NUMERICAL STUDY OF HEAT TRANSFER CHARACTERISTICS OF A CHANNEL WITH TRIANGULAR WALL

A Thesis submitted in partial fulfilment of the requirements for the Degree of

Bachelor of Technology

In

Mechanical Engineering

By

Hitesh Kumar Biswal

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NATIONAL INSTITUTE OF TECHNOLOGY ROURKELA 769008, INDIA

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Under the Guidance of

Dr. Amitesh Kumar

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CERTIFICATE

This is to certify that the work in this thesis entitled "***Numerical Study of Heat Transfer Characteristics of A Channel With Triangular Wall***" by Hitesh Kumar Biswal, has been carried out under my supervision in partial fulfilment of the requirements for the degree of Bachelor of Technology in Mechanical Engineering during session 2014-2015 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.

To the best of my knowledge, this work has not been submitted to any other University/Institute for the award of any degree or diploma.

Date:

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DECLARATION

I hereby declare that this thesis is my work and effort. Throughout this documentation wherever contributions of others are involved, every endeavour was made to acknowledge this clearly with due reference to literature. This work is being submitted for meeting the partial fulfilment for the degree of Bachelor of Technology in Mechanical Engineering at **National Institute of Technology, Rourkela** for the academic session **2011 – 2015.**

Hitesh Kumar Biswal

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ABSTRACT

The heat transfer characteristics of the channel are numerically investigated using ANSYS 15.0. The top-wall is made wavy with triangular in shape. Four different geometries were taken varying the number of waves on the top-wall. The length and width of the channel is 1000mm and 20mm respectively. The amplitude of the wave is 5mm. The numbers of waves are taken as 2, 4, 6 and 8. The inlet is kept as velocity-inlet with inlet velocity of 0.05m/s and temperature of 300K. The top-wall is set as wall and a constant temperature of 350K is set. The bottom-wall is kept as wall and a constant temperature of 300K is set. The outlet is set as pressure-outlet. The contours of velocity and temperature are investigated. The variation of Local Nusselt number with length of the channel is studied by extracting the values of wall flux on top-wall, top-wall temperature and fluid temperature from the solution. It is observed that as the numbers of waves increase, the average Nusselt number increases, thus enhances the convective heat transfer.

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CHAPTER 1: INTRODUCTION

The study of heat transfer is concerned with the computation of the heat flow rate across an interface, within a medium, or from one surface boundary to another, and the related statistical distribution of temperature. The first 3 sections describe the different ways of heat transfer:

1.1 Conduction:

It is defined as the transfer of internal energy between two bodies, or parts of the same body by molecular diffusion. Steady-state conduction is the conduction that occurs when the driving temperatures remain constant. It means the distribution of temperature with respect to space i.e. temperature field in the conducting body does not vary any further. In steady-state conduction, the heat energy entering any surface of a body is always same as the heat energy leaving out of it.

1.2. Convection:

It is defined as the heat transfer through mass movement of fluids. It involves conduction i.e. heat diffusion as well as advection i.e. heat transfer by bulk movement of fluids. The causes of convective heat transfer are either natural or forced called as natural convection and forced convection respectively. The reason behind natural convection is temperature differences. This temperature difference changes the density and thereby buoyancy of the fluid. The buoyancy of the fluid is also varied due to gravitational field and this convection is called gravitational convection. Forced convection is the consequences of the fluid movements by an external force e.g. a pump or a fan.

1.3. Radiation:

It can be defined as the energy emitted by bodies by virtue of their own temperatures which is the consequence of molecular thermal excitation.

As the heat transfer by radiation is very small compared to that of conduction and convection, it is not taken into account while calculating the heat transfer from a surface. The heat transfer through conduction and convection can be compared by computing the Nusselt number on the surface.

1.4. Nusselt number: It is defined as the ratio of convective heat transfer to conductive heat transfer across the boundary. For laminar flow it is closed to one but for turbulent flow its value varies from 100 to 1000.

In present day situation the enhancement of heat transfer is a significant worry as it has a wide range of industrial as well as household applications such as air-conditioners, refrigerators etc. So, to design a better and efficient thermal device we need to improve the heat transfer. A numerous methods have been studied and developed to further improve the characteristics of hear transfer. Extensive research has been done to upgrade the heat transfer rate. Earlier fins or extended surfaces were provided on thermal devices to do so but to make the device compact people have tried different geometries to design a device without fins like corrugated channels, grooved channels, symmetric and asymmetric wavy channels etc. as well as different arrangements in geometry have been tried such as channel extension, rear facing steps etc.

Passive and active methods for heat transfer enhancement have been investigated and developed to increase heat transfer processes by changing the characteristics of flow. Substantial attempts have been devoted to the role of mixing and destabilization of flow in enhancement of heat transfer. For example, modulation of active flow has shown to be successful in bringing out vibrant heat transfer improvement. Experiments carried out with grooved, wavy and communicating channels have presented that self-sustained oscillations that is developed in these geometries by flow instabilities excitement, lead to enhancement of heat transfer without any active forcing. It is observed that the heat transfer enhancement varies and it depends on boundary conditions and characteristics of specific geometry. Symmetric and asymmetric wavy channels are examined in various earlier studied to improve mass and heat transfer rates in compact heat exchange devices e.g. oxygenators and heat exchangers. These channels are manufactured without difficulty and can give sufficient heat transfer upgrade if employed in a suitable range of Reynolds number.

Heat transfer enhancement in laminar flow is also a major concern as it has an extensive practice in solar heat collectors, electronic devices, and design of compact heat exchanger and processing of micro-channel. This necessity has motivated scientists and researchers to carry out numerous experiments on laminar flow in different topological shapes including those with circular and circular sector, triangular, rhombic, trapezoidal and semi-elliptic cross-sections.

The present work analyses the channel with upper wall having geometry like triangular waves. The contours of velocity and temperature are studied. The variation of local Nusselt number with length of the channel is investigated. Initially the upper wall is having two triangular waves. The number of waves is varied to four, six and eight and the effect of the variation is studied. For validation of this work, two dimensional analysis of a circular tube is done. The dimensions of the circular tube are kept same as before. The local Nusselt number versus length of the channel is plotted and compared to the experimental values.

CHAPTER 2: LITERATURE SURVEY

Kuhn and Rohr [1] studied to direct mixed convection from a heated wavy surface. The channel flow between the sinusoidal surface and a flat top wall is investigated by means of a combined DPIV and PLIF method to inspect the spatial changes with space of the stream wise and the components of velocity normal to wall, and to evaluate the field conc. of a tracer dye injected into the fluid. They found that the transport properties are additionally enhanced compared to mixed convection from a flat plate by the presence of the wavy surface.

Amador et al. [2] found that heat transfer enhancement can be obtained without the requisite of greater volumetric flow rates in fluid with turbulent flow, in which much more pumping powers are required. When the channel with asymmetric wall is operated in a suitable Reynolds's number range, a substantial upgrade in heat transfer is noticed.

Sui et al. [3] studied heat transfer and laminar-fluid flow in 3-D wavy micro channel. It can be observed that the location and the quantity of the vortices may vary along the direction of flow, which leads to disorganized advection, so the convective fluid mixing can be considerably upgraded, and thus the performance of heat transfer of the current wavy micro channels is compared to straight micro channels with the same cross section the wavy one gives the better results.

Lee et al. [4] observed in his experiment of fully-developed flow and heat transfer in intermittent wavy channels with rectangular cross areas are contemplated utilizing direct numerical reproduction, for expanding Reynolds numbers crossing from the unfaltering laminar to transitional stream administrations. It is found that because of the effective blending in wavy channels, the heat transfer execution is fundamentally more better than that of straight channels with the same cross segments; in the meantime the penalty in pressure drops of wavy channels can be much littler than the heat transfer upgrade.

Ramagadia and Saha [5] numerically simulated fluid flow and heat transfer through a sinusoidal wavy channel with surface described by a sine wave function. Finally the thermal performance at a Reynolds number of 600 has been investigated for various geometries. The pressure drop penalties incurred by wavy walled geometries is observed to decrease in amplitude of the waviness, for fixed L/a ratio. The heat transfer through a wavy walled geometry is always higher that the parallel-plate configuration.

Naphon [6] presented the numerical results of the flow and temperature distributions in the channel with various geometry configuration wavy plates. The effects of relevant parameter on the flow and temperature structures are also considered. It can be found that the sharp edge of the wavy plate has a significant effect on the enhancement of heat transfer especially the V-shaped wavy plate. Therefore, using wavy plate is a suitable method to increase the thermal performance and higher compactness of the heat exchanger.

Mohammad et al. [7] numerically investigated heat transfer and water flow characteristics in wavy micro channel heat sink (WMCHS) with rectangular cross-section with various wavy amplitudes ranged from 125 to 500µm. This investigation covers Reynolds number in the range of 100 to 1000. The water flow field and heat transfer phenomena inside the heated wavy micro channels is simulated and the results are compared with the straight micro channels. The effect of using a wavy flow channel on the MCHS thermal performance, the pressure drop, the friction factor, and wall shear stress is reported in this article. It is found that the heat transfer performance of the wavy micro channels is much better than the straight micro channels with the same cross-section.

Nandi and Chattopadhyay [8] numerically studied the simultaneously developing unsteady laminar fluid flow and heat transfer inside a two dimensional wavy micro channel, due to sinusoidaly varying velocity component at inlet. The flow was both thermally and hydrodynamically developing while the channel walls were kept at a uniform temperature.
The simulation was performed in the laminar regime for Prandtl number 7 and Reynolds number ranging from 0.1 to 100. Based on the comparison with steady flow in wavy channel it was found that imposed sinusoidal velocity at inlet can provide improved heat transfer performance at different amplitude (0.2, 0.5, 0.8) and frequency (1, 5, 10).

Kuhn et al. [9] numerically studied the mixed convective flow of water over a heated wavy surface over a range of Reynolds and Richardson numbers, including transitional and turbulent flow regimes. The integral heat transfer for the wavy wall configuration is significantly enhanced (approximately 2.5 times) for Re = 1000, 2000 in comparison with the standard flat horizontal wall configuration.

Sui et al. [10] conducted experimental investigation on the flow friction and heat transfer in sinusoidal micro channels with rectangular cross sections. The micro channels considered consist of ten identical wavy units with average width of about 205 mm, depth of 404 mm, wavelength of 2.5 mm and wavy amplitude of 0-259 mm. Each test piece is made of copper and contains 60-62 wavy micro channels in parallel. Deionized water is employed as the working fluid and the Reynolds numbers considered range from about 300 to 800. The experimental results, mainly the overall Nusselt number and friction factor, for wavy micro channels are compared with those of straight baseline channels with the same cross section and footprint length. It is found that the heat transfer performance of the present wavy micro channels is much better than that of straight baseline micro channels; at the same time the pressure drop penalty of the present wavy micro channels can be much smaller than the heat transfer enhancement.

Earlier studies on turbulent flow characteristics and heat transfer performances in square channels with different cylindrical- shaped grooves are analysed and compared numerically in his research. The novel groove geometries are conventional cylindrical grooves with rounded transitions to the adjacent flat surfaces and with modifications to their bases. The objective of this work is to determine optimal configuration for augmenting heat transfer rates with minimal pressure drop penalties. This investigation shows that the conventional cylindrical grooves have similar overall heat transfer enhancement with conventional square ribs, but the pressure loss penalty is much decreased from square rib values. The rounded transition of the grooves has a large advantage over conventional cylindrical grooved surfaces in both enhancing heat transfer and reducing pressure loss penalty.

CHAPTER 3: NUMERICAL ANALYSIS

3.1-Pre-Processing:

3.1.1-*Design module*

In design module, XY plane is chosen to draw the geometry and then looked at. Unit is set to mm and auto-constraint is turned on. Six points are taken using construction points and then joined with the help of polyline. The dimensions are set to (0, 20), (125, 25), (375, 15), (525, 25), (875, 15), (1000, 20). Two vertical lines are drawn from (0, 20) to origin and (1000, 20) to Y-axis respectively. After that a horizontal line is drawn from origin to (1000, 20). So the top-wall looked like 2 triangular waves. Three more geometries are drawn with 4, 6 and 8 triangular waves respectively.

From the Concept toolbar, Surfaces from Sketches is taken and the sketch drawn above is applied and the surface is generated as shown in fig.1.

3.1.2-*Meshing*

In mesh, relevance centre is set to fine and smoothing is set to high to make a better grid size. Then edge sizing is inserted selecting the two vertical edges and 20 number of divisions are taken making the behaviour hard with no bias. The same is repeated for two horizontal edges with 1000 number of divisions. After that for a fine grid size Mapped Face Meshing is applied to the face of the surface and finally mesh is generated.

After meshing each edge is given a name using Create Named Selection. The names given are inlet, top-wall, bottom-wall and outlet selecting the appropriate edges.

3.1.3-*Physical Set-up*

In General, the unit of Scale is taken mm as all the dimensions are in mm. The energy equation is turned on and laminar flow is taken. Water is taken as the fluid and aluminium is taken the solid material.

Now the boundary conditions are taken:

Inlet : Velocity-inlet with inlet velocity 0.05m/s and temperature is set to 300K

Top-wall: Wall with constant wall temperature 350K

Bottom-Wall: Wall with constant wall temperature 300K

Outlet: Pressure outlet

After that solution is initialized using standard initialization computed from inlet and values are calculated relative to cell zone. For better results and convergence with lesser errors, residuals are set to e-06 for continuity, y-velocity and x-velocity whereas for energy it is chosen to e-08, in Monitors.

The calculation is iterated for 10000 times. Wait till the solution is converged.

3.2-Post-Processing:

Contours of total temperature and velocity are plotted and their images are taken.

From Surface tool bar, point is chosen and 32 points are created on the top-wall. Initially, the gap between the points is 5mm and gradually increased to 10mm and finally to 50mm e.g. 5, 10, 15….50, 60…100, 150….950. Similarly 32 lines are created joining the top-wall and the bottom-wall vertically with spacing same as given in creation of points. This is done because the variation of Nusselt number with length is more initially and when the flow is fully developed it becomes constant.

The area-weighted average of temperature is calculated and extracted to a text file. From this, temperature on the top-wall is obtained. Again, area-weighted average of wall fluxes is calculated and extracted to a text file. From this, local wall flux is obtained. Now, mass-weighted average of the lines created is obtained to get the fluid temperature.

From the above obtained values and using the formula given below local Nusselt number is calculated and plotted against the length of the channel.

$h=q/(T_w-T_f)$ $N_u=h\,(Dh/k)$

3.3- Validation

3.3.1-*Design Module*

For validation of this project, another channel is taken for analysis. The geometry consists of a rectangle having horizontal dimension 1000mm and vertical dimension 20mm. These two dimensions are same as taken above.

3.3.2-*Meshing*

In mesh the relevance centre is set to fine and smoothing is set to high. The two horizontal edges are selected for edge sizing with 20 number of divisions with - -- --- -- - bias. The bias factor is 1.2 with smooth transition. Similarly, the two vertical edges are selected for edge sizing with 1000 number of divisions with - -- --- ---- bias. The bias factor is 1 with smooth transition. The face of the surface is selected for mapped face meshing and finally the mesh is generated.

3.3.3-*Physical Set-up*

The physical set up is kept same as taken in the problem and the solution is initialized. When the solution is converged, the values Top-wall heat flux and Top-wall temperatures are calculated from area-weighted average of the same on the Top-wall. The value of fluid temperature is calculated from mass weighted average of the lines drawn vertically from Top-wall to the centreline. Using these values, convective heat transfer coefficient of the wall is calculated. After that using the relation of Nusselt number with convective heat transfer coefficient, thermal conductivity of fluid (water, k=0.6 W/m-K) and characteristics length (20 mm i.e. 0.02m). Then a graph of Nusselt number versus length of the channel is plotted with Nusselt number taken in the y-axis and length of the channel taken in x-axis.

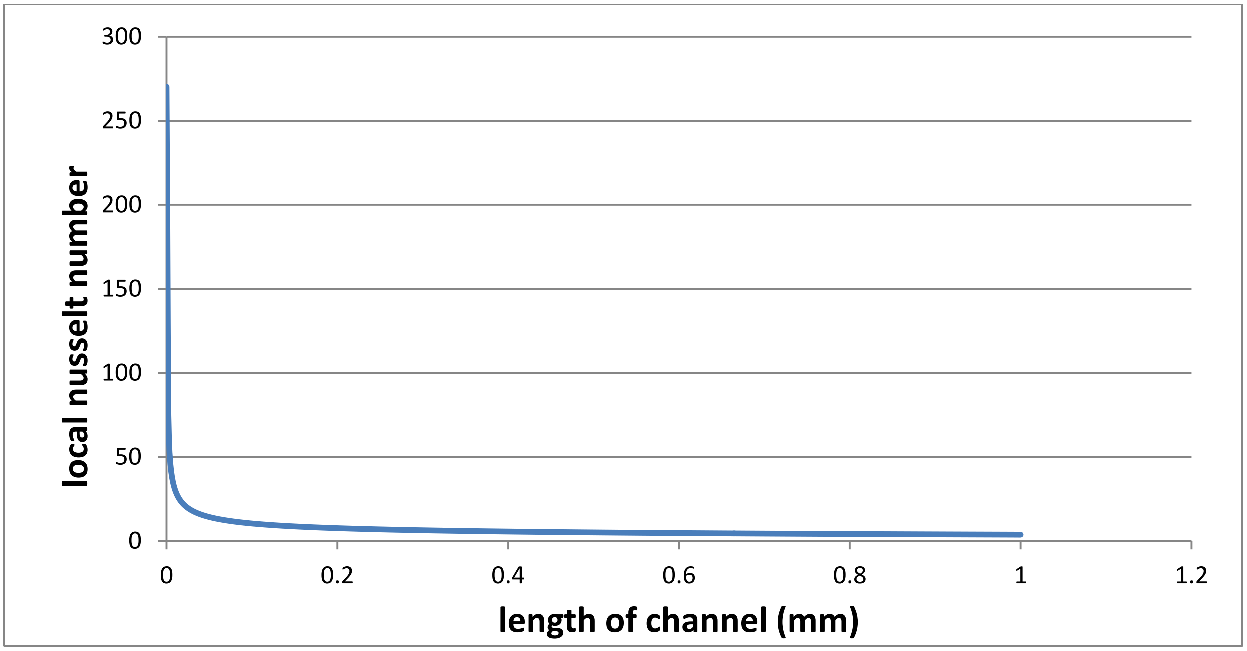

Fig.1- Graph of local Nusselt number vs. length of the channel for Circular Tube

The variation of Nusselt number with length of channel is shown in fig. (3). It is clearly seen that the value of local Nusselt number is high initially and then it rapidly decreases when the channel length is increased. When the flow is fully developed, it converges to a constant value of 3.71.

CHAPTER 4: RESULTS AND DISCUSSIONS

The results obtained from the numerical calculations are discussed below.

4.1-Contours

4.1.1-Velocity

From figure (2) it can be clearly seen that the fluid velocity inside the channel is varying from zero to 8.66e-02 m/s. The fluid velocity is zero at the wall and it is maximum at the centre. Now as the number of waves increases the maximum velocity of the fluid increases i.e. the geometry having 4 waves has maximum velocity of 8.69e-02 m/s and the geometry having 8 waves has maximum fluid velocity of 8.82e-02 m/s.

4.1.2-Temperature

From figure (3) it is clearly visible that the temperature on the top wall is maximum i.e. 3.50e+02 K and it is constant at every point on the top-wall. As we move from top-wall towards bottom wall the temperature rapidly decreases in a small span of width and then it gradually decreases till it reaches its minimum value on the bottom-wall.

4.2. Nusselt number

The variation of local Nusselt number with the length of the channel is shown in the figure (4). It can be observed that initially i.e. when the channel length is small compared to width, the Nusselt number is very high (40-50). So in this region convective heat transfer is more compared to heat transfer by conduction. When length of the channel is much more compared to width Nu decreases. It can be clearly seen from fig that in upper wavy region Nu is 8-12 while at normal width it is 4-7. So in the upper wavy region convective heat transfer is more and in this way it enhances the heat transfer of the channel with triangular wavy wall.

Again as the number of triangular waves increase from 2 to 8, the area affected by the triangular waves increase. It can also be observed that the Nusselt number value for channel with 8 waves is more than channel with 2 waves (when channel length is more compared to width). So, more the number of waves in a channel more will be the convective heat transfer.

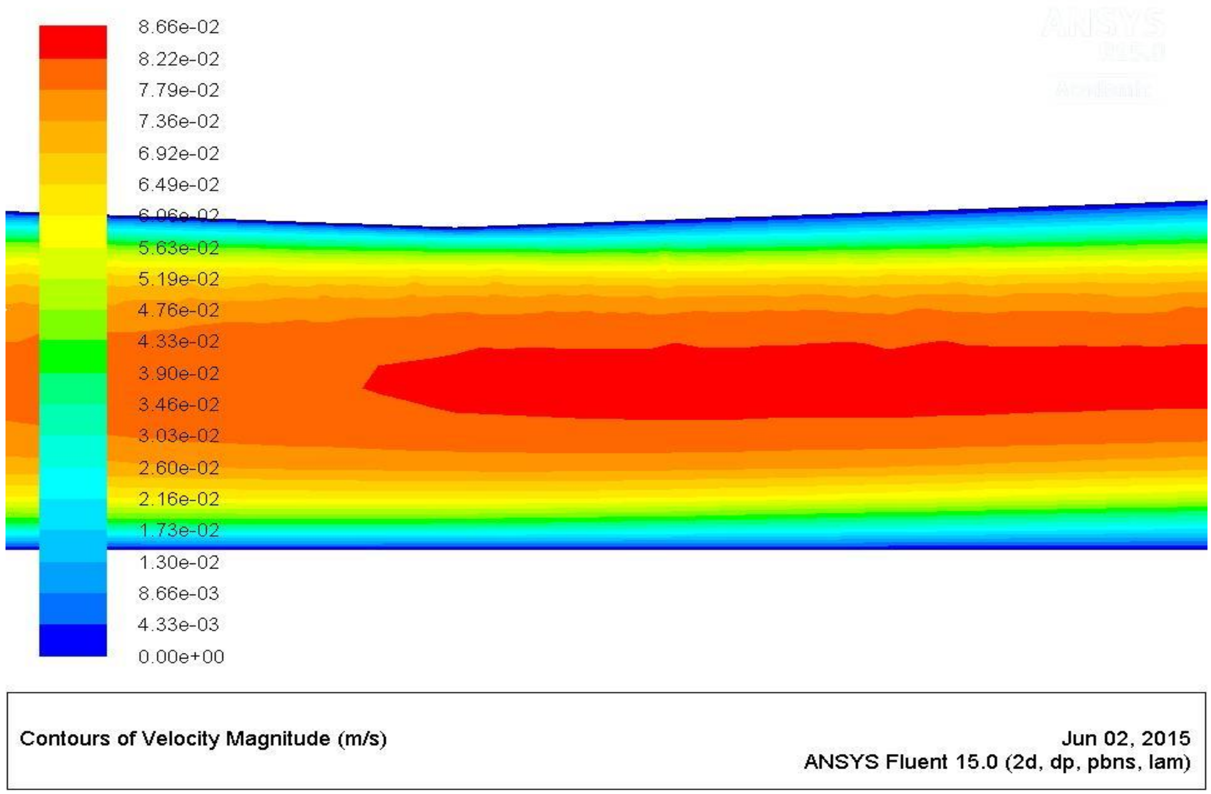

Fig.2.1- Velocity contours for channel with 2 triangular waves

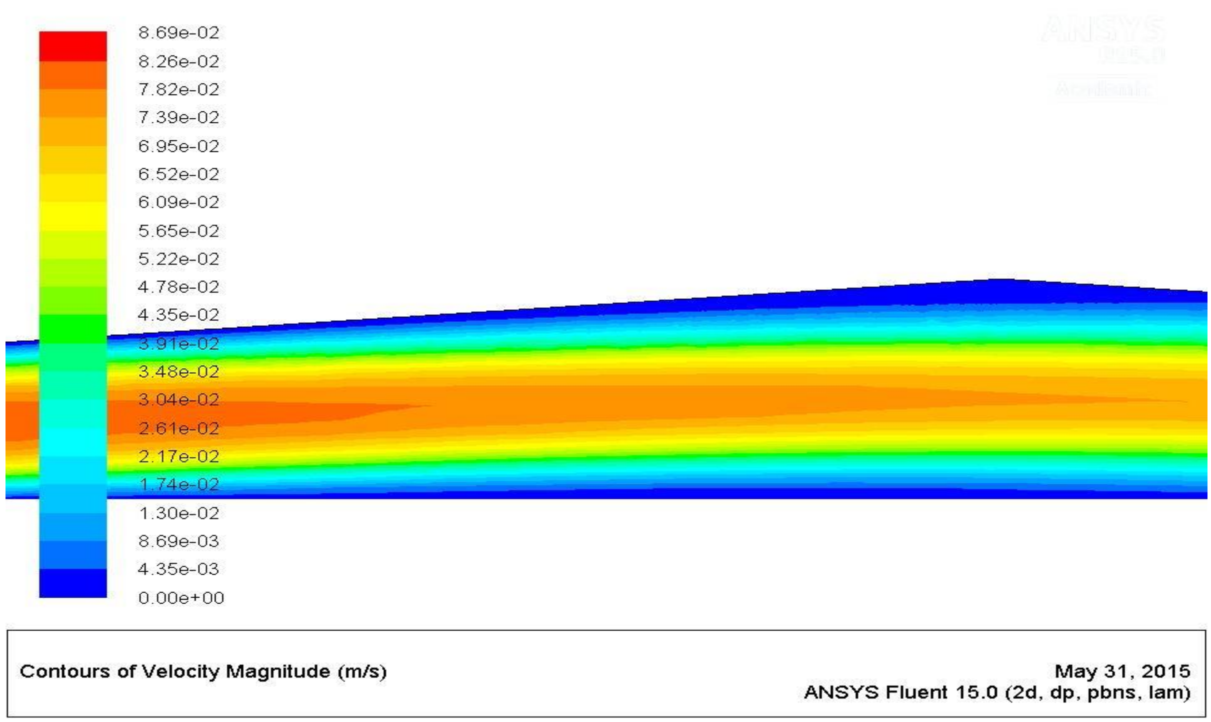

Fig.2.2- Velocity contours for the channel with 4 triangular waves

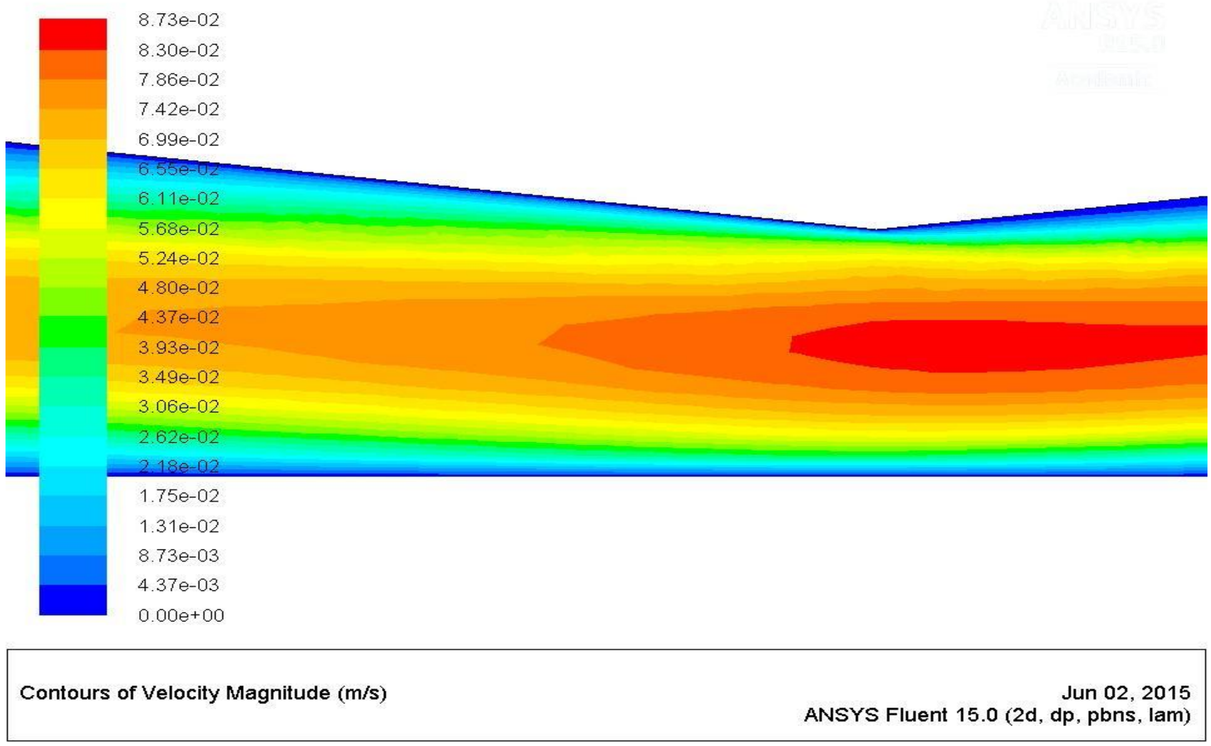

Fig.2.3- Velocity contours of channel with 6 triangular waves

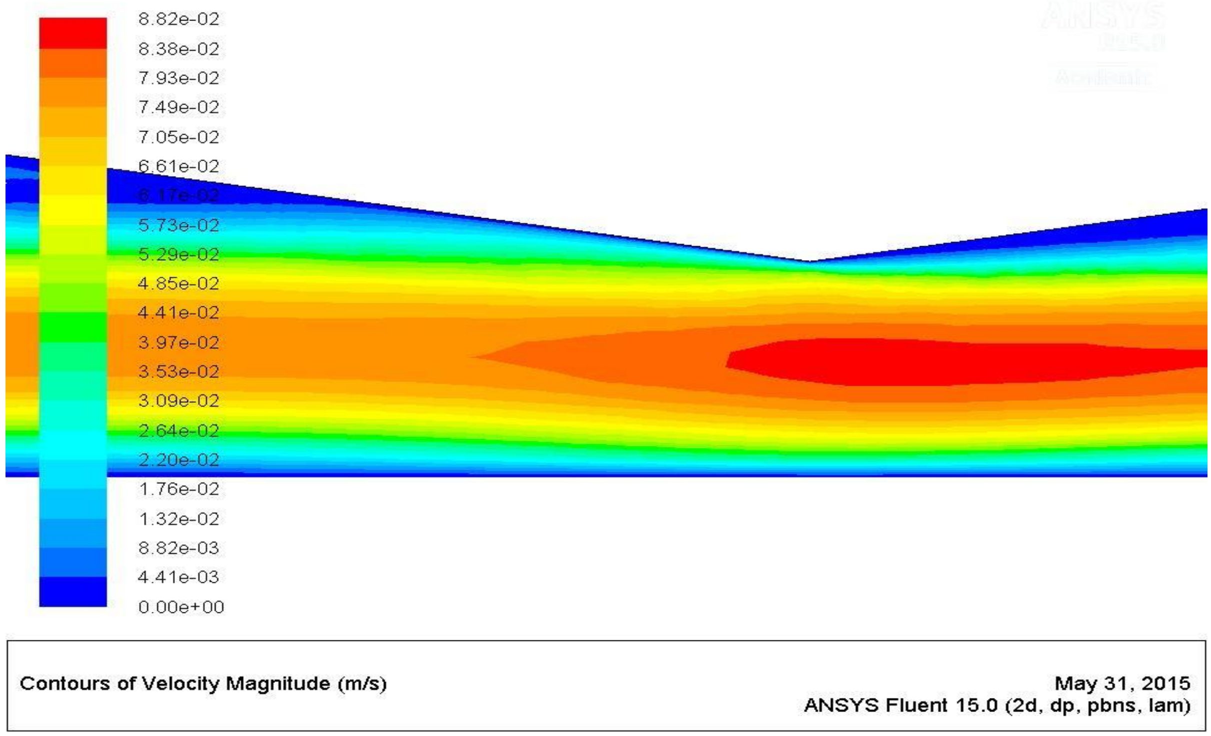

Fig.2.4- Velocity contours of channel with 8 triangular waves

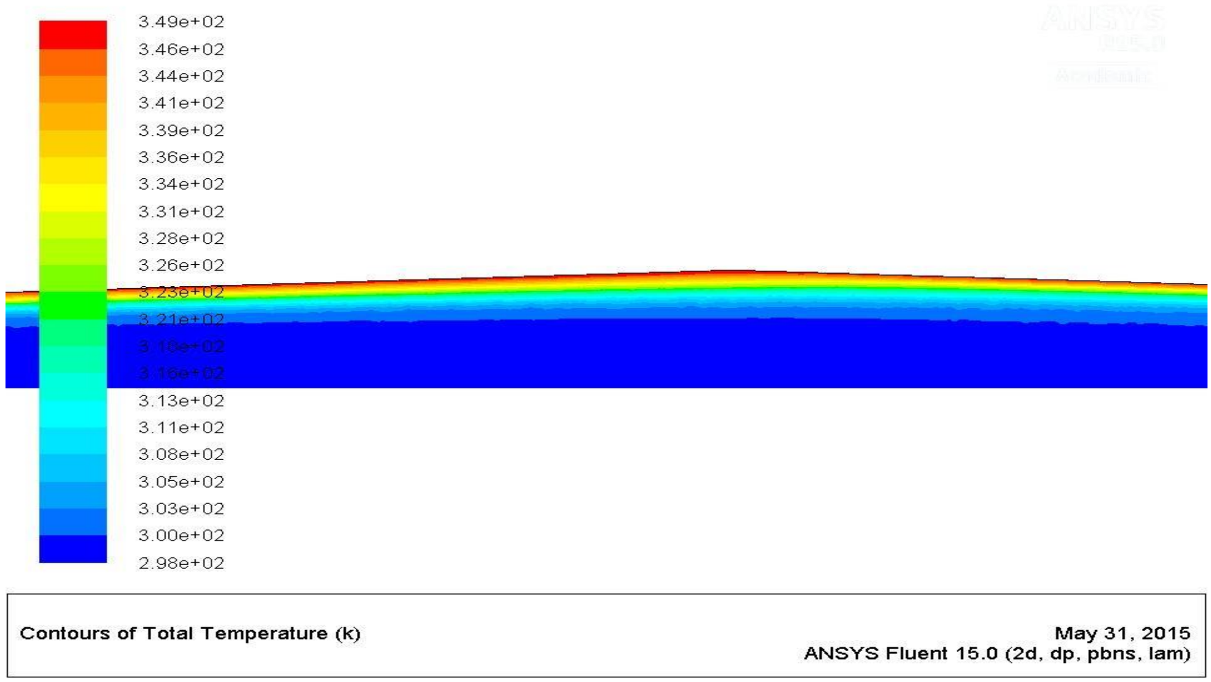

Fig.3.1- Temperature contours for flow with 2 triangular waves

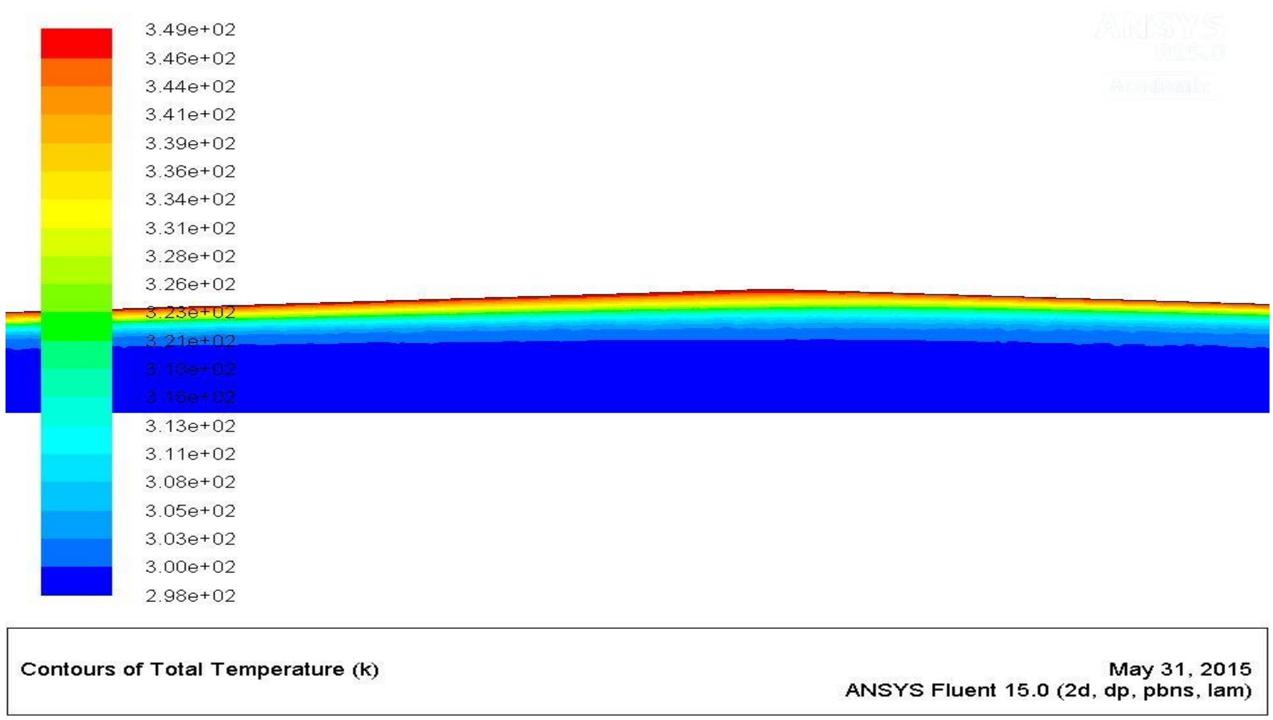

Fig.3.2- Temperature contours of channel with 4 triangular waves

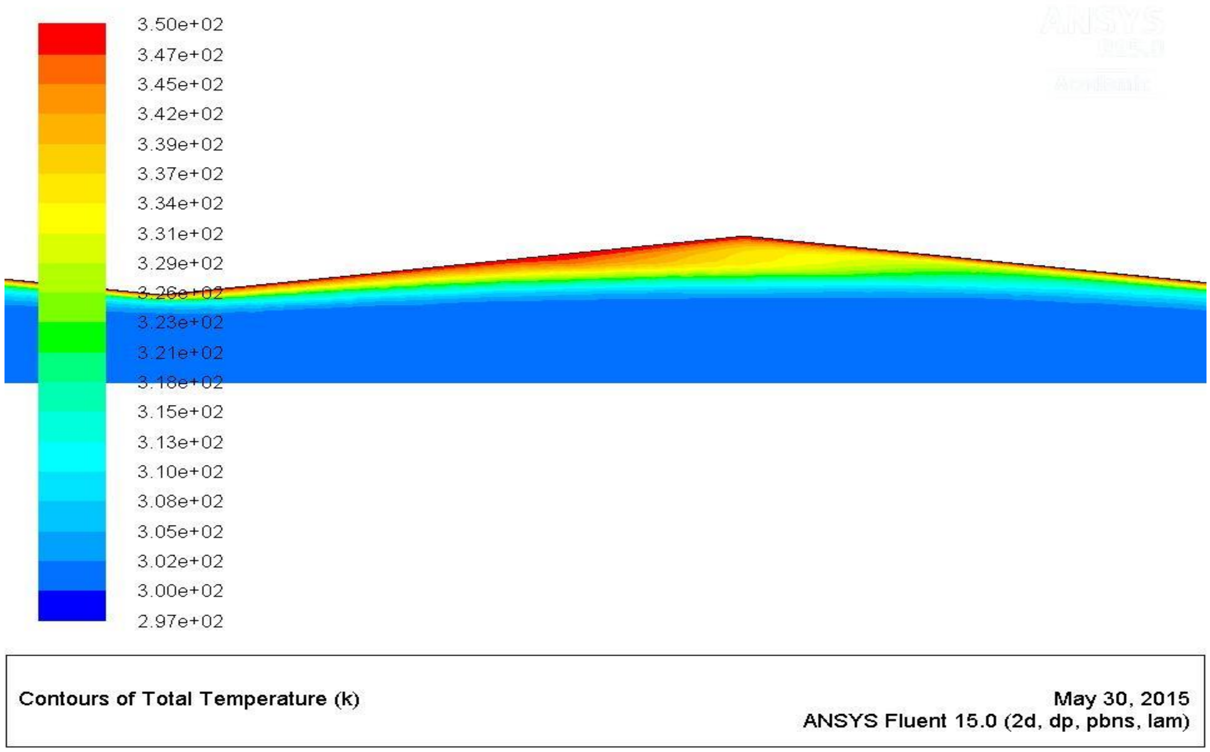

Fig.3.3- Temperature contours of channel with 6 triangular waves

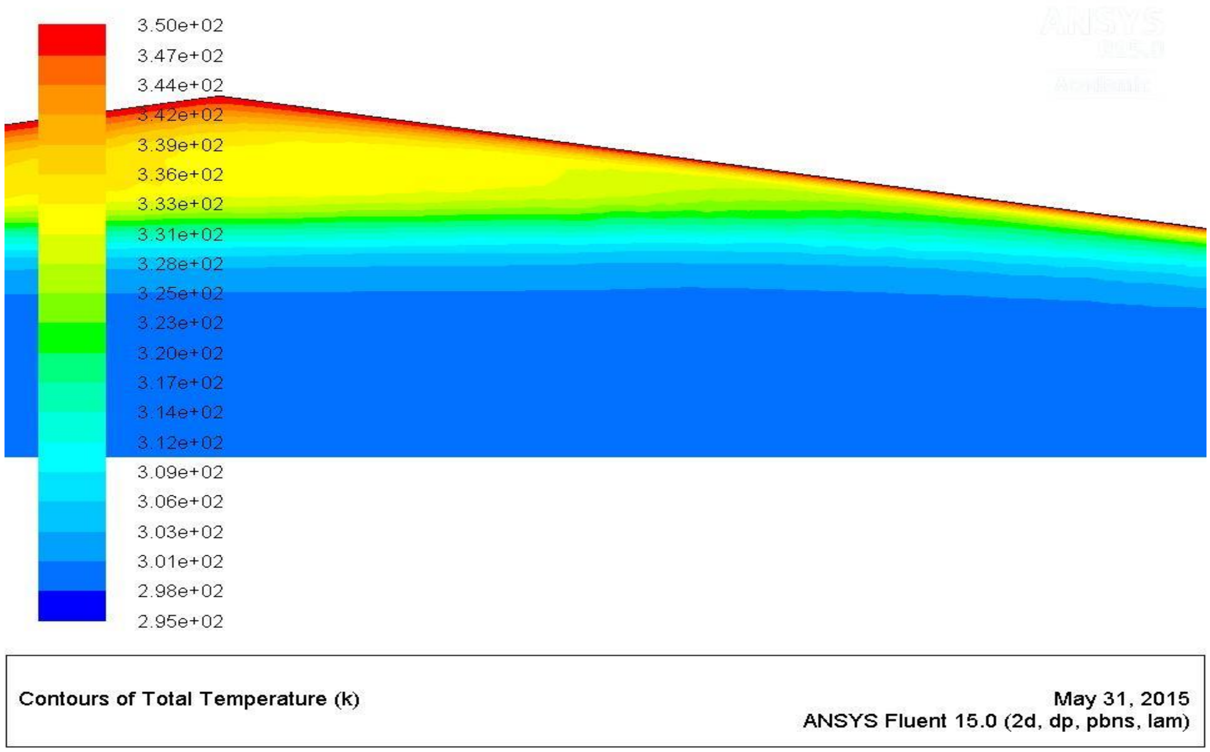

Fig.3.4- Temperature contours of channel with 8 triangular waves

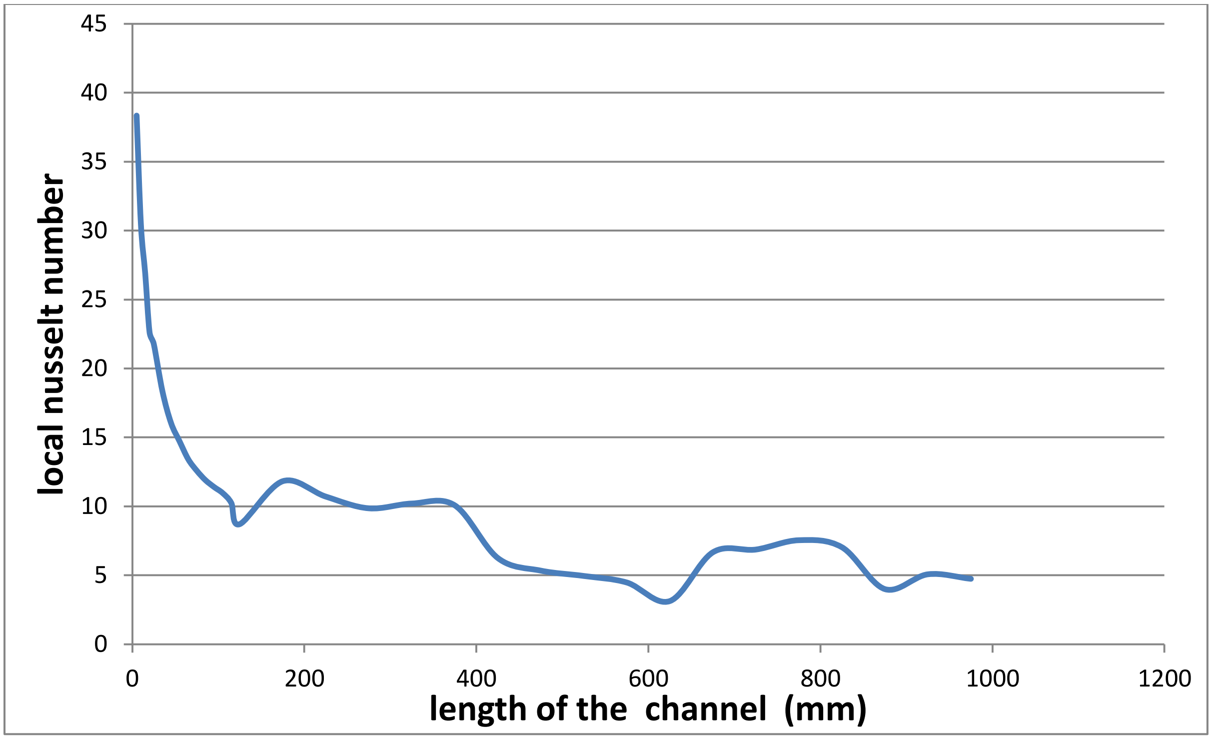

Fig. 4.1- Graph of local Nusselt number vs. length of the channel with 2 triangular waves

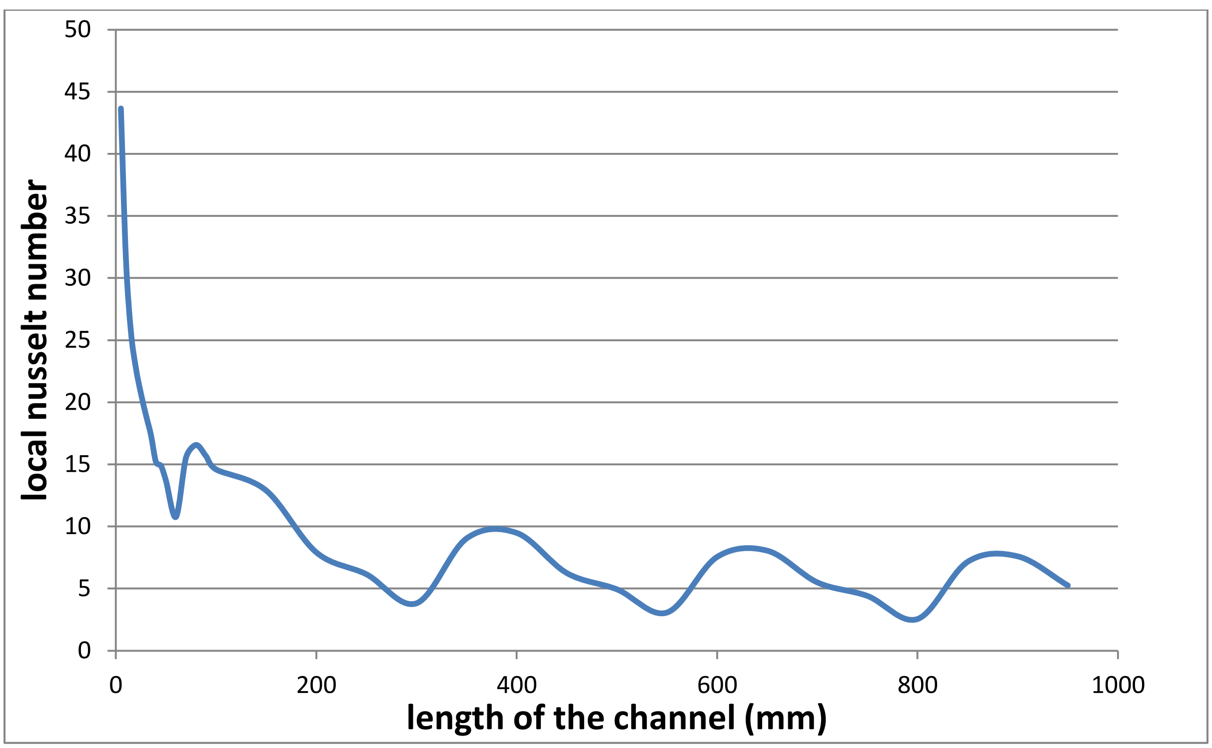

Fig. 4.2- Graph of local Nusselt number vs. length of the channel with 4 triangular waves

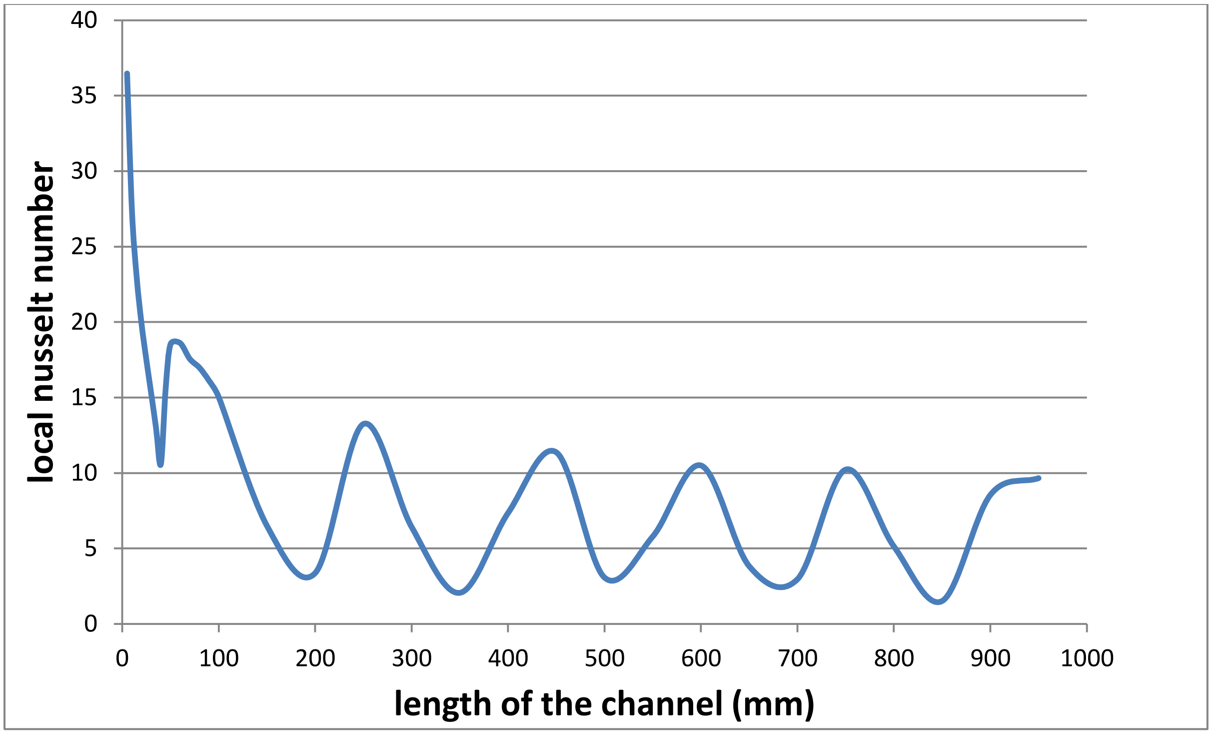

Fig. 4.3- Graph of local Nusselt number vs. length of the channel with 6 triangular waves

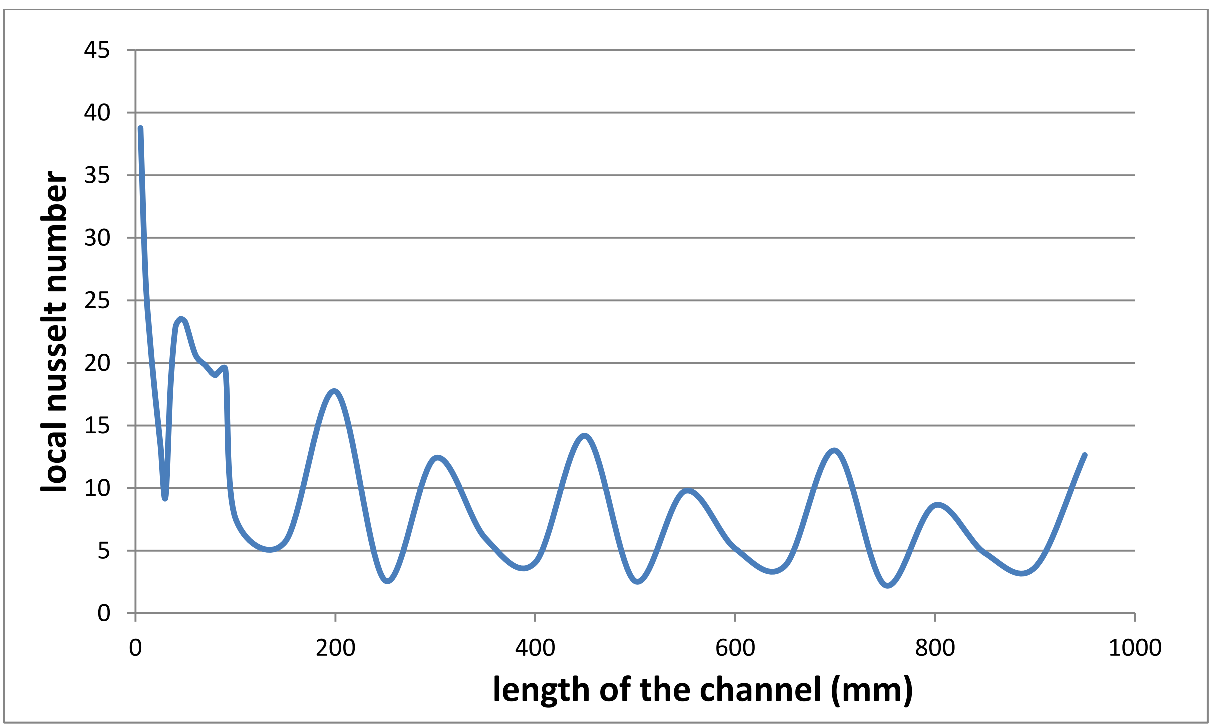

Fig. 4.4- Graph of local Nusselt number vs. length of the channel with 8 triangular waves

CHAPTER 5: CONCLUSION

This project work investigates the heat transfer characteristics of a channel with triangular top-wall. The variation Nusselt number with length of the channel is studied and following conclusions are drawn:

- The maximum fluid temperature increases as the number of wave increases.
- The average Nusselt number increases as the number of waves increases which in turn increases the convective heat transfer from the top-wall.
- The Nusselt number is more in the upper wavy region and as the number of waves increases the area affected by the triangular waves increases and thus the convective heat transfer increases.

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