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SIMULATION ON HEAT TRANSFER ENHANCEMENT IN A CIRCULAR TUBE FOR LAMINAR FLOW WITH AND WITHOUT INSERT

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Abstract *- Theoretical simulation was carried out to study the heat transfer enhancement in a circular tube with and without insert. Among different techniques passive type enhancement has been applied in this study using both horizontal and cross strip inserts. The mathematical models are formulated for the inserts in tube using the fundamental heat transfer principles. Temperature profiles of different cross sections are simulated for laminar flow through the circular tube with and without inserts. It is found that heat transfer coefficient and Nusselt Number remains almost constant for different Reynolds number. However, heat transfer rate increases linearly with different Reynolds Number in case of both insert and no insert. It could be due to the fact of mathematical simplification and thin insertions that are being used for laminar flow.*

Keywords: Heat transfer enhancement technique, Passive method, Horizontal, Cross strip insert.

1. INTRODUCTION

Heat transfer enhancement is the process of improving the performance of a heat transfer system. It generally means increasing the heat transfer coefficient. The performance of heat exchanger depends how effectively heat is utilized [1]. Heat transfer enhancement technology has been developed rapidly and employed in a wide variety of engineering problems, such as condensing gas boiler refrigeration, automotives, solar water heater process industry and chemical industry etc [2,3]. The convection, utilization, and recovery of energy in every industrial, commercial, and domestic application involve a heat transfer process. Improved heat exchanger over and above that use on standard practice, can significantly improve the thermal efficiency in such applications as well as the economics of their design operation. Researchers throughout the world have made great efforts to the development of heat transfer augmentation, which are classified either as passive or active [4]. Passive heat transfer augmentation methods as stated earlier does not need any external power input. In the convective heat transfer one of the ways to enhance heat transfer rate is to increase the effective surface area and residence time of the heat transfer fluid [5]. Therefore, they are more frequently applied in practical situations. Various kinds of tube inserts, such as the twisted tape, conical ring, porous media inserts, etc., belong to the passive techniques [6]. This type of inserts generates swirls which enhance the fluid mixing of the near-wall and central regions [7]. Bergles [8, 9] presented a comprehensive survey on heat transfer enhancement by various techniques. Among many techniques (both passive and active investigated for augmentation of heat transfer swirl flow devices) has been shown to be very effective, due to imparting of helical path to the flow.

Turki *et al.* [10] investigated the effect of the blockage ratio on the laminar flow in a channel with built-in square cylinder. The results show that the critical value of Reynolds number relative to transition from steady to periodic flow increases by increasing blockage ratio and Reynolds number, the square cylinder has a stable transversal posture to the flow. Liao *et al.* [11] studied the heat transfer and frictional factor characteristics in tubes with three-dimensional internal extended surfaces and twisted tape inserts. The experimental results showed that this of was of particular advantage to enhance the convective heat transfer for laminar tube side flow of highly viscous fluid.

In the present paper, heat transfer enhancement with straight and cross strip inserts flows of fluid through circular tube is investigated for laminar flow. A passive heat transfer enhancement method is used for the investigator. The mathematical models are formulated for horizontal and cross strip inserts. And laminar flow in a circular tube is considered for the numerical simulation.

2. PHYSICAL MODEL

A schematic of the physical model is shown in Fig. 1. Air is selected as the working fluid. The horizontal and the cross strip are inserted within the whole length inside the circular tube. The tube surface is considered smooth with no friction. Laminar flow of air is induced from some external source. Turbulence of the working fluid is not considered to simplify the analysis. However, turbulent flow is aimed to analyze in next study.

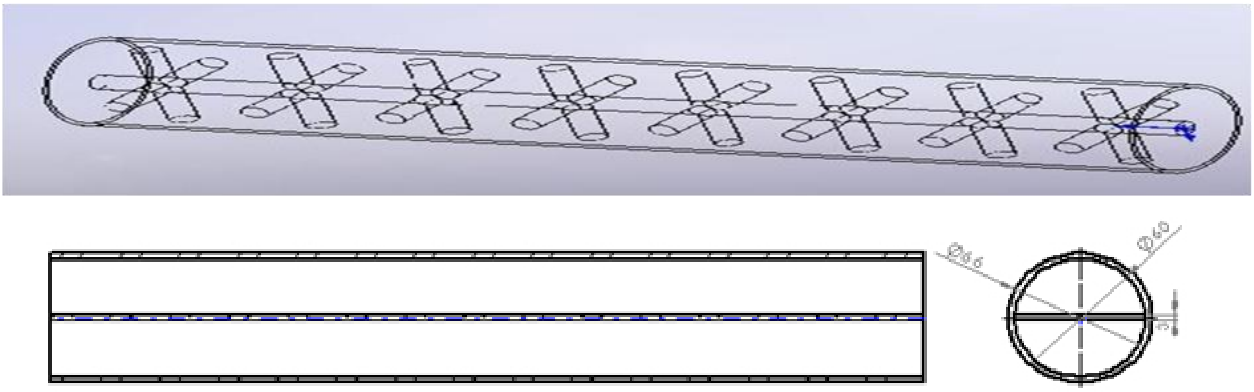

Fig 1: Circular tube fitted with Cross & Horizontal strip inserts.

3. MATHEMATICAL MODELS

This section provides governing equations and boundary conditions for the flow through the tube. Consider the tube flow system in Fig 2. In this paper, the laminar heat transfer in the circular tube with & without inserts has been investigated. The heat transfer coefficient is calculated under developed flow condition when flow remains laminar. Here the governing equations are formulated from the rigor of heat transfer principles. The pressure forces are balanced by the viscous shear forces for the flow of air in the circular tube [13].

Assuming, heat flux is constant at the tube wall, i.e.

$$\frac{dq_w}{dx} = 0$$

Appling the first boundary condition,

at $r = 0$, $\frac{\partial T}{\partial x} = 0$

Appling the second boundary condition,

at $r = r_0$, $T = T_w$

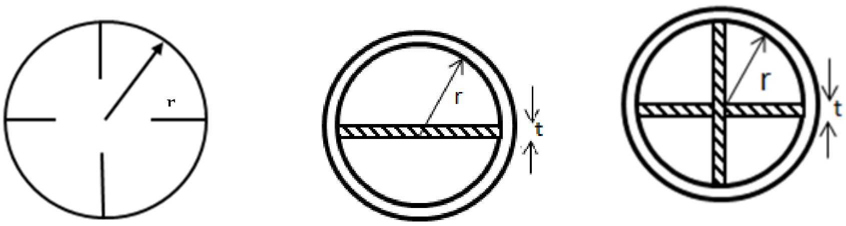

Fig 2. Cross sectional area of circular tube with & without inserts.

3.1 Temperature profile equation for laminar flow without insert:

The pressure force and viscous shear force can be balanced as follows,

$$\pi r^2 dp = \tau 2\pi r dx = 2\pi r \mu dx \frac{du}{dr}$$

$$du = \frac{1}{2\mu} r \frac{dp}{dx} dr$$

Integrating the both sides:

$$u = \frac{1}{4\mu}\frac{dp}{dx} r^2 + C_1 \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (i)$$

Appling the boundary conditions:
at $r = r_0$, $u = 0$

$$\frac{u}{u_0} = \left(1 - \frac{r^2}{r_0^2}\right) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (ii)$$

$$T = T_w + \frac{\partial T}{\partial x}\left[\frac{1}{\alpha}u_0\left(\frac{r^2}{4} - \frac{r^4}{16r_0^2}\right) - \frac{1}{\alpha}u_0\left(\frac{3r_0^2}{16}\right)\right]$$

Where, $Y = \frac{1}{\alpha}u_0\left(\frac{r^2}{4} - \frac{r^4}{16r_0^2}\right) - \frac{1}{\alpha}u_0\left(\frac{3r_0^2}{16}\right)$

Therefore, the temperature equation can be written as

$$T = T_w + \frac{\partial T}{\partial x}\left[Y\right] \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (iii)$$

This is the governing equation of temperature profile for laminar flow in a tube without insert. Here, T is the temperature at any radial distance, T_w is the temperature at the wall, u_0 is the velocity at the centre, r_0 is the inner radius of the tube and r is the radial distance.

For the temperature distribution given in equation (*iii*), the bulk temperature is a linear function of x because the heat flux at the wall is constant.

Now, the bulk temperature can be calculated as

$$T_b = T_c + \frac{7}{96}\frac{u_0 r_0^2}{\alpha}\frac{\partial T}{\partial x}$$

and the wall temperature is,

$$T_w = T_c + \frac{3}{16}\frac{u_0 r_0^2}{\alpha}\frac{\partial T}{\partial x}$$

The heat transfer coefficient is,

$$h = \frac{k}{T_w - T_b}\frac{u_0 r_0}{4\alpha}\frac{\partial T}{\partial x}$$

Thus, the Nusselt number (*Nu*) and the heat transfer rate can be calculated as belows.

$$Nu_d = \frac{hd}{k} \quad \text{and} \quad q = h(T_w - T_b)$$

3.2 Temperature profile equation for laminar flow with horizontal strip insertion:

$$T = T_w + \left(\frac{9}{7.88}\right)\frac{\partial T}{\partial x}\left[\frac{1}{\alpha}u_0\left(\frac{r^2}{4} - \frac{r^4}{16r_0^2}\right) - \frac{1}{\alpha}u_0\left(\frac{3r_0^2}{16}\right)\right]$$

Again, $Y = \frac{1}{\alpha}u_0\left(\frac{r^2}{4} - \frac{r^4}{16r_0^2}\right) - \frac{1}{\alpha}u_0\left(\frac{3r_0^2}{16}\right)$

Therefore, $T = T_w + \left(\frac{9}{7.88}\right)\frac{\partial T}{\partial x}[Y]$ .... (*iv*)

This is the governing equation of temperature profile for laminar flow with horizontal strip.

3.3 Temperature profile equation for laminar flow with cross strip insert:

$$T = T_w + \left(\frac{9}{5.76}\right)\frac{\partial T}{\partial x}\left[\frac{1}{\alpha}u_0\left(\frac{r^2}{4} - \frac{r^4}{16r_0^2}\right) - \frac{1}{\alpha}u_0\left(\frac{3r_0^2}{16}\right)\right]$$

Again, $Y = \frac{1}{\alpha}u_0\left(\frac{r^2}{4} - \frac{r^4}{16r_0^2}\right) - \frac{1}{\alpha}u_0\left(\frac{3r_0^2}{16}\right)$

Therefore, $T = T_w + \left(\frac{9}{5.76}\right)\frac{\partial T}{\partial x}[Y]$ .... (*iv*)

This is the governing equation of temperature profile for laminar flow with cross strip.

4. RESULTS & DISCUSSION

4.1 Analysis of tube flow with no insertion

Considering the air is flowing through a tube. In this section, the temperature profiles are simulated using the mathematical model. The mathematical models are simulated using FORTRAN language. The temperature profiles are plotted for the tubes with and without inserts. The heat transfer co-efficient (*h*), Nusselt number (Nu) and heat transfer rate (*q*) are also plotted against the Reynolds number (*Re*) for the tube using with insert and without insert.

The graphical representation of temperature profiles are shown in Fig. 3 with respect to radius at different velocity with no insertion. It is found that temperature at any point in the tube increases with the increase of radial distance.

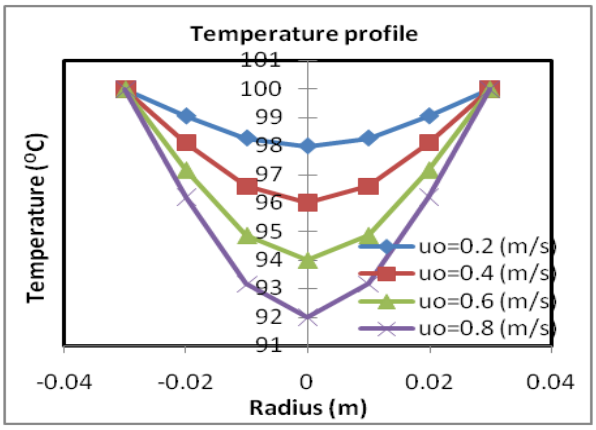

Fig 3: Comparison of temperature with radius using no insertion.

Here, it is shown (Fig. 4) that there is no change of heat transfer coefficient with Reynolds number because of laminar flow analysis. Due to laminar flow, the velocity of fluid flow through the tube is small and therefore, the interaction among fluid particles is also low. Due to low velocity, the heat transfer among fluid particle occurs very slowly. This is why heat transfer coefficient becomes approximately similar at different Reynolds number.

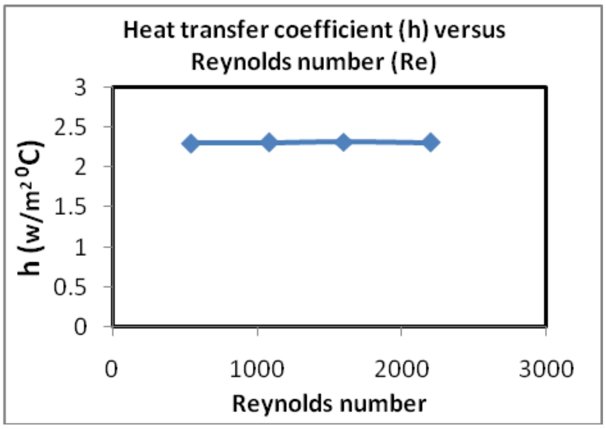

Fig. 4: Comparison of heat transfer coefficient with Reynolds number using no insertion.

It is shown is Fig. 5 that there is no change of Nusselt number with respect to Reynolds number. We know that Nusselt number is the function of heat transfer coefficient. Since there is no change of heat transfer coefficient (Fig. 4) with changing Reynolds number, the heat transfer coefficient is almost same for varying Reynolds number.

It is shown in Fig. 6 that heat transfer rate linearly increases with the increase of Reynolds number with no insertion. The heat transfer rate linearly increases since the temperature difference between wall temperature and bulk temperature is increasing.

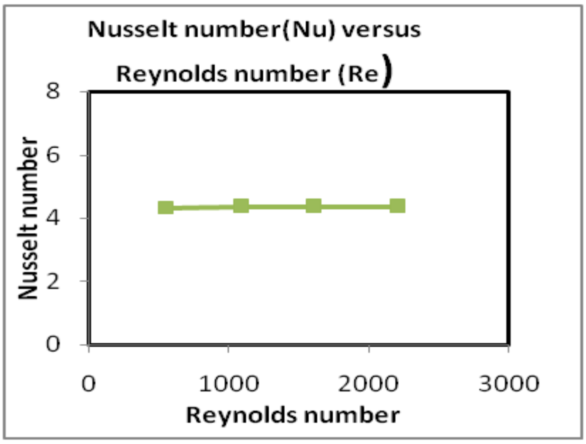

Fig 5: Comparison of Nusselt number with Reynolds number using no insertion.

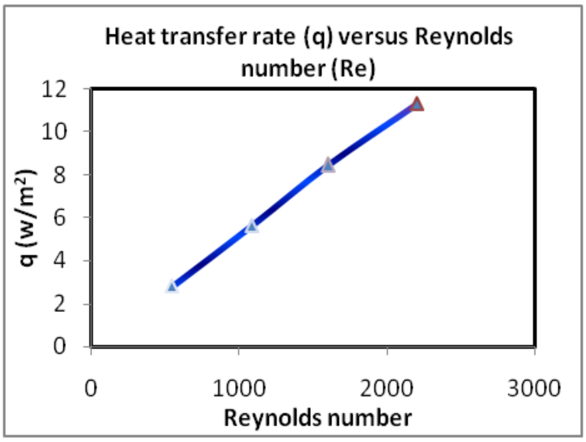

Fig 6: Comparison of heat transfer rate with Reynolds number using no insertion.

4.2 Analysis for tube flow with horizontal strip inserts:

It is shown in Fig. 7 that the temperature at any point in the tube increases with increasing of radial distance in case of horizontal strip inserts. The graph for horizontal strip insert is almost same as with no insert.

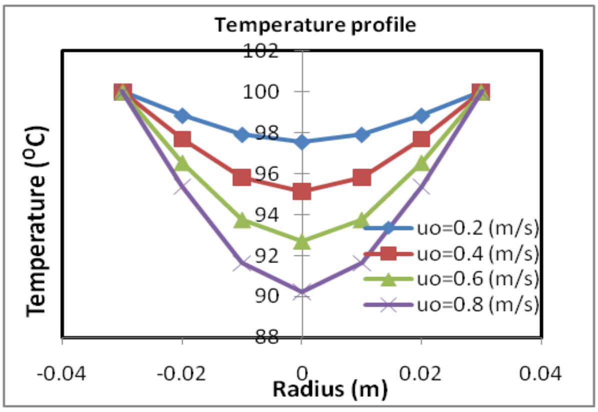

Fig. 7: Comparison of temperature with radius using horizontal strip insert.

It is shown in Fig. 8 that there is no change of heat transfer coefficient with Reynolds number. The main purpose for using the insertion in a tube is to increase the heat transfer coefficient. However, it can be seen from the analysis that heat transfer coefficient remains almost same. Although the movement area of the fluid particles decreases due to the strip insert in the tube, the heat transfer coefficient remains constant because of the laminar flow. A similar phenomenon happens in case of Nusselt number (Nu) versus Reynolds number plot (Fig. 9).

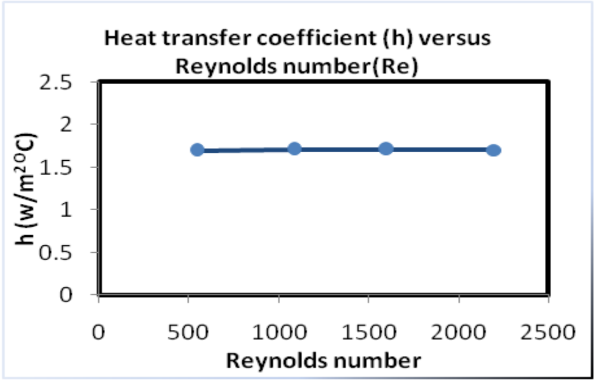

Fig. 8: Comparison of heat transfer coefficient with Reynolds number using horizontal strip insert.

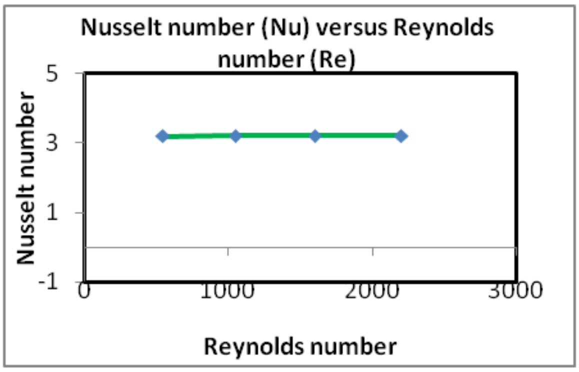

Fig 9: Comparison of Nusselt number with Reynolds number using horizontal strip insert.

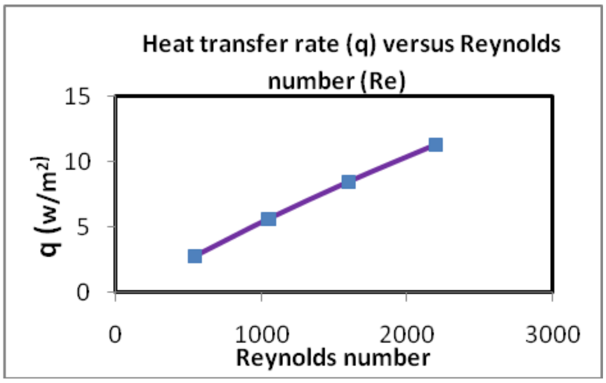

Fig. 10: Comparison of heat transfer rate with Reynolds number using horizontal strip insert.

Using the horizontal strip insert, heat transfer rate linearly increases with the increase of Reynolds number as shown in Fig. 10.

4.3 Analysis for tube flow with cross strip inserts

It is shown from Fig. 11 that temperature at any point in the tube increases at different radial distance. For lower range to higher velocity, temperature at the centre of the tube gradually decreases. It means that heat transfer rate slightly increases with the increase of velocity.

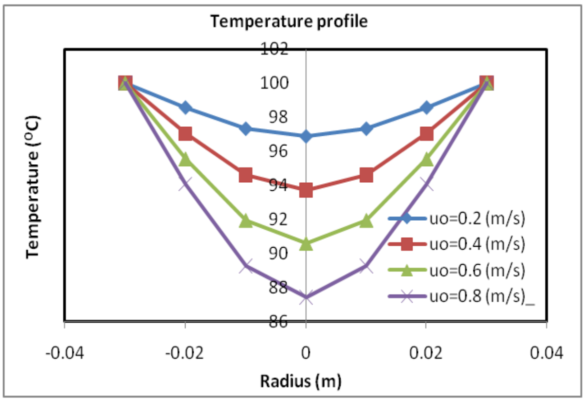

Fig. 11: Comparison of temperature with radius using cross strip insert.

It is shown in Fig. 12 that there is no change of heat transfer coefficient with the increase of Reynolds number. The higher velocity of fluid flow causes higher collision among the fluid particles. Due to higher collision among fluid particles, heat transition rate increases and it is possible in case of turbulent flow. Since our analysis is limited to laminar flow, the velocity of fluid flow is lower and therefore, the heat transfer among the fluid particles is poor. This is why there is no change of heat transfer coefficient with the increase of Reynolds number.

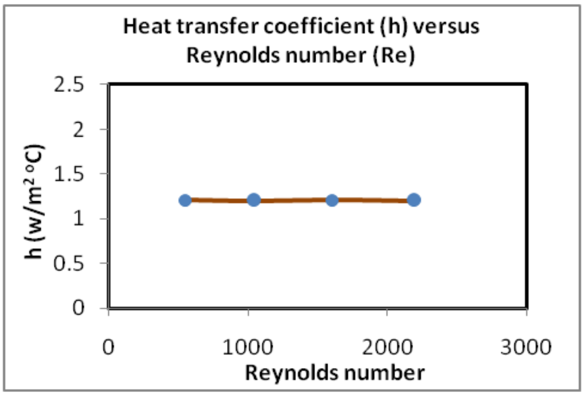

Fig. 12: Comparison of heat transfer coefficient with Reynolds numbers using cross strip.

It is shown in Fig. 13 that there is no change of Nusselt number with changing Reynolds number. Since the analysis is for laminar flow and also simple insert is used for this technique which does not affect on the flow enhancement.

It is also shown in Fig. 14 that heat transfer rate increases with the increase of Reynolds number.

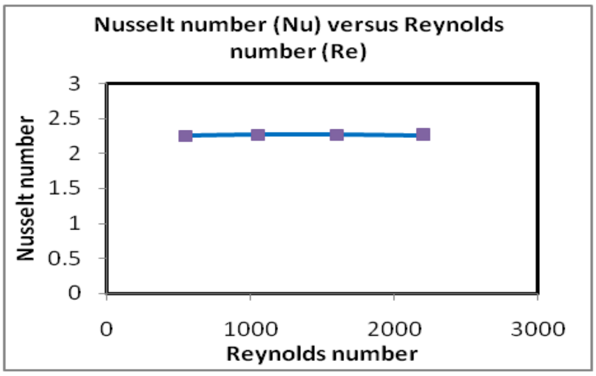

Fig. 13: Comparison of Nusselt number with Reynolds number using cross strip inserts.

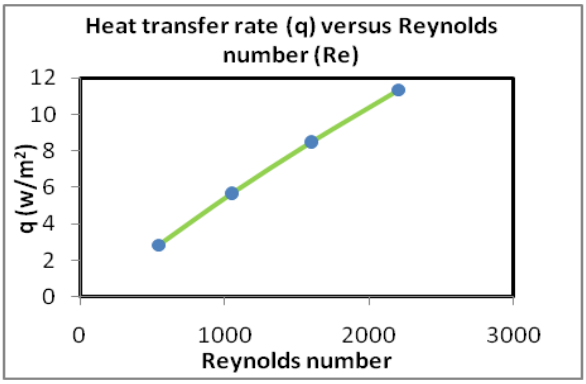

Fig. 14: Comparison of heat transfer rate with Reynolds numbers using cross strip insert.

4.4 Combined analysis for tube flow with insert and without insert

It can be seen from Fig. 15 that the heat transfer coefficient remains almost constant for different Reynolds number in case of both insert and no insert. However, the coefficient is higher in case of flow through tube without insert than with insert. It is due to the increase in temperature difference between wall temperature and bulk temperature. Similarly heat transfer rate increases (Fig. 17) with the increase of Reynolds number. For this reason the heat transfer coefficient is also almost constant for different Reynolds number.

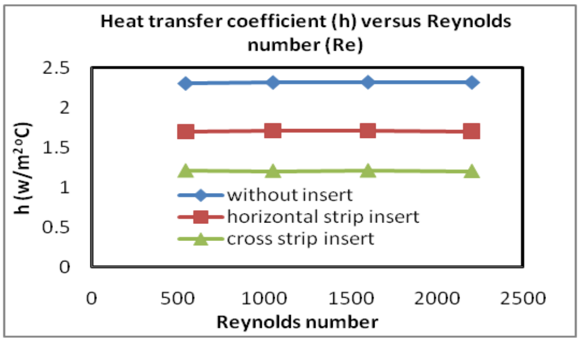

Fig. 15: Comparison of heat transfer coefficient with Reynolds number using insert and without insert.

Now, it can be seen from Fig. 16 that no change of Nusselt number for varying Reynolds number. However, Nusselt number is higher in case of flow through tube without insert than with insert. Already it has explained that how heat transfer coefficient has become similar value. Since at the governing equation of Nusselt number, only heat transfer coefficient parameter is used as a variable and other parameters are constant. Due to similar values of heat transfer coefficient, Nusselt number also shows similar values.

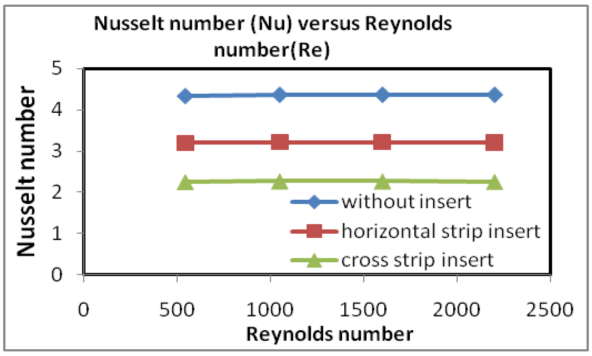

Fig 16: Comparison of Nusselt number with Reynolds number using insert and without insert.

It is shown in Fig. 17 that heat transfer rate is increasing linearly with the increase of Reynolds number. It is very interesting that heat transfer rate is simultaneously increasing for insert and without insert. Since air is flowing through a tube in a laminar tube flow. Due to laminar flow, velocity of air through tube is low and also simple insert is used for this analysis. For this reason, it does not affect on flow enhancement. However, heat transfer rate is same for both insert and without insert.

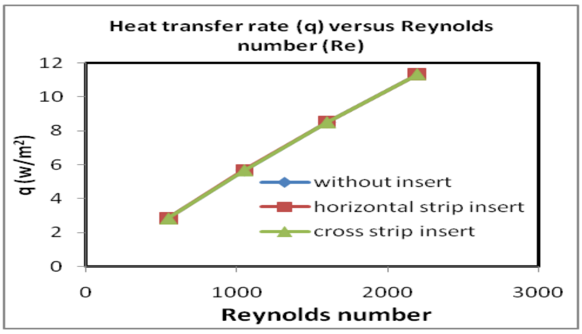

Fig. 17: Comparison of heat transfer rate with Reynolds number using insert and without insert.

5. CONCLUSION

Heat transfer enhancement and other criterion of laminar flow in a circular tube with horizontal and cross strip inserts have been investigated numerically. It has been observed from the results that the heat transfer rate have been increased equally at different Reynolds number for both insert and without insert. However, the heat transfer coefficient and the Nusselt number are almost similar in values with varying Reynolds number due to laminar flow. Therefore, it is recommended to study for higher velocities in turbulent region or by creating turbulence required to break down the isolative laminar fluid layer. This can be accomplished by using a corrugated insert surface instead of a flat insert surface.

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8. NOMENCLATURE

Symbol	Meaning	Unit
A	Surface area of heating tube	(m^2)
d	Diameter of the heating tube	(m)
L	Length of the tube	(m)
T_w	Wall temperature	(^0C)
T_b	Bulk temperature	(^0C)
ΔT	Temperature difference	(^0C)
T	Temperature	(^0C)
ρ	Fluid density	kg/m^3
μ	(Absolute)dynamic fluid viscosity	kg/m.s
υ	Kinematic fluid viscosity	m^2/s
K	Thermal conductivity	W/m ^0C
C_P	Specific heat	KJ/kg ^0C
Q	Heat transfer rate	W
h	Convective heat transfer coefficient	$W/m^2\ ^0C$
u_0	Velocity	m/s
α	Thermal diffusivity	
q	Heat flux	W/m^2
Pr	Pandalt number	-
Nu	Nusselt number	-
Re	Reynolds number	-