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INVESTIGATIONAL ANALYSIS OF HEAT SINK AND ITS COMPARISON WITH COMMERCIALLY AVAILABLE HEAT SINK

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ABSTRACT :- In the current work, a numerical exploration is made on the flow characteristics and improvement of heat transfer in 2D channel with wavy wall covering a broad range of Reynolds numbers. For enhanced understanding, the numerical analysis is carried out by considering three different wall-geometry (triangular, sinusoidal, and trapezoidal corrugated wall). The outcomes are examined by drawing graph of the wall-Nusselt number along the channel length by varying the operating parameters like Reynolds number, amplitude of geometry, heat flux. The flow characteristics (such as pressure, temperature, velocity) deviation down the channel are also being examined to encapsulate the hydrodynamics. The two modes of boundary conditions employed are constant heat flux and constant wall temperature at the channel-wall, at the inlet velocity is completely stated, and atmospheric pressure is specified at outlet. The fluid used for the simulation is water. In this analysis, it is perceived that with the increase in the geometry amplitude and Reynolds number there is significant enhancement in heat transfer. It is obtained from the analysis that the heat transfer rate is maximum with triangular channel and the pressure drop is minimum with triangular channel.

Keywords—Heat transfer enhancement, corrugated wall, Wall-Nusselt number, trapezoidal, sinusoidal, triangular

1.Introduction

Heat transfer augmentation with nominal pressure drop by any heat exchanging devices is extremely important phenomena within the thermal engineering [5]. It has a bunch of application in Heat exchangers, method industries, Evaporators, Condenser, Thermal power plants, Air-conditioning systems etc. Overheating is the major problem associated with any power plants that triggers the failure of the system and the efficiency of the system is also reduced due to loss of heat in various forms. To overcome this problem effective cooling is required for which a heat exchanger is employed [2]. A heat exchanger is a device that transfer heat from hot fluid to the cold fluid with maximum rate and minimum investment. Employing heat exchanger also improve the efficiency of the system for super-heater, feed hot-water heater, condenser, air pre-heater used in power plant is used to increase the efficiency of the system [1]. The two major parameters associated with heat exchanger are heat transfer rate and the pressure drop across it (if it is high then additional power would need to pump the fluid). In a heat exchanger device heat transfer takes place mainly due to convection and from newton's law of cooling for convection heat transfer depend on surface area exposed and difference of wall temperature and fluid temperature [3]. Since temperature difference can be varied only to certain limit, other ways to improve the heat transfer rate is by either varying heat transfer coefficient or to vary the area exposed in such a way that it has minimum pressure drop across it. A number of the ways to improve the warmth transfer rate are given below.

1.1 Methods to improve the Heat Transfer:

Transfer of heat in a heat exchangers takes place primarily due to convection and from newton's law of cooling for convection we all know that the heat transfer is proportional to convective heat transfer coefficient, the surface area exposed, and difference between the surface and fluid temperature [4]. The convective heat transfer coefficient is the function of fluid properties like its density, velocity, viscosity, specific heat, velocity etc.

a) ActiveMethod: This method is based on the forced convection that is an external devices like blowers, pumps, fan etc. are used to agitate the fluid. Due to which convective heat transfer coefficient increases [10]. The Figure 1.1 shows a compact machine that is cooled by axial flow fan is an example of active method.

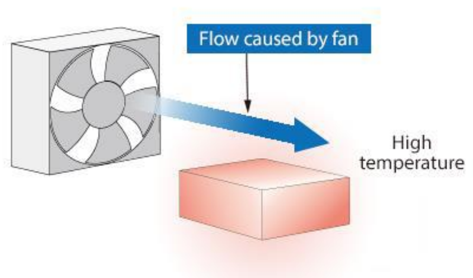


Figure 1.1: Active Method of enhancing heat transfer rate

b) Passive Method: This technique based on the surface treatment method without aid of any external power device. Various surface treatment like: imposing surface roughness on the wall, use of baffles or fins, changing the shape of the wall of pipe/channel (corrugated wall), etc. are used.

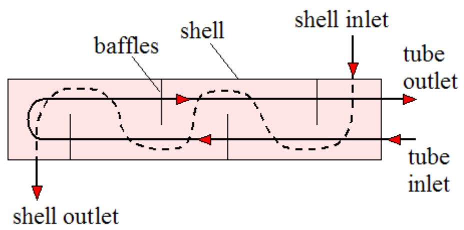


Figure 1.2: Passive Method of enhancing heat transfer rate

Figure 1.2 shows shell and tube heat exchanger using baffles. This method is very simple and does not require any further power. Therefore the cost of operation is less as compared to active method. The baffles creates barrier to the flow so that thorough mixing of the fluid takes place, which in turns increases the heat transfer rate. But the major disadvantage associated with it is that the pressure drop across it is increased and eventually more power for pumping of working fluid is required.

c) Compound Method: This method is the amalgamation of the two methods discussed above. It is clear from the Figure 1.3 that the fins are incorporated to the wall of the channel and the external device (Fan) forced the fluid in the channel.

This thesis is solely based on the passive method of heat transfer enhancement.

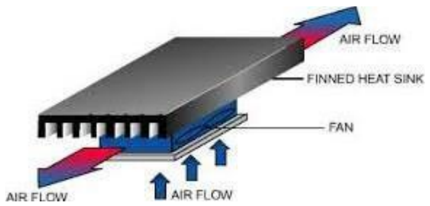


Figure 1.3: Compound Method of enhancing heat transfer

1.2 Corrugated Channel and Its Application: The term corrugated means that wavy or uneven that's, it consists series of repetitive and parallel formed wall such as: curving, triangular grooves, square, trapezoidal etc [10]. As a result of this uneven geometry, it creates the disruption within the flow and causes the reversal or recirculation of the flow. Recirculation regions at the wall boosts the blending of the fluid and diminish thermal boundary layer, which results in rise of the heat transfer rate. However the corrugated channel has major disadvantage of accumulated overall pressure loss, which ends in additional power demand to pump the fluid.

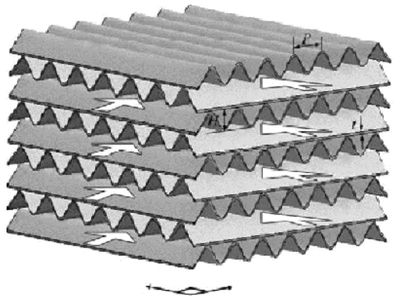
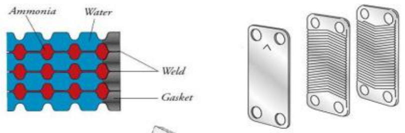


Figure 1.4: Sinusoidal Corrugated channel

Corrugated channels are being often utilized in heat exchanger devices preponderantly in plate-fin heat exchangers or plate heat exchanger [6]. As a result of its simplicity and additional surface exposure, it imposingly increase the heat transfer rate. The Figure 1.4 shows furrowed channel (sinusoidal) and Figure 1.5 shows the furrowed channel in plate device. Furrowed channel plays terribly crucial role in cooling the compact parts like portable computer and alternative little electronic parts because it improves the ratio of heat transfer rate to volume.



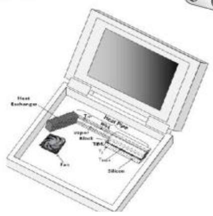


Figure 1.5: Different applications of the corrugated channels

2. LITERATURE REVIEW

Corrugated channel are being well utilized in small heat exchangers, micro heat exchangers, and lots of different heat exchangers such as: thermal power plants, condenser, air-con system, evaporators, cooling of turbine blades, radiators of vehicles and vehicles etc.

Enhancement of heat transfer rate using furrowed channels (passive method) is associate rising space of analysis currently a days. Several scientist are operating completely in this area. They indeed done numerous experimental, numerical, and analytical study with numerous forms of furrowed channels. Several of them have given their contribution in this area. For instance,

Hamza et al. [1] has done associate experimental study on the results of operational parameters on stratified forced convection. He thought of V-corrugated channel with air as an operating fluid for this experiment. For this experiment associate constant heat flux condition is applied to the upper wall of the V- furrowed channel and the lower one was insulated. The assorted variable parameter taken are

Reynolds number, temperature of air, painter variety the angle of tilt of the V-corrugated channel. And therefore the result of those parameters on the Nusselt numbers were evaluated by varying these parameters in respective ranges.

Islamoglu et al. [2] performed associate experiment to judge the friction factor and convective heat transfer coefficients of furrowed channel in a plate heat exchanger during which operating fluid used are air. He performed the experiment for 2 totally different heights of furrowed channel and single tilt angle of V- furrowed channel for different value of Reynolds number. The results displays that there's a vast increment in each the Nusselt variety and the pressure drop with the heights of the wave.

Paisarn [3] did associate experiment to review the heat transfer characteristics and pressure drop of the streamline flow through the triangular formed furrowed channel. He performed the experiment for various angle of 20°, 40°, and 60° whereas the peak being constant at 12.5 mm. He maintains a continuing heat flux through the channel and varied the Reynolds numbers within the range of 500 to 1400. And created associate observation that for higher worth of Reynolds numbers and the tilt angle, the rate of heat transfer is higher on the expense of pressure drop.

Mohammed et al. [4] studied numerically the fluid flow and therefore the stratified forced convective heat transfer characteristics. He performed the simulation on V-furrowed channel having the worth of Prandtl number as 0.71. For his studies he varied the Reynolds numbers from 500 to 2500 and the angle of V-corrugated channel from 00 (straight pipe) to 600.

Pethkool et al. [5] performed associate experiment on helical furrowed tube and examined the convective heat transfer with flow. He collected the result for various pitch to diameter ratio and Reynolds numbers and observed a rise within the convective heat transfer of 1.23 to 2.32 times than that of sleek pipe, reckoning on the rib height of the helical tube. Friction factor conjointly will increase within the range of 1.46 to 1.93 times of the sleek tube.

Yin et al. [6] investigated numerically the flow and also the convective heat transfer characteristics of curving furrowed channel by varying the phase angle between the higher and lower furrowed wall. He meted out the simulation for constant wall temperature condition at the wall and periodic condition at recess of the channel for different Reynolds numbers. The results shows that by increasing the phase angle, The shear stress, friction factor and the average Nusselt number decreases linearly. It absolutely was concluded within the paper that higher overall performance is extracted by the channel having part shift angle of 0o and 90o. The best result was obtained by the channel having part shift angle of 0o.

Ozbolat, V. et al. [8] coherently describes the heat transfer improvement and flow characteristics in two-dimensional furrowed (sinusoidal and square) channel. During this work the comparison has been created between completely different shapes of furrowed geometries with different Reynolds numbers and different wall temperatures boundary conditions. This work additionally provides completely different rate and temperature contours for various conditions. This work also compare the Nusselt number variation among the straight, curving and square furrowed channel and eventually they reported that curving furrowed channel provides higher results than different two shapes.

Pehlivan et al. [9] performed experiment by taking three differing kinds triangular furrowed surfaces. Two completely different channel height was taken and angle of tilt was varied. The experiment was performed for the various Reynolds numbers keeping the heat flux constant. This paper conclude that converging-diverging furrowed channel has comparatively higher result than that of furrowed channel having same phase angle and straight channel.

Jafari et al. [10] numerically studied the pulsating flow with forced convection through curving furrowed channel and its effects on the convective heat transfer. The results were obtained by varying the frequency of the pulsating flow for different Reynolds numbers. The results showed the linear relationship between the oscillatory rate amplitude and convective heat transfer rate.

Benzenine et al. [11] numerically studied the channel with transverse waved baffles for the turbulent flow. During this paper the inclination angle of baffles was varied and its effects on the skin friction were studied. The results showed that by incorporating the baffles with 15o inclination angle improve the pressure loss by 9.91% compare to the straight baffles.

Mohammed et al. [12] meted out the numerical analysis on furrowed channel employed in the plate heat exchangers to analyze the convective heat transfer of this forced flow. The simulation was performed by varying the tilt angle, amplitude and Reynolds number. Finally from the results the optimum specification of the channel for better convective heat transfer, pressure drop and compactness has been found that are height of wavy channel 2.5 mm, 60o angle of tilt, and 17.5 millimeter height of channel.

Ahmed et al. [13] performed the numerical investigation on Nano fluid of water and Cu-O with completely different volume fractions and Reynolds numbers for straight, triangular, curving and quadrangle furrowed channel. From the result it had been all over that for higher volume fraction and Reynolds number higher is the heat transfer rate and the pressure drop and vice-versa. This paper additionally conclude that for same operating parameters the heat transfer rate for the quadrangle furrowed channel was higher than alternative channel.

Maryam et al. [17] performed the big eddy simulation on turbulent flow. Additionally detected convective heat

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transfer for half furrowed channel. The amplitude of furrowed channel and Reynolds number has been modified. The special result from this paper got was the reattachment and separation regions are strongly affected from the amplitude of channel. More precisely the attachment region is more affected from the amplitude than that of the separation region. Additionally the thermal boundary layer is thickest at the separation region and it's thinnest at the reattachment region in order that the convective heat transfer is most at reattachment region and lowest at the separation region.

Umavathi, J. C. et al. [14] numerically studied the mixed convective flow of two viscous fluid that are immiscible within the vertical furrowed channel. The simulation had been performed by them by varying the parameters like viscosity ratio, Grashof variety, thermal conductivity ratio, width ratio, frequency of the flow. The results of this parameter on the temperature, shear stress, speed and the Nusselt variety were also been studied.

3. PROBLEM FORMULATION

3.1 ASSUMPTION FOR THE SIMULATION

The 2D incompressible steady flow of water (Newtonian fluid) is taken into account here. The fluid taken into account is assumed to be in single phase once it's flowing through the channel. The radiation heat transfer compare to the convection heat transfer is neglected here. The thermo-physical properties of the fluid are taken as constant.

3.2 CASES PERFORMED UNDER PRESENT PROBLEM

The problem is solved by considering different boundary conditions and different geometrical shapes that are described below.

3.2.1 Fully developed streamline flow with constant wall heat flux boundary condition: The fully developed streamline flow is taken at the water of the channel. Inlet temperature of the water is taken as 300 K. The constant heat flux with no slip and no penetration boundary condition is applied at the wall of the channel. Different values of heat flux taken are 50 kW/m2, 100 kW/m2, and 150 kW/m2 and200 kW/m2. Three different furrowed channel with varying amplitude are taken into account that are represented below.

3.2.1.1 Sinusoidal geometry: The amplitude of this channel is varied as 1.75 mm, 3.50 mm and 7.00 mm. The specifications of the channel are shown in Table 3.1.

Amplitude (a) in mm	Wavelength (L_w) in mm	Minimum space (H_{min}) in mm	Maximum space (H_{max}) in mm
1.75	28	6	13
3.50	28	6	20
7.00	28	6	34

Table 3.1: Overall specifications of the channel with totally different amplitudes

With these amplitudes and constant heat flux BC, the problem are being solved for four different Reynolds number: 500, 1000, 1500 and 1900. The geometry for sinusoidal channel is shown in Figure 3.1. The profile equation for curving wall profile is taken as:

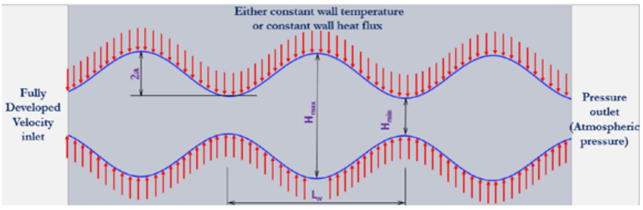


Figure 3.1: Schematic representation of the sinusoidal corrugated channel

3.2.1.2 Trapezoidal geometry: The schematic diagram of trapezoidal profile is shown Figure 3.2. This case studied by considering amplitudes as 1.75 mm, 3.50 mm and 7.00 millimeter (See Table 3.1). With these amplitudes and constant heat flux BC, the problem are solves for four totally different Reynolds number: 500, 1000, 1500 and 1900.

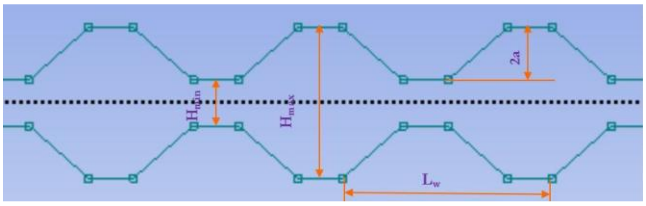


Figure 3.2: Computational domain of the Trapezoidal corrugated channel

3.2.1.3 Triangular geometry: The schematic diagram of triangular profile is shown in the Figure 3.3. Additional the amplitude of this channel is varied as 1.75 mm, 3.50 mm and 7.00 millimeter (See Table 3.1). With these amplitudes and constant heat flux BC, the problem are solved for four totally different Reynolds number: 500, 1000, 1500 and 1900.

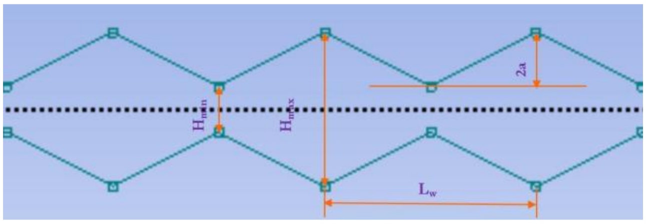


Figure 3.3 Computational domain of the triangular corrugated channel

3.2.2 Fully developed streamline flow with constant wall temperature BC: The totally develop streamline flow (given by using User outlined function) is taken at the inlet of the channel. Inlet temperature of the water is taken as 363 K. The constant wall temperature with no slip and no penetration condition is applied at the wall of the channel. Three totally different values of the wall temperature (365 K, 367 K, and 369.25 K) are considered. Three totally different furrowed channel and their amplitude considered are represented below.

4. NUMERICAL CALCULATIONS

4.1 GRID PATTERN EMPLOYED

After an extensive verification quadrilateral grids are employed for the present study. To capture the wall effect as well as to save the computation time, finer grid are selected near wall and coarser at the middle portion of the channel. The mesh employed for the sinusoidal geometry is shown in the Figure 4.1. For trapezoidal and triangular geometries, the mesh are shown in the Figure 4.2 and Figure 4.3 respectively.

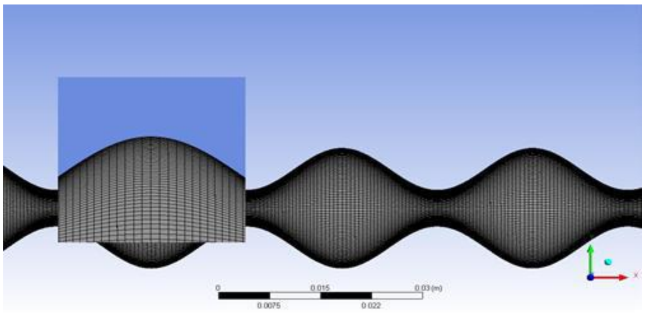


Figure 4.1 Pattern of grid for sinusoidal corrugated channel

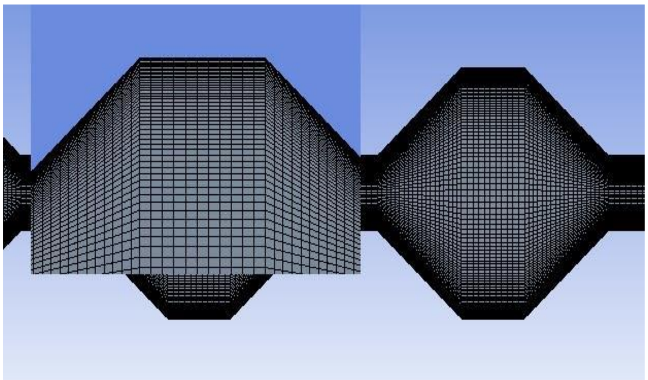


Figure 4.2 Pattern of grid for trapezoidal corrugated channel

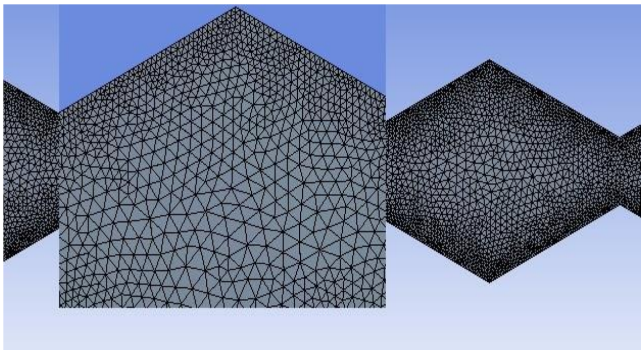


Figure 4.3 Pattern of grid for triangular corrugated channel

4.2 GOVERNING EQUATIONS

The flow characteristics are govern by continuity and Navier-Stokes or momentum equation. The heat transfer characteristics are govern by the energy equations. The flow is steady, two dimensional and incompressible flow. The governing equations are shown below.

Continuity Equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad \text{(i)}$$

Navier-Stokes Equation:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \qquad \text{(ii)}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \qquad \text{(iii)}$$

Energy Equation:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \qquad \text{(iv)}$$

Where $\alpha = \frac{k}{\rho} c_v$ Turbulence presence in the domain has been modelled using standard k-ε model:

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

$$\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x_i}(\rho \varepsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\epsilon}{k}(G_k+C_{3\epsilon}G_b)-C_{2\varepsilon}\rho\frac{\epsilon^2}{k}+S_\varepsilon \qquad \text{(vi)}$$

5. RESULTS AND DISCUSSION

5.1 GRID INDEPENDENCE TEST

The mesh size and its quality affects accuracy and the computation time strongly. So the Grid independence test is carried out to choose the optimum grid size. The Grid independence test is performed by plotting surface Nusselt number along channel wall considering grid distribution as, 20×300, 50×300, 50×600, 100×300, 100×600, 150×600, 200×600 and 200×700 on the sinusoidal corrugated channel.

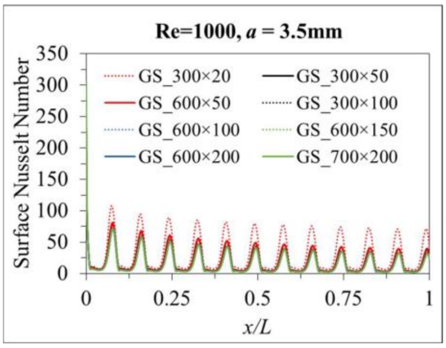


Figure 5.1 Grid independence test for surface

For this test, the boundary condition at wall is taken as constant wall temperature (369 K) and at inlet, is taken as 'velocity inlet' with inlet water temperature of 367 K. At inlet fully developed velocity profile is used. Figure 5.1 shows the variation of surface Nusselt number along the corrugated wall of the channel with different grid sizes. It is found that when the number of element is more than 100×600, the variation in the surface Nusselt number with grid is very negligible. It is also confirmed from the Figure 5.2. Figure 5.2 shows a plot of average Nusselt number along the length of corrugated sinusoidal channel with different grid sizes. After an exhaustive verification of the results for different grid sizes and grid element, the mapped quadrilateral grid with 100 division along width and 600 division along length is considered for the present problem.

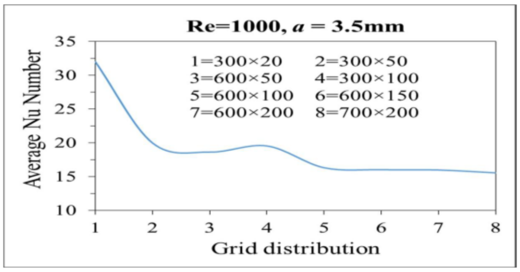


Figure 5.2 Grid independence test: Variation of average surface Nusselt number

5.2 VALIDATION

The validation of the present methodology is performed with Ozbolat et al. Validation performed with this work with the help of plot of surface Nusselt number along the wall of channel. The validation is shown on Figure 5.3. The result obtained are almost matching with the paper results.

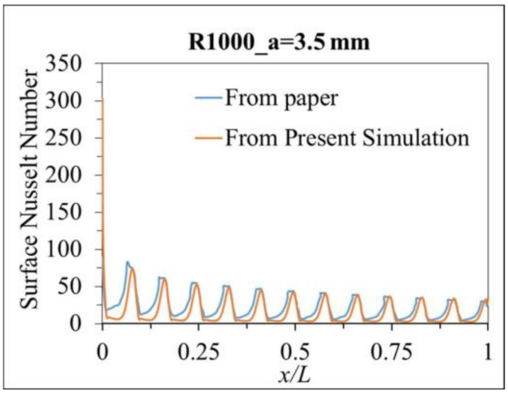


Figure 5.3: Validation of surface Nusselt number along the sinusoidal corrugated channel with “Ozbolat, V. et al” research work

5.3 LAMINAR FLOW WITH CONSTANT WALL HEAT FLUX BOUNDARY CONDITION

Results for the present case are shown below categorically. Effect of *Re*, heat flux, geometry profile and the value of amplitude considering present case are shown.

5.3.1 Effects of the Reynolds Numbers: In this particular case, effects of Reynolds number on the surface Nusselt number for different heat flux are shown in Figure 5.4 to 5.7. The sinusoidal corrugated channel with amplitude (a =1.75 mm) is taken. The result shows that surface Nusselt number increases as the Reynolds number increasing. This may due to the reason that more recirculation is created consequently more mixing occurred as the Reynolds number increases.

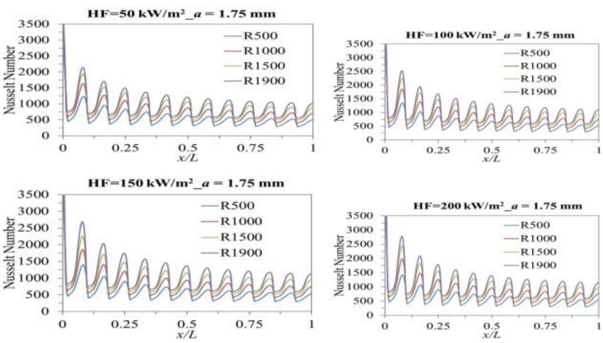


Figure 5.4. Effect of Reynolds number on Surface

The results of these study. For all the cases in these Figures, the amplitude is fixed as a = 1.75 mm and Reynolds number is fixed as 500.

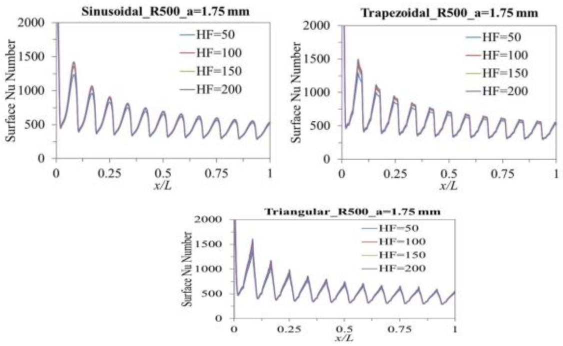
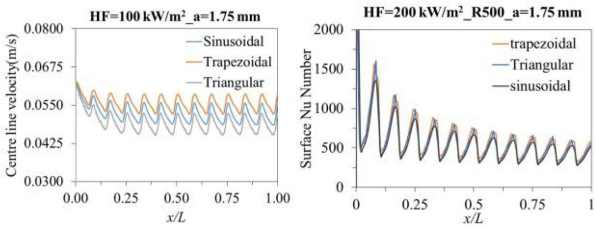


Figure 5.10: Effect of HF on surface

5.3.3 Effects of the Geometry Profile: The Figure 5.11 shows the effects of the geometry profile on the velocity along center line. Effect of geometry of the surface Nusselt number are shown on Figure 5.12. Effect of geometry on the outlet fluid temperature is shown on Figure 5.13. From the Figure 5.12 it is observed that Nusselt number variation along the length is not much affected by the shape of the geometry. It is also observed that fluid flowing in the trapezoidal corrugated channel have the highest outlet temperature. It is lowest in the triangular corrugated channel.



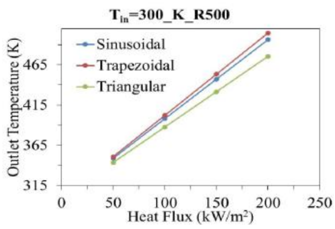


Figure 5.13: Effect of geometry profile on outlet temperature

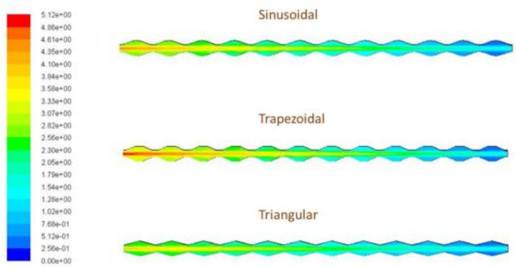
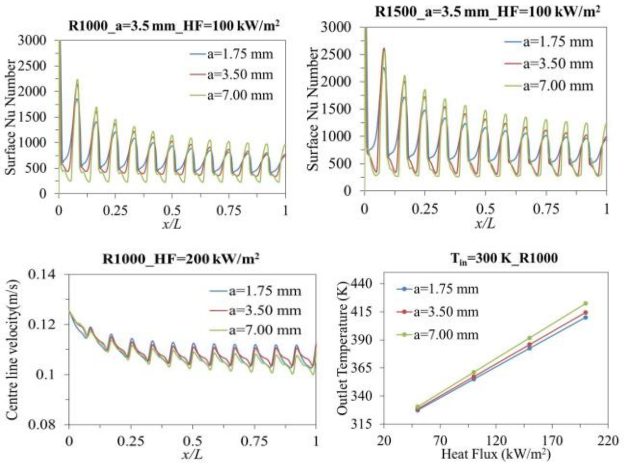


Figure 5.14: Effect of the geometry profile on the total pressure

5.3.4 Effects of the Amplitude of Geometry: The Figure 5.15 and.5.16 shows the effects of amplitude on the surface Nusselt number plot considering channel with sinusoidal geometry profile. With the increment in amplitude, the average gap between the two channel increases and consequently the recirculation of the flow increases and ultimately the Nusselt number also increases.



Effects of amplitude of sinusoidal profile on the center line velocity and outlet temperature are shown in Figure 5.17 and 5.18 respectively. It is observed that outlet temperature increases with the amplitude. Because higher is the amplitude more is the mixing of flow and more is the out temperature.

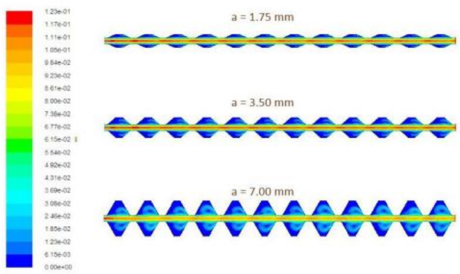


Figure 5.19: Effect of amplitude of trapezoidal geometry on velocity contour.Heat flux=200 kW/m2 and Re = 1000.

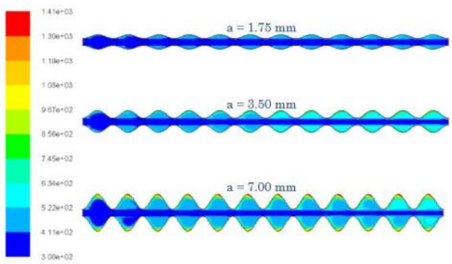


Figure 5.20: Effect of amplitude of sinusoidal geometry on temperature-distribution

5.4 LAMINAR FLOW WITH CONSTANT WALL TEMPERATURE BC

Results for the present case are shown below categorically. Effect of *Re*, wall temperature, geometry profile and the value of amplitude considering present case are shown.

5.4.1 Effects of the Reynolds Number: Figure 5.21 and 5.22 shows the effects of Reynolds number on surface Nusselt number along the channel and outlet temperature respectively. From this Figure also it is clear that surface Nusselt number increases as the Reynolds number increases and outlet temperature decreases with increase in Reynolds number.

6. CONCLUSIONS & FUTURE WORK

6.1 CONCLUSION

The Numerical simulation is performed on the sinusoidal, trapezoidal and the triangular corrugated channel with water as working fluid. At the wall, boundary condition are taken as constant wall heat flux or constant wall temperature with no slip and no penetration boundary condition. The simulations cover a wide range of the Reynolds numbers, heat fluxes and wall temperatures. At the inlet, velocity is specified by using the user defined function. For each of the geometry profiles, its amplitude has been varied as 1.75 mm, 3.50 mm and 7.00 mm. From the present study some important findings have been observed which are given below.

Heat transfer enhancement is maximum for triangular corrugated channel followed by trapezoidal and triangular channel for laminar flow.

In case of turbulent flow, the triangular corrugated channel shows the better heat transfer rate than others two corrugated channel.

Pressure loss is maximum in case of trapezoidal corrugated channel followed by sinusoidal and triangular channel for laminar flow.

For higher Reynolds number the surface-Nusselt number is higher. On increasing the amplitude of wavy wall, the Nusselt number increases.

For laminar flow, it is observed that the problem is not converged when *Re* and channel amplitude exceeds certain value.

With increment of *Re*, outlet temperature decreases. For laminar flow, reduction in the center line velocity is minimum in trapezoidal corrugated channel.

6.2 FUTURE SCOPE

The 3-D simulation of the present work can be made in the future. It is required to continue this study extensively by considering other possible geometries. A hybrid scheme can be develop in future for automatic optimization of the geometry profile in order to get maximum heat transfer rate with minimum pressure loss.

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