CFD Analysis of Heat Transfer Enhancement Using Nanofluid And Observation Of The Geometric effect in case of Flat Cross Section.

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ABSTRACT

In the present study, the heat transfer enhancement using aluminium oxide with 1% volume fraction nanofluids in laminar pipe flow for different cross-sections with constant heat flux(H1 boundary condition) has been studied by computational fluid dynamic modeling of the fluid flow adopting the single phase approach. After investigating different cross sections FLAT cross section was selected as it showed better heat transfer performance and again this FLAT section is further flattened to observe it's effect on Nusselt number and to find out the optimum geometry for better heat transfer performance.

DEDICATION

To our respected mentor Professor Dr. A.K.M SADRUL ISLAM

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Contents

1 . Introduction

Heat transfer depends on various factors among them shape of cross sections play a vital role. Also Heat transfer can be sufficiently increased by introducing nanofluid.

1.1 Aims And Objectives

First, single phase fluid (water) flow is numerically simulated in straight pipe with circular and rectangular cross sections with different aspect ratios. Nusselt number for fully developed flow conditions for these geometries is validated with results available in the literature. Then boundary condition, H1 (constant axial heat flux with constant peripheral wall temperature) and the corresponding Nusselt numbers were validated.

Once the results were validated for rectangular cross sections for different aspect ratios and circular cross sections then Flat cross sections of different aspect ratios were simulated using nanofluid and corresponding nusselt numbers were calculated.

2.LITERATURE REVIEW

In this section, a brief background of related literature is presented. Literature review is divided into three sections. The first section summarizes the findings on the use of nanofluids for heat transfer. In the second section, a review of the available literature in the use of Rectangular cross sections and circular cross sections is presented.

2.1 Nanofluids

In the last decades, many efforts have been made to produce more efficient heat exchangers in order to conserve energy. These efforts deal with improving the heat transfer rate by means of extended surfaces, mini-channels and micro-channels. In addition, many studies have been carried out on thermal properties of suspensions of solid particles in conventional heat transfer fluids to enhance their poor thermal performance. For this purpose, Ahuja [1] dispersed micro/millimeter sized particles with high thermal conductivity in the base fluid. Choiet al. [2] dispersed nanometer sized particles in the base fluid which were called nanofluid. Nanofluids were an improvement on previous suspensions, because they suffered from less sedimentation, large pressure drops, and were less corrosive.The development of nanofluids as a new class of heat transfer with the use of nanotechnology has been the subject of much attention in recent years. Hwang et al. [3] investigated the convective heat transfer coefficient and pressure drop of Al_2O_3/water nanofluids inside a circular tube in the fully developed laminar flow regime with constant heat flux. They reported that, convective heat transfer coefficient increased by up to 8% at 0.3% volume concentration. In addition, the thermal conductivity of nanofluid depends on particles shape and size, and also the thermal properties of base fluid and nanoparticles.

2.2 Rectangular Cross-Sections

Shah and London[4] did an extensive work on various cross sections including Rectangular cross sections and evaluated Nusselt number. That literature was helpful for finding out analytical solution.

3. METHODOLOGY

3.1 Background

This section provides the basic equations, assumptions, properties and specific heat model used in the current study.

3.1.1 Governing equations

The basic equations to solve the fluid motion and the heat transfer are mass conservation (Continuity), momentum conservation (Navier-Stokes) and energy equation. These basic equations [5] are listed below:

$$\nabla \cdot (\vec{v}) = 0 \qquad (1)$$

$$\nabla \cdot (\vec{v}\vec{v}) = -\nabla p + \nabla \cdot (\bar{\bar{\tau}}) \qquad (2)$$

$$\nabla \cdot (\vec{v}(\rho E + p)) = k\nabla \cdot \nabla T + \nabla \cdot (\bar{\bar{\tau}} \cdot \vec{v}) \qquad (3)$$

These equations do not have a direct analytical solution except for very simple cases, and hence have to be solved in the discretized form in a domain. In this study, they are solved using finite volume technique using commercial computational fluid dynamics (CFD) software FLUENT.

3.1.2 Assumptions

Basic assumptions taken in this study are listed below. They are valid for both Pure and nanofluid.

a. Fluid continuum is assumed in microchannel flow.

b. Fluid is assumed to be Newtonian.

c. Incompressible flow.

d. Surface is considered to be smooth and no-slip condition is used on all wall surfaces.

e. Fluid properties are assumed to be constant with respect to temperature.

f. Buoyancy effects are neglected as density is assumed to be constant.

g. Natural convection effects are neglected.

3.1.3 Explanation of thermal boundary conditions

There are two main types of thermal boundary conditions that can be applied at the wall. One is specifying a constant heat flux and the other is specifying a constant wall temperature. Also, there are two different ways of specifying constant heat flux. These boundary conditions are listed and explained below:

1. Constant axial heat flux with constant peripheral wall temperature (H1)
2. Constant axial and peripheral heat flux (H2)

The constant heat flux boundary condition has two variations, H1 and H2. In the case of H1 condition, the heat flux is constant in the axial direction and the temperature is held constant in the peripheral (angular) direction. The H1 boundary condition is more reasonable, practical and considers the actual physics of most heat transfer phenomena. It has been found in majority of the studies that H1 condition is more appropriate and close to the experimental findings than H2 condition. In the case of H2 boundary condition, both the axial and peripheral heat flux is held constant and hence peripheral wall temperature varies. This type of boundary condition is widely used in numerical studies for its modeling simplicity, but it does not have much practical importance as majority of the heat exchangers fall into either H1 or T boundary conditions. It is important to note that H1 and H2 difference occurs only when the solid wall thickness is assumed to be zero. When the wall thickness is considered, as in the case of conjugate heat transfer problems, H1 and H2 does not make any difference as the heat transfer inside the solid is modeled and the heat flux is applied at the bottom wall, away from the fluid-solid interface. Another exception is the case of circular duct and channel with zero wall thickness, where both the heat flux and wall temperature remain constant in the peripheral direction because of symmetric geometry.

3.2 Fluid properties and flow parameters

This section explains the fluid properties, calculation of the nanofluid properties, flow parameters used and how they were calculated.

3.2.1 Fluid Properties

All thermophysical properties of nanofluid are calculated with equations (1) to (4) which can be found in the literature[6]-[9]. All properties are calculated using bulk temperatures between inlet and outlet

$$\rho_{nf} = (1-\Phi)\rho_{bf} + \Phi\rho_p \qquad (1)$$

$$(\rho C_p)_{nf} = ((1-\Phi)(\rho C_p)_{bf} + \Phi(\rho C_p)_p \qquad (2)$$

$$\mu_{nf} = (1+2.5\,\Phi)\,\mu_{bf} \qquad (3)$$

$$k = [\frac{k_p + 2k + 2(k_p - k)(1+ß)^3\Phi}{k_p + 2k - 2(k_p - k)(1+ß)^3\Phi}] \qquad (4)$$

In the above equations, μ stands for viscosity, ρ stands for density and k stands for thermal conductivity. The subscripts nf, bf and p stand for nanofluid,base fluid and particle respectively and ß is the ratio of nano layer thickness to the nano particle diameter which is equal to 0.1.

The properties of fluid and nanofluid used, and the resulting properties are given in Table 1 below

Property names	Water (From chart)	Nanofluid (From eqns to be shown on next slide)
Density,Kg/m3	1000	1029.5
Dynamic viscosity,Ns/m2	1.002*10^-3	1.027*10^-3
Thermal Conductivity,W/m-K	0.6	0.624
Specific Heat,j/kg-K	4200	4068.7

Table 1 Properties of water and nanofluid

3.2.2 Flow and heat transfer parameters

Flow and heat transfer characteristics are generally specified using a set of dimensionless numbers. These dimensionless numbers were calculated in different ways, depending upon the geometry and the flow conditions. The flow parameters used and the ways they were calculated are explained in this section.

HYDRAULIC DIAMETER

Hydraulic diameter is the characteristic length used for determining the flow characteristics. It is defined as the ratio of four times the cross-sectional area (perpendicular to the flow) to the wetted perimeter.

$$D = \frac{4A}{P}$$

Where,

A= cross sectional Area

P= wetted perimeter

Reynolds number

Reynolds number is generally used to determine whether the flow is laminar or turbulent. In this study, Reynolds numbers were taken such that the flow remains laminar for the geometries considered. Reynolds number (Re) were calculated based on pin and microchannel hydraulic diameters using the equation given below

$$Re = \frac{\rho V D}{\mu}$$

Fluid bulk temperature

Bulk temperature of the fluid is the mass weighted average temperature of the fluid at a given cross section in the channel. Among the different ways of calculating the average fluid temperature like area-weighted, facet average etc., mass weighted average gives the best estimate (from energy balance) . Mass weighted average is calculated based on the following equation,

$$T_b = \frac{\int_A (V*T)dA}{\int_A VdA}$$

Heat transfer coefficient

The convective heat transfer coefficient is calculated based on the average fluid and wall temperatures at a given location. The average heat flux from the wall is known from the simulation and this is used to calculate the heat transfer coefficient, given by the equation,

$$h = \frac{q''}{T_w - T_b}$$

where,

q''=Average wall heat flux along the periphery at a particular axial location
T_w=Average wall temperature along the periphery at a particular axial location

T_b=Mass weighted average of fluid temperature on the cross section at a Particular axial location

For the constant heat flux boundary condition, $q''$is known since it is one the input variables, but T_w needs to be calculated from the simulation. For the case of constant wall temperature, T_w is known whereas q'' needs to be calculated from the simulation. Whether it is T_w or q'' that needs to be determined from the simulation, both are calculated in the same way by taking the average value along the periphery of the wall.

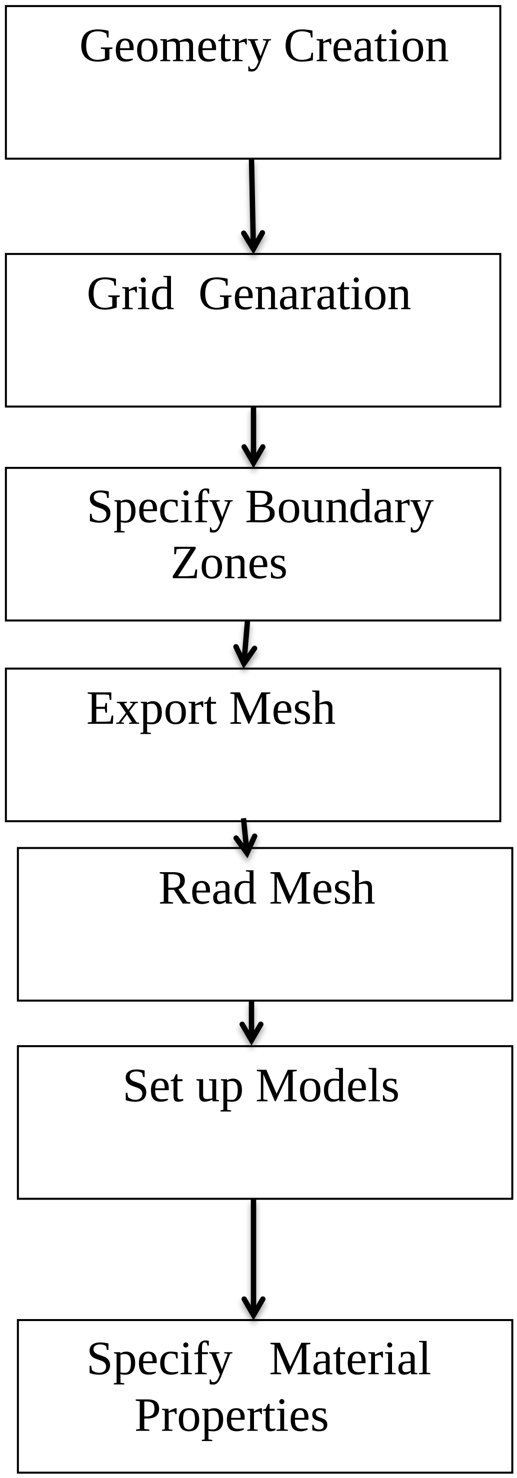
<u>Nusselt number</u>

Nusselt number is one of the very important parameters for analyzing heat transfer performance. Higher the Nusselt number, more effective is the heat transfer process. This dimensionless number essentially indicates the effectiveness of the convective heat transfer process taking place between the walls and the fluid. It is based on the convective heat transfer coefficient and is given by,

$$Nu = \frac{hD}{K}$$

3.3 Modeling Procedure

Heat transfer and fluid simulations of laminar flow in microchannels were performed using computational fluid dynamics (CFD) software Fluent. The geometry and grid of interest were created in Gambit. The flow and heat transfer simulations were performed using Fluent. The steps followed for the entire modeling are shown in Figure 1



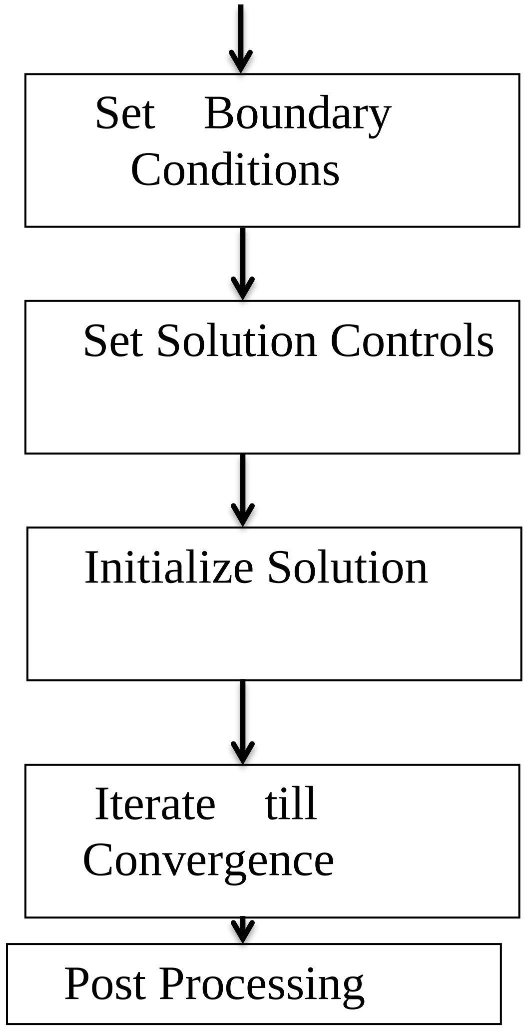


Figure 1Modeling procedure

3.3.1 Geometry and Grid

First, circular , Rectangular, Flat, Elliptical pipe were simulated in the study to determine the effect of cross-sectional shape on Nusselt number. Then Flat Cross section was selected, then Flat cross sections of eleven aspect ratios were simulated as shown in Figure 2.

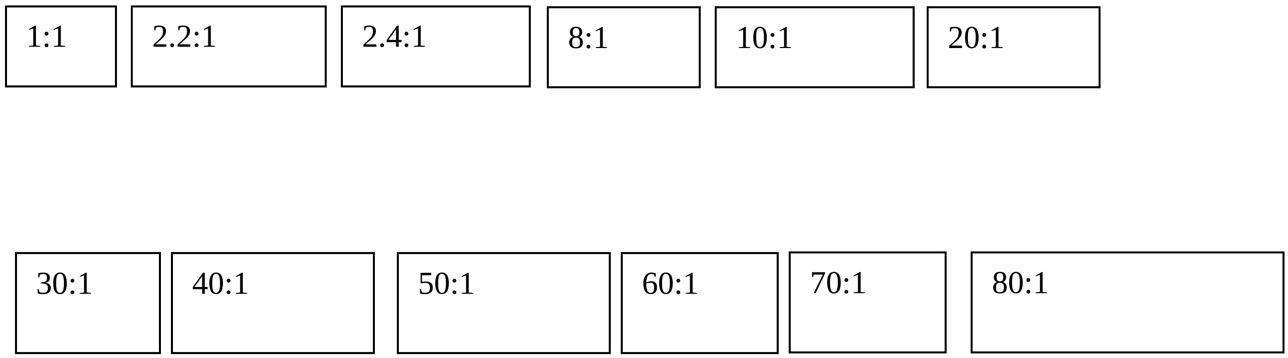


Figure 2 Aspect ratios of different flat cross sections

Once the geometries were created, the corresponding grids were generated consisting entirely of hexahedral elements. Three different grid resolutions were considered to obtain grid independent solution. Grid convergence results are presented in the Results and Discussion section. Figure 3 shows a typical grid near the inlet section.

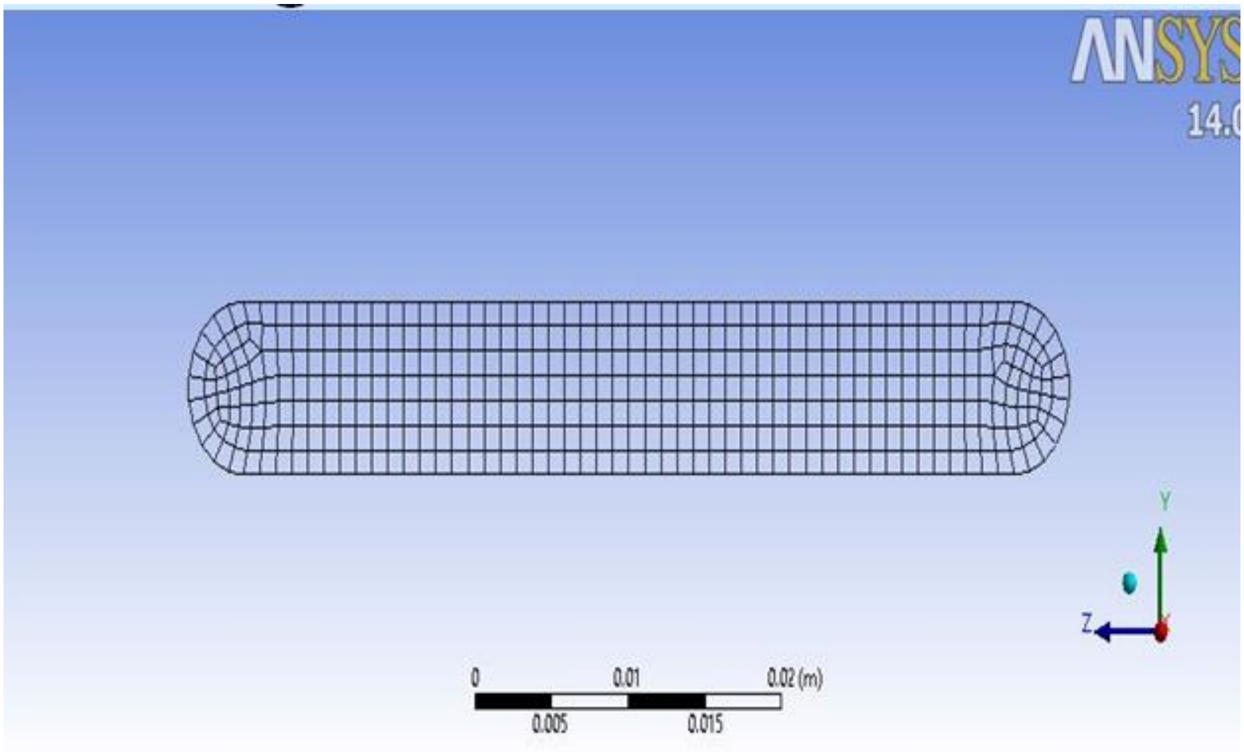


Figure 3 Typical grid near inlet

3.3.2 Solving in FLUENT 6.3

Once the required grid was generated, it was exported from Gambit and read by Fluent. Scaling was performed if the geometry and grid were generated in different units. Fluent solution options such as “laminar flow”, “steady flow”, “turning on energy equation” were set. Material properties were set as calculated for the nano fluid. First, velocity at the inlet was set as constant with uniform flow as initial condition. Temperature at the inlet was set to 300 K. The fluid pressure at the outlet was set to 1 Atm. Fluent uses this back pressure to calculate the inlet pressure, so that the required mass flow rate (velocity) based on continuity and momentum could be obtained. Walls were set to non-slip condition. The wall thermal boundary condition was specified as either constant heat flux or constant wall temperature. In the case of constant heat flux, only H1 boundary condition was considered. However, setting the H1 boundary condition was difficult, because most of the commercial CFD solvers do not have this condition available in their software package. Lee and Garimella [10] explained how this condition can be set in Fluent using a "thin wall" model with "shell conduction". "Thin wall" model requires the user to set a small thickness (can be very small) to allow for peripheral heat conduction. Setting the "shell conduction" model ensures that the applied heat flux redistributes inside the wall in the peripheral direction. This redistribution of heat flux allows for temperature uniformity in the peripheral direction, resulting in the required H1 boundary condition. It is to be noted that in this case, the heat flux varies in the peripheral direction as opposed to being constant, as in the case of H2 boundary condition. Flow was simulated using second order accuracy. Continuity, x, y and z momentum equations, and energy equation residuals were monitored. The convergence criterion was set to 10-6for all these equations. Under-relaxation factors within the range of 0.3 to 1.0 were used and were found to give good convergence behavior and accurate results. Apart from checking residuals for convergence, velocity and temperature at specific locations inside the geometry were monitored and made sure that they reach constant values (with no variation for further iterations).

4. RESULTS AND DISCUSSION

4.1 Validation

It is very important to validate the simulation methodology. Many lessons were learnt during the validation process, especially in the application of boundary conditions and extracting (post processing) the results from the simulation. For validating the numerical heat transfer results, the theoretical or experimental Nusselt numbers values (available in the literature) were compared with the results from the simulation. This procedure was done for Rectangular and Circular cross sections only.

The consecutive tables shown below show the validations;

Geometry	Nu – Simulation	Nu– literature [21]
Circular	3.67	3.66
1:2	3.39	3.39
1:4	4.45	4.44
1:8	5.62	5.95

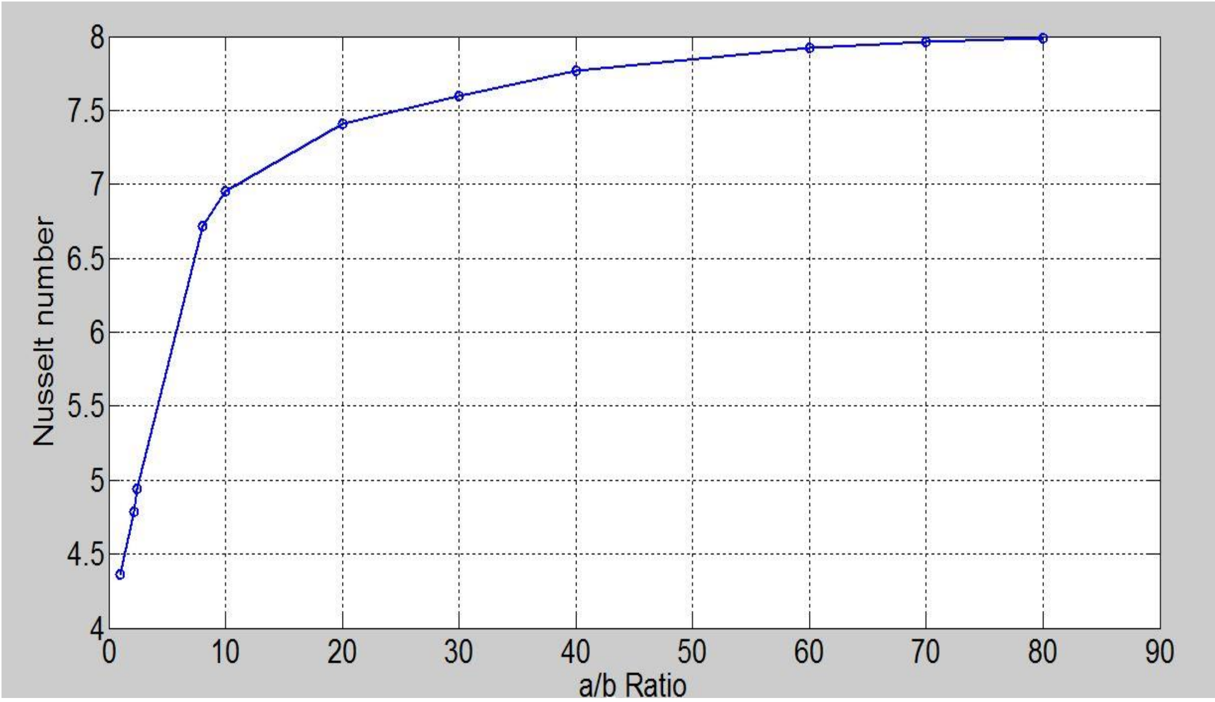
Table 2 Nusselt number validation for T boundary condition

Geometry	Nu – Simulation	Nu– literature [21]
Circular	4.40	4.36
1:2	4.1	4.11
1:4	5.29	5.35
1:8	6.41	6.6

Table 3 Nusselt number validation for H1 boundary condition

4.2 Results using Flat tube

The following figure shows the result of the several simulations



Fugure 4 Results shows Nusselt number variation depending upon flattening the geometry

5. Conclusions

1. Heat transfer increases sigficantly by introducing nano particles.
2. Flat cross section gives better heat transfer
3. The fully developed flow Nusselt number is higher for H1 boundary condition than for H2 or T boundary conditions for all aspect ratios considered in the study.
4. Nusselt number increases with increasing the aspect ratio and after reaching beyond 70 nusselt number becomes more or less constant.

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