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Computational Analysis of Mixed Convection of Internal Finned Tube

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Abstract— **Tubes with internal fins are widely been used in various in engineering applications such as heat exchanger, industrial mixer, nuclear reactor, automotives, solar air heater, etc. utilizing fins internal facilities an effective and efficient means to advances convective heat transfer execute differently depending on whether the flow is laminar or turbulent.**

This work concerns with computational analysis of pipe with internal finned subjected to mixed convection which evolves both natural and forced convective heat transfer. The analysis has been performed through FEV tool ANSYS FLUENT where SIMPLE algorithm has been used for solving governing equation of tube. The tube surface temperature has been evaluated for considering affect of various parameters such as fin height, fin configuration i.e. rectangular, T shaped triangular, heat flux, Reynolds number, and tube orientation i.e. horizontal and vertical. It has been observed that for triangular fin the heat transfer is significant and in vertical orientation due to buoyancy effect and boundary layer heat transfer I is more as compared to horizontal orientated. The obtained results have been compared with available literature and shows good agreement.

Key Words: Mixed convection, Nusselt number, Tube surface temperature

I. INTRODUCTION

In the field of heat transfer internally finned tubes are widely be used due to having effective and efficient means to advance convective heat transfer perform differently depending on whether the flow is laminar or turbulent. For laminar flow and heat transfer, comprehensive experimental and numerical investigations
have been performed for variable fluid properties, mixed convection and fin geometry. Heat transfer escalation techniques play a crucial role here while heat transfer coefficients are usually low for laminar flow in plain tubes. Designing a tubular heat exchanger with fins having different shapes and sizes is one such escalation technique discussed in this paper.

II. LITERATURE SURVEY

Jackson et al. 1989 present an exclusive review in the flied of mixed convection in vertical tubes. On the basis of analytical and experimental work the review has been categorized into which two subsection sections are there i.e. on type of flow laminar and turbulent. And illustrated the all governing equation associates with mixed convection

Cheng and Yang 1994 analyze the heat transfer and fluid flow of a vertical parallel-plate channel with fin array. The governing equation of mixed convection has been solved through power-law scheme. Results show that the stream-wise periodic difference of the cross-sectional area leads the flow and temperature fields to attain a periodically fully build up character after a number of modules from the inlet.

Gian Pieroet al. 1998 experimentally investigates the up flow turbulent mixed convection heat transfer in vertical pipes. A new method for the estimating the heat transfer coefficient in upward mixed convection heated flow has been proposed the result were compared with other numerical approach and found that they are in good agreement.

Sandar and Gross 2004 numerically studied the effect of fin-spacing in annular-finned tube heat exchangers. The unsteady flow and conjugate heat transfer are predicted through (RNG) based k–ε turbulence model and the results are compared with the existing correlations.

S.S. Saneet al. 2008 made their comparison and establish a match between experimental result and CFD software result for Notched fin array. They used notched fin array for single chimney flow pattern. Notched fin array shows the enhancement of more than 20 %. And the heat transfer coefficient is found in the range of 5%.

Ganguli et al. 2009 studied the variation in the heat transfer coefficient through natural convection for tall slender vertical geometries. The various parameters are like height with ranging 100-1000 mm, gap widths 5- 84.7 mm, temperature differences 5 - 90 K, and develop a Nusselt number correlation.

Fahiminia et al. 2011 investigate the laminar natural convection on vertical surfaces computationally. The CFD simulations are carried out using fluent software. Governing equations are solved using a finite volume approach. Relation between the velocity and pressure is made with SIMPLE algorithm

Pourya and Hooman 2013 numerically analyzed the turbulent convection in inclined pipes with significant buoyancy influence and found that Secondary flows are observed as a result of buoyancy force leading to non-uniformity of local heat transfer coefficient along the pipe periphery.

Das and giri 2014 numerically simulate the Non-Boussinesq laminar mixed convection in a non-isothermal fin array. Parametric analyses has been carried out fir Gr umber, Re number fin spacing and comparison between non-Boussinesq fluid flow and Boussinesq fluid flow has been done.

Dogonchi and Ganji 2016 studied the heat transfer through a moving fin in simultaneous convection–radiation mode. The result shows that the fin tip temperature augments with an increase in the heat generation gradient and a decline in the Peclet number and the radiation–conduction parameter.

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Senapati et al. 2016 investigate the natural convection over annular finned horizontal cylinder. They conclude that the NuD increases quickly with S/d at lower value and remains steady for higher value and the fin efficiency decreases significantly. Moreover, at fin spacing of 5-6mm optimum heat transfer has been achieved.

III. MATHEMATICAL MODEL

The Navier–Stokes equations can be written in the most useful form for the development of the finite volume method:

$$\rho\frac{Du}{Dt}=-\frac{\partial p}{\partial x}+div(\mu\, grad u)+S_{Mx} \quad (1)$$

$$\rho\frac{Dv}{Dt}=-\frac{\partial p}{\partial y}+div(\mu\, grad v)+S_{My} \quad (2)$$

$$\rho\frac{Dw}{Dt}=-\frac{\partial p}{\partial z}+div(\mu\, grad w)+S_{Mz} \quad (3)$$

Governing equations of the flow of a compressible Newtonian fluid

Continuity $\frac{\partial\rho}{\partial x}+div(\rho u)=0$ (

x-momentum

$$\frac{\partial(\rho u)}{\partial x}+div(\rho uu)=-\frac{\partial p}{\partial x}+div(\mu\, grad u)+S_{Mx} \quad (4)$$

y-momentum

$$\frac{\partial(\rho v)}{\partial y}+div(\rho vu)=-\frac{\partial p}{\partial y}+div(\mu\, grad v)+S_{My} \quad (5)$$

z-momentum

$$\frac{\partial(\rho w)}{\partial z}+div(\rho wu)=-\frac{\partial p}{\partial z}+div(\mu\, grad w)+S_{Mz} \quad (6)$$

Energy

$$\frac{\partial(\rho i)}{\partial t}+div(\rho iu)=-p\,div u+div(k\, grad T)+\Phi+S_i \quad (7)$$

Using various correlation FEV results are been compared analytically

$$h_f = f\frac{LV^2}{D_h 2g}$$

Where,

f is the friction factor for fully developed laminar flow

L: length of the pipe

V: mean velocity of the flow

d: diameter of the pipe

f is the friction factor for fully developed laminar flow:

$$f=\frac{64}{\mathrm{Re}} \text{ (For Re<2000)} \qquad \mathrm{Re}=\frac{\rho u_{avg} d}{\mu}$$

Cf is the skin friction coefficient or Fanning's friction factor.

For Hagen-Poiseuille flow: $C_f=\tau_{wall}\, l\,\frac{1}{2}\rho u_{avg}^2=\frac{16}{\mathrm{Re}}$

For turbulent flow: $\frac{1}{\sqrt{f}}=1.74-2.0\log_{10}\left[\frac{\varepsilon_p}{R}+\frac{18.7}{\mathrm{Re}\sqrt{f}}\right]$

Moody's Chart

R: radius of the pipe

εp: degree of roughness (for smooth pipe, εp=0)

$\mathrm{Re}\rightarrow\infty$: Completely rough pipe

IV. METHODOLOGY

A 20-node three-dimensional structural solid element was selected to model the pipe and pipe with internal fins. The pipe and pipe with internal fins was discretized into 38841 elements with 39726 nodes. The pipe and pipe with internal fins boundary conditions can also be provided in mesh section through naming the portion of modeled Pipe i.e Inlet, Outlet, Top wall, Bottom Wall, fins. The tube having length (L) 5m, diameter (D) 0.07 m the flow is considered to be laminar of Re = 1200. The tube wall is subjected to a constant wall heat flux of 200 W/m2. The geometrical model has been shown in figure 1

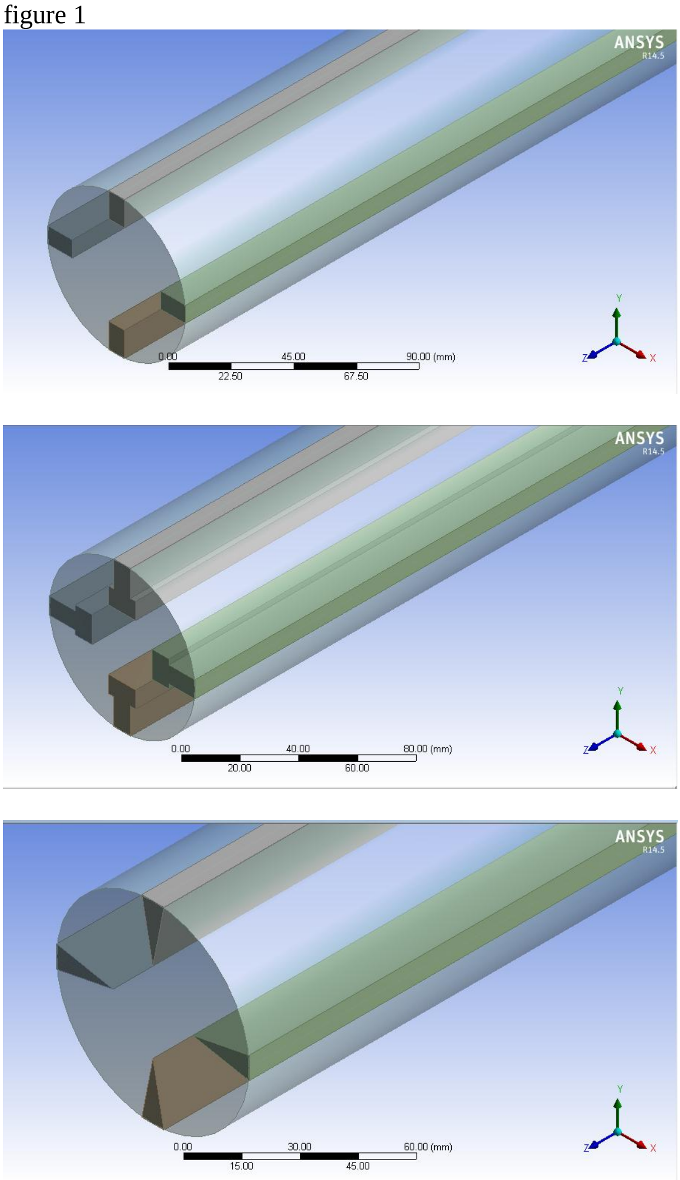

Figure 1 Model Geometry

V. RESULTS AND DISCUSSIONS

Using Ansys fluent the governing equation of pipe and pipe with internal fins i.e. The Navier stokes continuity equation has been solved. On the basis of this FEV work the thermo-hydrodynamic characteristic of pipe and pipe with internal fins has been evaluated with a grid size of 106 shows good

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agreement during the grid dependence test. Moreover, the performance of pipe and pipe with internal fins are illustrated in the corresponding results.

The precision of obtained results has been validated by comparing the present result with available literature of mohammed and salman [6] and Rout et al. [7] whose works are based on experimental, analytical and FVM results.

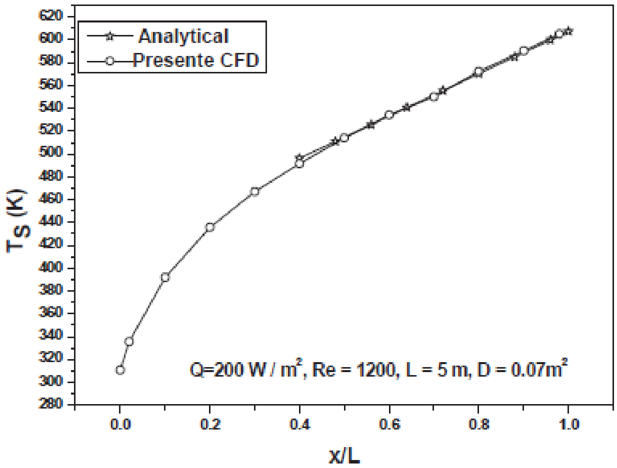

Figure 2 Smooth tube surface temperature distribution: a comparison of present CFD results with computed analytical solution [7]

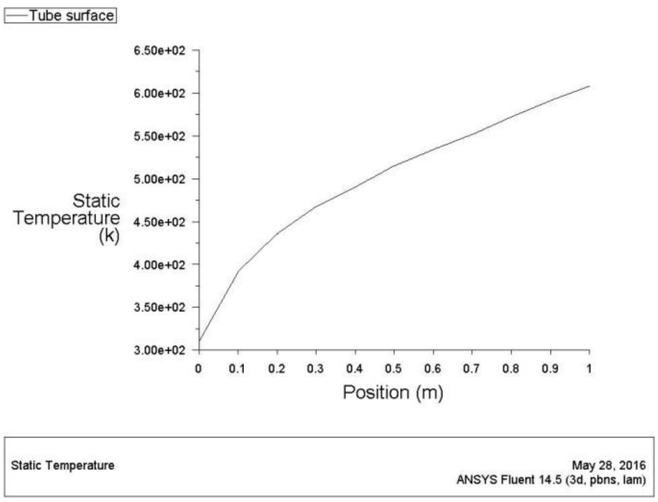

Figure 3 Validation of Tube surface temperature

The variation of the surface temperature with the pipe with internal fin may be affected by many inconsistent such as heat flux, Reynolds number, flow direction, pipe orientation, and the entrance section length. During analysis it has been observed that at the entrance of the tube, the tube surface temperature increases along the certain length, this is due to thickness of the boundary layer is zero. Then it steadily increases until it reaches maximum value where boundary layer fills the tube. During these processes the heat transfer results gradually decreases and the tube surface temperature (ts) gradually increases this is because of laminarization effect in the near wall region (buoyancy effect) and due to tube end losses. Various contour plots has been plotted for pressure, temperature velocity, Nusselt number as shown in figure

Fig. 2-3 shows the validation of tube with internal fin result obtained from the ANSYS Fluent. It has been seen that the obtained result for tube with internal fin with different boundary condition shows good agreement with the analytical, Simulation and FVM of available literature. The small variation is results is due to variation in grid sizing, operating condition, geometrical parameters, etc. but the obtained result shows the same trend so that the results are suitably verified.

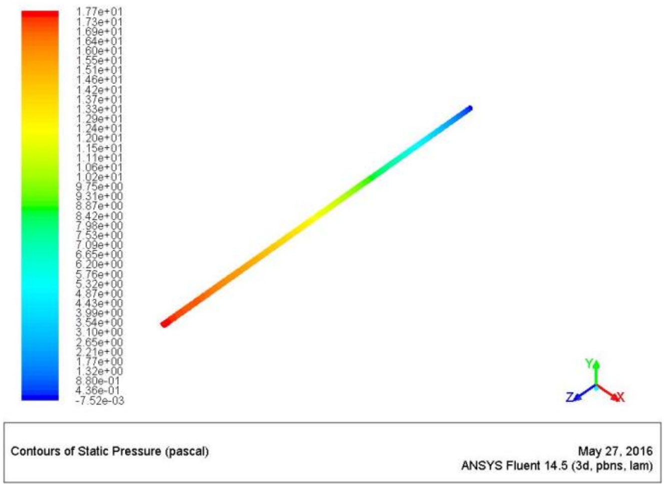

Figure 4 contour plot of pressure distribution

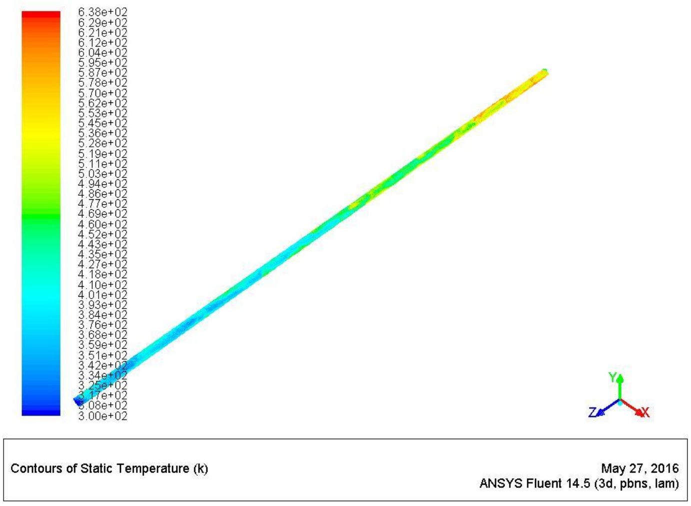

Figure 5 contour plot of temperature distribution

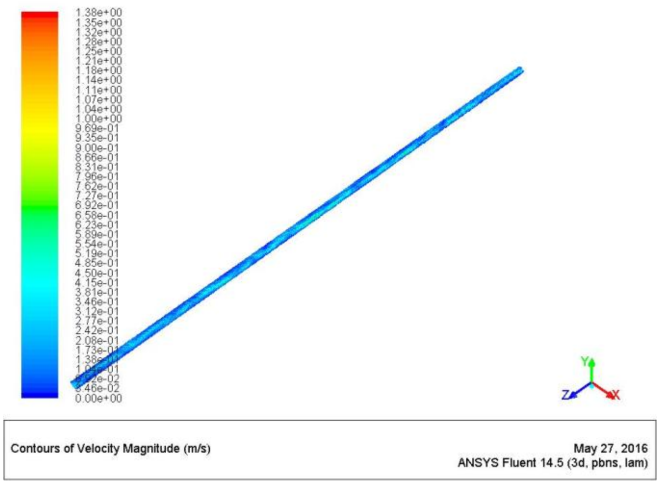

Figure 6 contour plot of velocity magnitude

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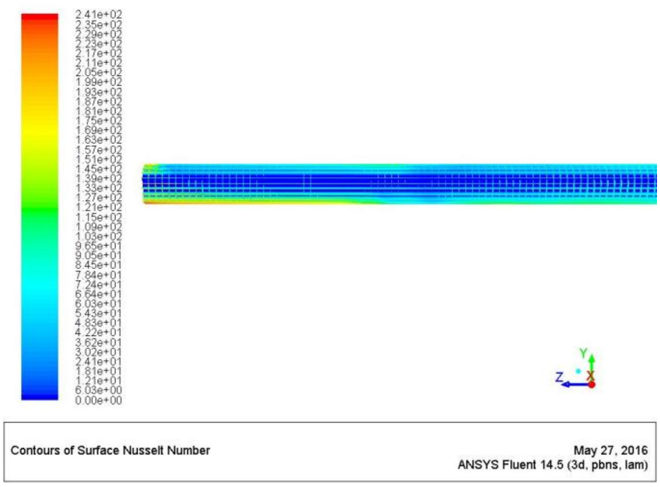

Figure 7 contour plot of Nusselt number

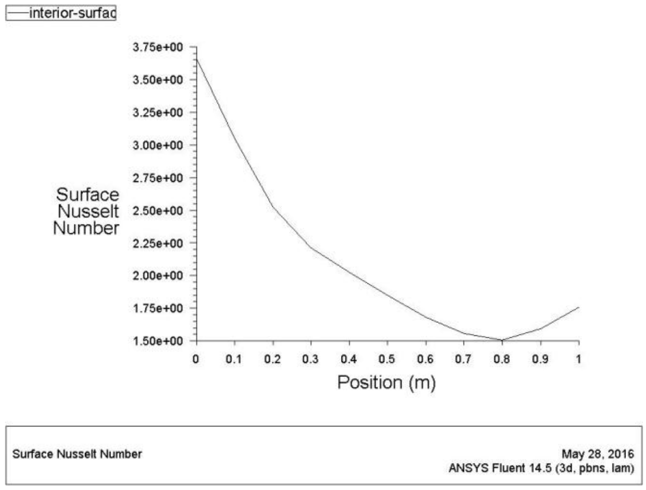

Figure 8Validation of Heat transfer coefficient

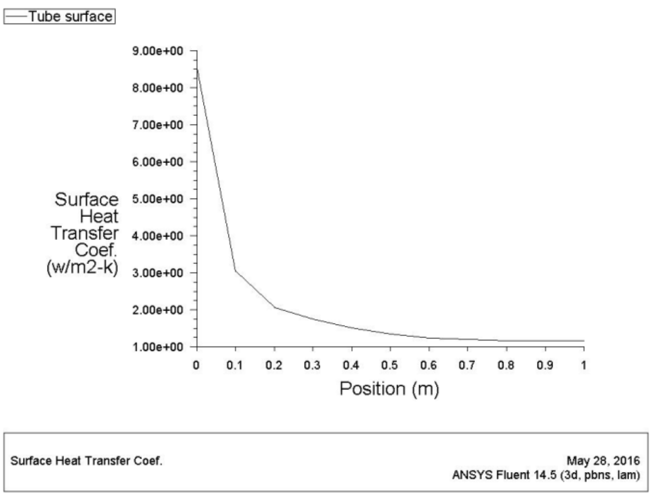

Figure 9 Validation of Heat transfer coefficient

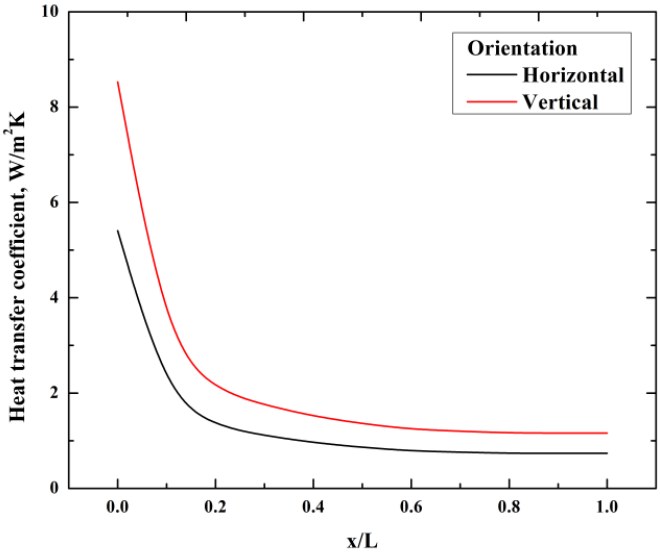

Figure 10 Effect of tube orientation on heat transfer coefficient

Figure 10 shows the effect of tube orientation on heat transfer coefficient. It has been observed that that the heat transfer coefficient significantly decreases throughout the tube length. It has also been observed that the vertical orientated tube has higher heat transfer coefficient as compared to horizontal orientated tube. It is because of buoyancy effect of thermal boundary layer during mixed convection process.

Figure 11 shows the effect of fin number on tube surface temperature. It has been observed that increasing number of fin within the tube. The tube surface temperature significantly decreases this is due to higher heat transfer rate to the air flows inside the tube when number of fin increased.

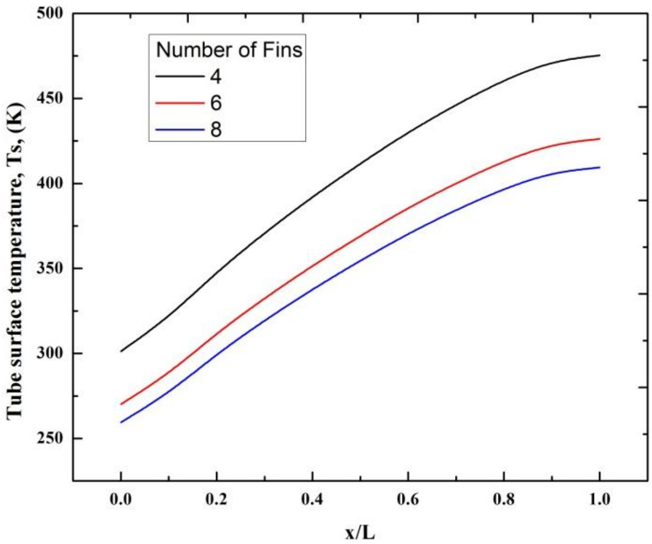

Figure 11 Effect of fin number on tube surface temperature

From the above it can be conclude that increasing number of fins the heat transfer rate increases. Therefore tube surface temperature decreases.

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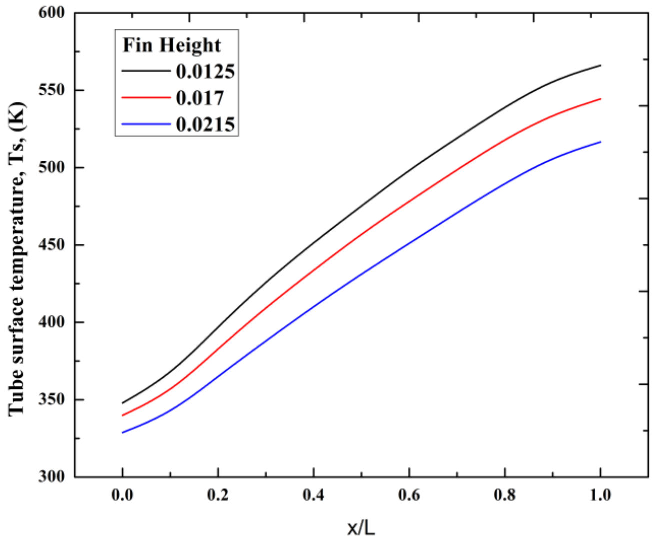

Figure 12 Effect of fin height on tube surface temperature

figure 12 shows the effect of fin height on tube surface temperature. It has been observed that increasing fin height the tube surface temperature decreases. This is because of more surface area through which heat is dissipated for the tube wall.

It can also be conclude that the rate of heat transfer increases as fin height increases.

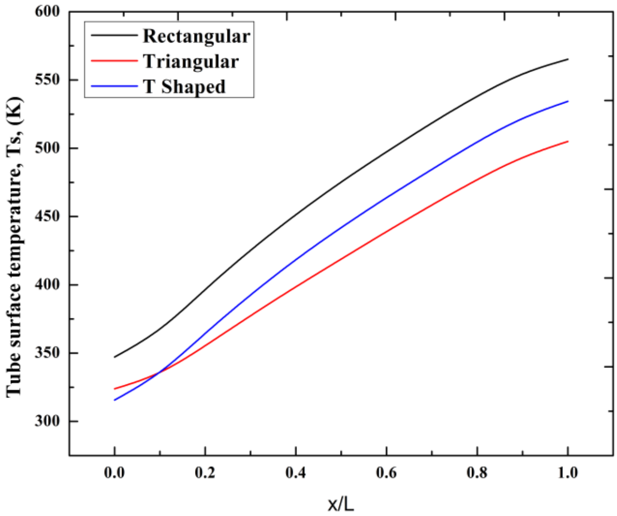

Figure 13 Effect of fin configuration on tube surface temperature

Figure 13 shows the effect of fin configuration on tube surface temperature. It has been observed that the fin with rectangular configuration has higher tube surface temperature, while for triangular configuration the tube surface temperature is comparative low from other configuration i.e. rectangular and T shaped.

This is also due to triangular fin has higher average heat transfer coefficient, therefore leads to higher heat transfer rate.

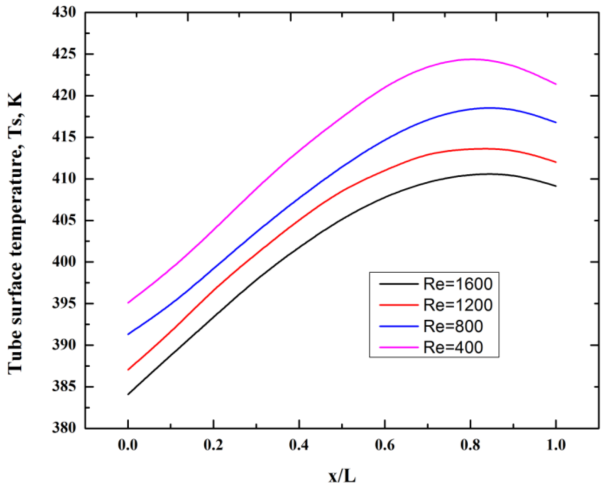

Figure 14 Effect of Reynolds number on tube surface temperature

Figure 6.13 shows the effect of Reynolds number on tube surface temperature. It has been observed that increasing Reynolds number the tube surface temperature significantly decreases. This is because of the forced convection domination on the heat transfer process at higher Reynolds number.

From the above it can be revealed that increasing Reynolds number tube surface temperature decreases as due to forced convection.

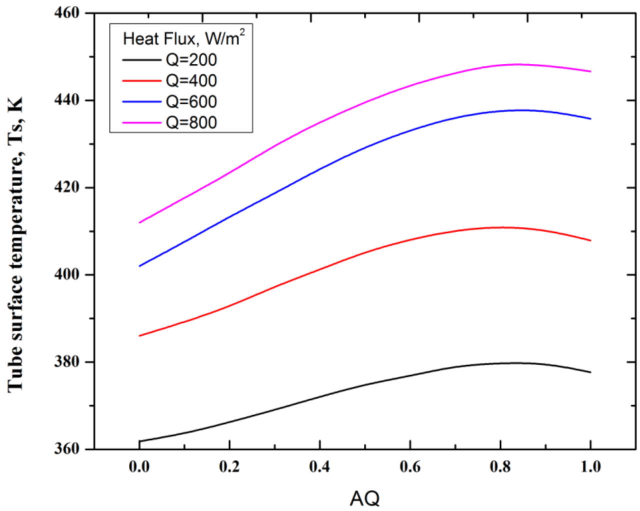

Figure 15 Effect of heat flux on tube surface temperature

Figure 15 shows the effect of heat flux on tube surface temperature. It has been observed that the tube surface temperature significantly goes on increasing throughout the tube length because of thickness of boundary layer is very low. However, it has also been observed that increasing heat flux the tube surface temperature increases in higher rate. This is due to the development of the thermal boundary layer more rapidly due to buoyancy effect as the heat flux increases for the same Reynolds number

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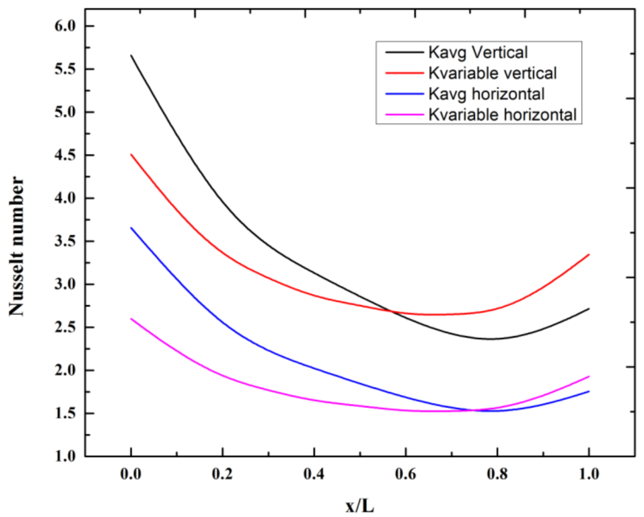

Figure 16 Effect of average thermal conductivity and variable thermal conductivity on tube surface Nusselt number

Figure 17 Effect of average thermal conductivity and variable thermal conductivity on tube surface Nusselt number. It has been observed that for vertical oriented tube average thermal conductivity is significantly higher as compared to horizontal orientated variable thermal conductivity. It has also been seen that the Nusselt number drastically goes on decrease die to weaker thermal boundary layer and then increases due to buoyancy effect. The same trend has been observed for both average thermal conductivity and variable thermal conductivity.

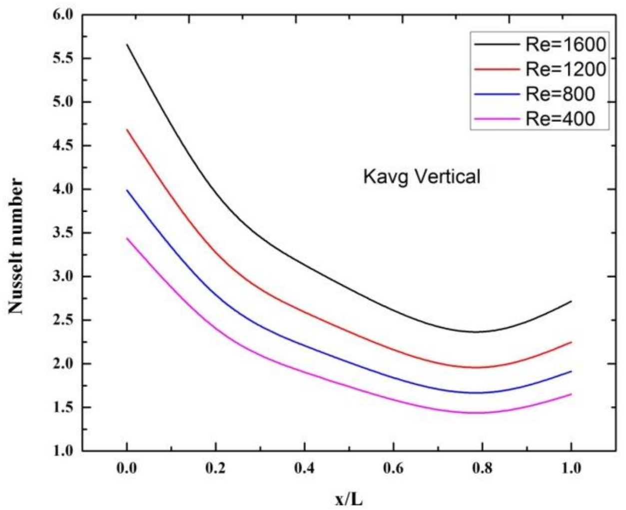

Figure 18 Effect of Reynolds number on tube surface Nusselt number

Figure 18 shows the effect of Reynolds number on tube surface Nusselt number. It has been observed that the keep variable thermal conductivity constant for vertical orientated tube. The Nusselt number significantly goes on decreasing throughout the tube length till 0.8 and then after increases. This is because of dominance of convective heat transfer over conductive transfer at the inlet of the tube therefore Nusselt number decreases and increase in Nusselt number again signifies the effect of convective heat transfer is becoming more pronounced due to the buoyancy effect.

VI. CONCLUSION

CFD analysis of pipe with internal fin has been carried out using ANSYS FLUENT. The affect of various parametric such as fin height, Reynolds number, fin configuration, pipe orientation i.e horizontal and vertical, etc has been examined on in terms of heat transfer coefficient, tube surface temperature, Nusselt number. On these basis various conclusions has been drawn which are as follows:

- The tube surface temperature for low Reynolds number would be higher than that for high Reynolds number, for the same heat flux, this is due to the free convection domination.
- The tube surface temperature decreases when the pipe orientated vertically to horizontally due to natural convection. While the tube surface temperature was increased from vertical to horizontal orientation this is due to the forced convection is leading.
- For constant heat flux the pipe with internal fin having vertical orientation have higher Nusselt number that the horizontal oriented pipe.
- Increasing fin height the tube surface temperature decrease significantly.
- The heat dissipation for triangular fin has remarkably more as compared to other fin configurations i.e. T shape and rectangular.
- Increasing number of fins tube surface temperature decreases.
- For vertical orientated pipe heat transfer coefficient is significantly high has compared to horizontal.
- At constant Reynolds number increasing heat flux tube surface temperature increases.

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