid used in the analysis is air with a density of 1.225 kg/m^3. The air with an inlet temperature of 350 K is passed through the channel between the plates and in the process it transfers heat to the plates which are at room temperature. This paper shows contours of temperature for plates arranged in phase and out of phase and also comparison of Nusselt number, pressure difference and skin friction coefficient for different mass flow rate of 0.002 kg/s, 0.009 kg/s, 0.013 kg/s and 0.05 kg/s.

2.1 Mathematical model

2.1.1. Standard k-ε model

The standard model in ANSYS Fluent has become the essential tool of practical engineering flow calculations in the time since it was proposed by Launder and Spalding. The standard k-ε model is a model based on model transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε). The model transport equation for k is derived from the exact equation, while the model transport equation for ε was obtained using physical reasoning and bears little resemblance to its mathematically exact counterpart. In the derivation of the k-ε model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible. The standard k-ε model is therefore valid only for fully turbulent flows.

2.1.2. Transport equations for standard *k-ε* model

The turbulence kinetic energy, k, and its rate of dissipation, ε, are obtained from the following transport equations:

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}[(\mu+\frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_j}]+G_k+G_b-\rho\varepsilon-Y_M+S_k$$

And

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}[(\mu+\frac{\mu_j}{\sigma_k})\frac{\partial\varepsilon}{\partial x_j}]+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{K}+S_\varepsilon$$

In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients. G_b is the generation of turbulence kinetic energy due to buoyancy. $Y_M$is the contribution of the fluctuatingdilatation in compressible turbulence to the overall dissipation rate.

2.1.3. Modelling the turbulent viscosity model

The turbulent (or eddy) viscosity, $\mu_{t,}$ is computed by combining k and ε as follows:

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon}$$

2.1.4. Constants

$C_{1\varepsilon}=1.44, C_{2\varepsilon}=1.92, C_\mu=0.09, \sigma_k=1.0, \sigma_\varepsilon=1.3$ These default values have been determined from experiments for fundamental turbulent flows including frequently encountered shear flows like boundary layers, mixing layers and jets as well as for decaying isotropic grid turbulence. It has been found to work fairly well for a wide range of wall-bounded and free shear flows.

3. GOVERNING EQUATIONS

Continuity equation:

$$\frac{\partial\rho}{\partial t}+\frac{\partial}{\partial x_i}\left(\rho u_j\right)=0$$

Momentum equation:

$$\frac{\partial}{\partial t}\left(\rho u_i\right)+\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=\frac{\partial}{\partial x_j}\left[-p\delta_{ij}+\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right]+\rho g_i$$

Energy equation:

$$\frac{\partial}{\partial t}\left(\rho C_a T\right)+\frac{\partial}{\partial x_j}\left(\rho u_j C_a T\right)-\frac{\partial}{\partial x_j}\left(\lambda\frac{\partial T}{\partial x_j}\right)=s_T$$

Heat transfer coefficient:

$$h=\frac{Q_{cycle}}{(T_b-T_w)A_{cycle}}$$

Nusselt number:

$$Nu = \frac{hD_h}{k}$$

Reynold number

$$Re = \frac{VD_h}{\upsilon}$$

Friction factor:

$$f = \frac{-\frac{dP}{dX}D_h}{\frac{1}{2}\rho V^2}$$

4. RESULTS AND DISCUSSION

This paper shows contours of temperature for plates arranged in phase and out of phase and also comparison of nusselt number, pressure difference and skin friction coefficient for different mass flow rate of 0.002 kg/s, 0.009kg/s, 0.013 kg/s and 0.05 kg/s.

The variation of the temperature contours with air flow rate for different wavy plate arrangements have been shown in the figure1. The contour is for 40 mm and 20 mm channel in which effect of phase shift is shown for 20 mm channel width. From the contours it is seen that as the mass flow rate is increased from 0.002 kg/s to 0.05 kg/s, the heat flow through the channel also increases.

Fig 2, 3 and 4 shows the variation of Nusselt number, pressure drop skin friction for different mass flow rate along a channel of 20 mm channel height phase shift while Fig 5, 6 and 7 shows it for same phase channel with channel height of 20 mm. The local Nusselt number is higher in the converging section of each wave than in the diverging section (furrow).This is because the converging section has a higher average velocity and velocity gradient which increases the heat transfer ratio.

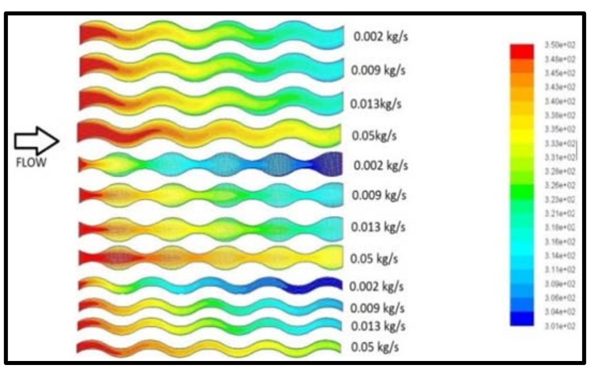

Fig. 1: Variation of temperature contour with air flow rate for different wavy plate

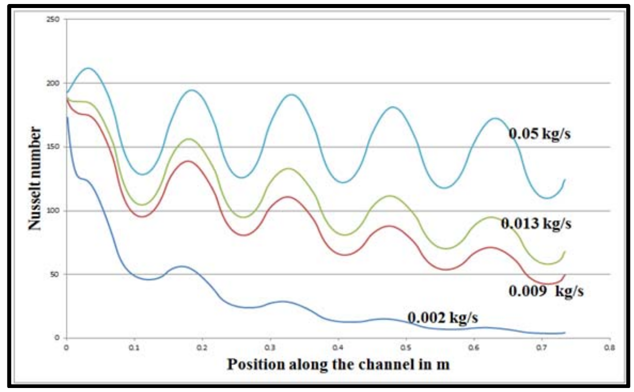

Fig. 2: Variation of nusselt number for different mass flow rate for channel of phase shift

Conversely, the flow has a low velocity gradient near the wall surface in each furrow, which decreases the heat transfer ratio. As the axial coordinate increases, the amplitude of the average Nusselt number decreases. This is to be expected since the average Nusselt number is obtained by integrating the local Nusseltnumber distribution over the area.

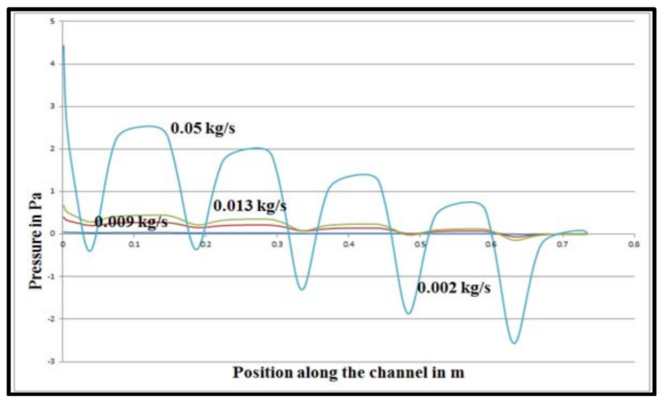

Fig. 3: Variation of pressure drop along the channel of 20 mm same phase

The average Nusselt number is constant after few waves, i.e. as soon as the flow becomes periodically fully developed.

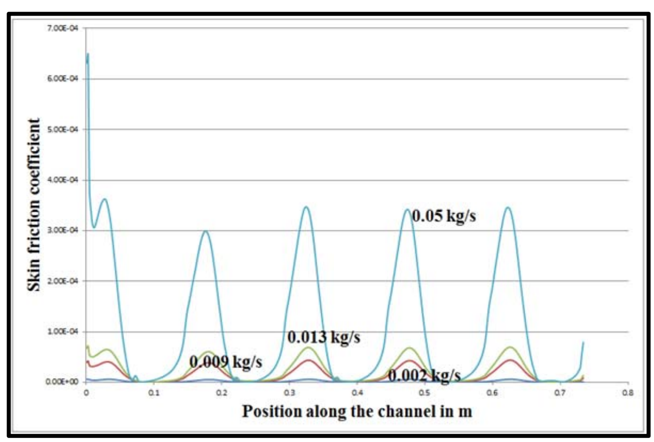

Fig. 4: Variation of skin friction coefficient along the channel of 20 mm phase shift

The skin friction coefficient for the phase shift as shown in fig 3 attains its maximum and minimum values in the minimum and maximum cross sectional area. While fig 6 shows that for channel with same phase the wave for skin friction coefficient suffers increase in amplitude in the ridge region.

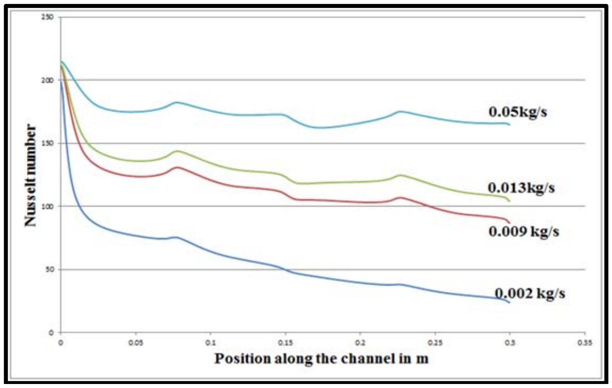

Fig. 5: Variation of nusselt number along the channel of 20 mm same phase

Fig 2 shows that the pressure drop decreases in the region with minimum crosssection because in these regions the velocity of flow increases. As the axial coordinate increases the amplitude of pressure drop is decreased. It is seen that the Nusselt number pressure drop and skin friction coefficient decreases with the increase in mass flow rate.

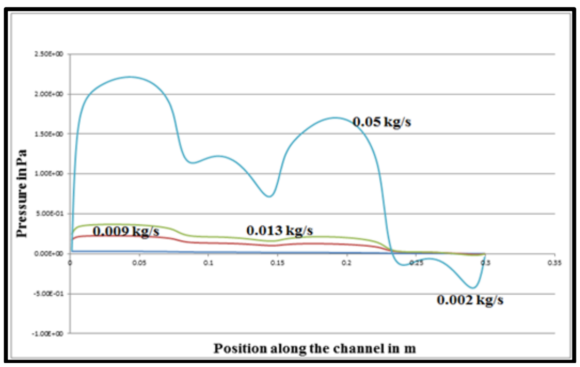

Fig. 6: Variation of pressure drop along the channel of 20 mm same phase

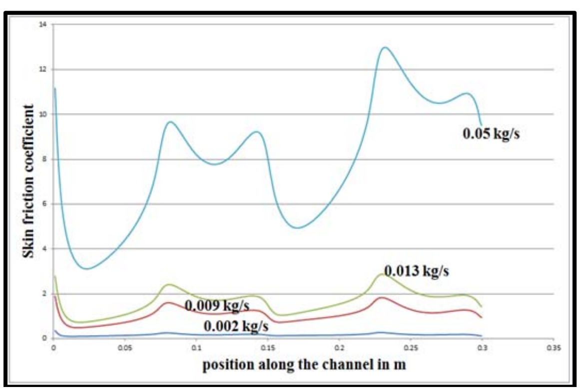

Fig. 7: Variation of skin friction coefficient along a 20 mm channel with same phase

5. CONCLUSION

Forced convection for flow through a periodic array of a wavy-wall channel has been investigated numerically. The characteristics of fluid flow and heat transfer in periodic fully developed region of the corrugated duct were obtained numerically using the finite element software in this study. Nusselt number, Skin friction coefficient and pressure drop increases with the increase of mass flow rate i.e higher for the mass flow rate of 0.05 kg/s and lower for mass flow rate of 0.002 kg/s. For the phase shift plate, the harmonic curves for the local skin-friction coefficient and the local Nusselt number have the same frequency as that of the wavy surface at lower Reynolds numbers. Nusselt number is found to be as high as 230 at the inlet for the same phase and at the portion with minimum cross section for the phase shift. For the plate with same phase the Nusselt number and friction coefficient have higher value at the ridge and lower at the furrow. The pressure drop for the phase shift increases in the increasing cross section and vice versa. While in the same phase channel, pressure drop increases in the furrow region.

6. ACKNOWLEDGEMENT

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