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# Heat Transfer Enhancement of Circular Pipe using Serrated Plate Inserts

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***Abstract*:**

*The heat exchangers are used to enhance heat transfer by providing high heat fluxes or heat transfer coefficient. The second law efficiency increases and the entropy generation decreases by reducing the temperature driving force which is caused with the increase in heat transfer coefficient. The present experimental work is carried out with copper and aluminium serrated plate by varying groove (slot) thickness from 6mm to 8mm and by varying thickness of plate from 3mm to 5mm respectively. The inserts when placed in the path of the flow of the fluid, create a high degree of turbulence resulting in an increase in the heat transfer rate and the pressure drop. The work includes the determination of friction factor and heat transfer coefficient for various inserts with varying thickness and different materials. The results of varying thickness in serrated plates with two materials are compared with the smooth tube and found that copper with 5mm thick and 6mm serrated slot has optimum results in terms of heat transfer coefficient and friction factor.6mm slot and 5mm plate thickness copper shows higher Nusselt number of 60% to 76% and lesser friction factor by 39% as compared to without inserts*.

**Keywords**: *Enhancement efficiency, heat transfer, serrated plate inserts, turbulent; pressure drop.*

## 1. Introduction

Heat transfer enhancement techniques are used in order to improve the heat transfer or thermal performance of heat exchangers. Till date, many attempts have been made to reduce the size and cost of heat exchangers. The basic aim of heat transfer enhancement is to provide heat transfer coefficient. Heat transfer enhancement techniques can be divided into two group passive and active techniques. The passive techniques have been usually preferred by many researchers since no additional external power is required as extended surfaces, rough surfaces and swirl flow devices (SibelGunes, et. al,2010). Coiled wire, twisted tape or other swirl flow devices inserted into a flow provide swirling flow and periodic redevelopment of the boundary layer, increases effective heat transfer area and turbulence intensity.

The swirl induced tangential flow velocity component causes improved fluid mixing between the tube core and the wall region nearby. Thus enhancing the heat transfer by rapid fluid mixing, on the other hand, the swirl induced heat transfer enhancement brings along inevitable shear stress and pressure loss in coiled wire or twisted tape inserted tube. For a long time, many research have been carried out concerning the effect of wire coil inserts on the heat transfer and pressure drop.(A Garcia et al 2005)Several heat transfer enhancement techniques are utilized in order to improve the heat transfer or thermal performance of heat exchangers. The goal of the heat transfer enhancement is to provide high heat fluxes or heat transfer coefficient. The second law efficiency increases and the entropy generation decreases by reducing the temperature driving force which is caused with the effect of heat transfer coefficient increase. Generally, heat transfer enhancement techniques can be divided into two groups (passive and active techniques).

Plain fin surfaces are characterized by long uninterrupted flow passages with performance comparable to that obtained inside long circular tubes The plain fins that are most commonly used have flow channels with either a rectangular or triangular cross-section, corresponding to surfaces A and B in Figure 28. The enhancement in heat transfer achieved with plain fins is due mainly to increased area density, rather than any increase in the heat transfer coefficient. Plain fins require a smaller flow frontal area than interrupted surfaces (i. e. offset strip fins and louvered fins) for given values of heat duty, pressure drop, and flow rate, but the flow length with plain fins Louvered fm surfaces (Figure 28d) are commonly used in automobile radiators. The louvered fm geometry consists of an interrupted surface similar to that of the offset-strip fin. However, the slit strips of louvered fins are not completely offset. Instead, the slit fin is rotated between 2QO and room relative to the direction of the airflow. Most radiators use a louver strip width of 1.0 to 1.25 mm [2]. For equal strip width, the louvered fin geometry provides enhancement comparable to that of offset strip fins. Moreover, louvered fins are less expensive than offset strip fins for large-quantity production, because of their ease of manufacture using high-speed mass production technology.

Tabulators in forms of conical and circular rings belong to one important group of heat transfer enhancement devices. Mechanistically, turbulator increases the composite velocity, enhances the radial turbulent fluctuation which causes an efficient eruption of the thermal boundary layer. Turbulators in several shapes have been proposed. Yakut et al. [1,2] studied the heat transfer, friction factor and thermal performance characteristics in a circular tube fitted

withconical-ring turbulators under a uniform heat flux conditions. The analysis of entropy generation was alsoconducted for a practical use evaluation. Durmus [3] investigated the heat transfer and pressure drop in a heat exchanger tube fitted with the cut out conical turbulators with different installing angles. It was found that the turbulators considerably enhanced heat transfer and friction factor over those of the plain tube.

Heat transfer augmentations attributed to several types of V-nozzle/conical turbulators and snail with/without free-spacing entry were reported by Eiamsa-ard and Promvongeand Promvonge and Eiamsa-ardKongkaitpaiboon et al. evaluated performances of the perforated conical-ring turbulators with different perforation diameters and perforation numbers. Ozceyhanet al. reported the influences of the space between the circular cross sectional ring turbulators on the heat transfer rate and friction factor characteristics in turbulence region. It was suggested that the tube with circular cross sectional ring turbulators provided higher heat transfer rate and pressure drop than a plain tube due to the higher turbulence near the tube wall. Akansu numerically investigated the effect of the space between porous rings on the heat transfer rate and friction factor characteristics and found that a smaller spacing gave higher heat transfer rate and friction factor. Effect of the elliptic ring (angled/transverse ribs) on the flowstructure and circumferential heat transfer distributionwas reported by Kiml et al. Eventually, the angled ribs offered greater heat transfer than the transverse ribs which was responsible by a development of the rib-induced secondary flow in a form of a pair of vortices, resulting in more efficient fluid transfer between core region and wall region. Recently, Kongkaitpaiboon et al. experimentally the effects of the diameterratio and pitch ratio of circular-ring turbulators on the thermal performance and found that the circular-ring with the smallest diameter and pitch ratios gave the highest thermal performance. will be greater, resulting in a higher overall heat exchanger volume.

The passive techniques have been usually preferred by many researchers since no additional external power is required as extended surfaces, rough surfaces and swirl flow devices. Coiled wireinsert is one of the passive heat transfer enhancement techniques, which is extensively used in various heat transfer applications such as, air conditioning and refrigeration systems, heat recovery processes, food and dairy processes, chemical process plants. Coiled wire, twisted tape or other swirl flow devices inserted into a flow provide swirling flow and periodic redevelopment of the boundary layer, increase the effective heat transfer area and the turbulence intensity. The swirl induced tangential flow velocity component causes improved fluid mixing between the tube core and the wall region nearby. On the other hand, the swirl induced heat transfer enhancement brings along inevitable shear stress and pressure loss in coiled wire or twisted tape inserted tube.

## 2. Literature Survey

Sarmaet. al [2] gave generalized correlations to predict friction factor and convective heat transfer coefficient in a tube fitted with twisted tapes for a wide range of Reynolds number and Prandtl number. The contribution to thermal performance of the conventional, short-width and center-cleared twisted tapes was studied numerically [2]. Configuration optimization of regularly spaced short-length twisted Eiamsa-ard et al. [12]performed experimental works on heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with twisted tape elements. They made their analysis for both continuously placed twisted tape and twisted tape placed with various free spaced in circular tube.

Experimental data were obtained for water and ehtylene glycol with snug-fit tape inserts of three different twist ratios, y=3.0, 4.5, and 6.0; the tape thickness in each case was 0.483 mm. In continuation of their research, an extended review of the application of twisted-tape inserts in tubular heat exchangers and their thermal-hydraulic performance was discussed by Manglik and Bergles[3].Naphon [15] made experiments by using conventional twisted tape inserts in horizontal double pipe. Ozceyhan and Siebel Gunes [8] conducted experiments with equilateral triangle cross sectioned coiled wire inserts and they found the maximum performance evaluation of about 1.38 as compared to the plain tube. They performed their experiments for various Reynolds numbers and the obtained experimental data were then compared with those previously reported in the literature.

In some studies, researchers focused the thermal effects of twisted tape inserts in modified tube instead of smooth tube, for example; Thianpong et al. examined heat transfer enhancement in a dimpled tube with a twisted tape swirl generator inserted. They also presented the empirical correlations based on the experimental results of their study for prediction the Nusselt number and friction factor for Reynolds number from 12,000 to 44,000.Although many theories has been devised over past few years, there exist certain limitations and technological gaps. In an attempt to overcome some of the limitation and explore possibility of alternate approach to calculate the temperature of serrated plates, a method has been proposed. The following were the aims and objectives of the research being conducted during the study;

1. To investigate the effect of various thickness width and pitches of copper and aluminium plate inserts on heat transfer rate.
2. Compare theoretical and experimental values of heat transfer coefficient, Nusselt number, friction factor for copper and aluminium.
3. To find out the optimum result of Nusselt number with minimum pressure drop for that insert.

## 3. Experimentation

**Fig. 1** Experimental set up for forced convection

## 4. Specifications Of Set-Up

1 Inner diameter of pipe = ($d_o$) = 0.025 m.
2 Outer diameter of pipe = ($d_i$) = 0.028 m.
3 Length of test section L = 0.5 m.
4 Capacity of blower = 1 HP.
5 Diameter of orifice(d) = 0.014m.
6 Range of Dimmerstat =0 to 5 amp. 0-480V ac.
7 Temperature indicator = 0 to 300 $^0$C.
8 PR-IN and PR-OUT = Pressure gauge at start and end of the test pipe. 0 to 2 kg/cm$^2$
9 Calibrated for chromelalumel thermocouple.
10 Voltmeter = 0-480 V; ammeter 0-5 amp. Nichrome wire heater wound in test pipe.

The literature that has been discussed above deals with active as well as passive techniques used for heat transfer augmentation, but maximum researchers have used passive technique as it doesn't require any external agency for heat enhancement, hence it is advantageous over the active techniques thus giving sample of room for experimental studies.

The slot has been made on copper and aluminium inserts at 3mm and 5mm plate. The plate is 3mm and 5mm thick and the slots are of 6mm and 8mm width. The amount of material removed will be 1.0mm and 2.5mm respectively. The arrangement has been made in such a way that when the air enters in heat exchanger due to geometrical arrangements on plate the air will move in wavy form. Waves will create a turbulence and as the turbulence will increase it increases the heat enhancement by increasing heat transfer coefficient. The material is in solid form although it will increase the pumping power but due to the geometrical arrangements on plate comparative heat enhancement will also be considerable.

## 5. Methodology

The heat flux applied to the test tube cause an increase in the outer surface temperature (Tow) of the test tube in axial direction. Therefore, the heat loss is calculated for each part of the test tube in which the thermocouples exist. The total heat loss is taken as the sum of these 28 parts. The heat loss $Q_{loss}$ is the heat transfers from the outer tube wall to the surrounding and calculated as follows:

$$Q_{loss} = F(T_{ow} - T_{\infty}) \quad (1)$$

$$\frac{1}{F} = \frac{1}{h_o A_{ins}} + \frac{1}{K_{ins} A_{12}} \quad (2)$$

$$Q_{Air} = Q_{conv} \quad (3)$$

$$Q_{Air} = mc_{p,air}(T_o - T_i) = \Delta VI - Q_{loss} \quad (4)$$

$$q = \frac{Q_{air}}{\pi D_o L} \quad (5)$$

$$h(x) = \frac{q}{T_{iw}(x) - T_b(x)} \quad (6)$$

$k_{ins}$*$A_{12}$In Eq. (2), ho indicates the heat transfer coefficient of the natural convection occurs between the outer surface of the insulated test tube and the surroundings.

$$Nu_{th} = 0.023 \times Re^{0.8} \times Pr^{0.4} \quad (7)$$

This equation is called Dittus-Boettier equation.

$$f_s = (0.790 \times ln(Re) - 1.64)^2 \quad (8)$$

This equation is used to find friction factor called as Petukhov equation for smooth surface. [2]

Where,
$f_s$= Friction factor for smooth tube .
Re= Reynolds number.

$$\frac{1}{\sqrt{fr}} = -0.86 \ln\{[\frac{\frac{\varepsilon}{D}}{3.7}] + \frac{2.51}{Re\sqrt{fr}}\} \quad (9)$$

This equation is used to find friction factor called as Nikuradse equation for rough surface, with inserts. [2]
The actual pressure drop is calculated with the help of pressure gauge at both the ends of test pipe and the friction factor is calculated from the formula given below:

$$\nabla P = \frac{fplv^2}{2D} \quad (10)$$

Where,
ΔP= pressure difference at both ends of test pipe.
L= length of test pipe.
D= Inner diameter of pipe.

The experimental Nusselt numbers are calculated for smooth as well as for rough surfaces are given below:

$$N_u = \frac{\frac{fr}{8}(Re-1000)Pr}{(1+[12.7(\frac{fr}{8})^{.5} Pr^{\frac{2}{3}}-1])} \quad (11)$$

This equation is called as Gnielieski equation to find theoretical and actual Nusselt number for smooth and rough pipe. [2]

Where,
$f_r$= friction factor for rough surface
Nu = Nusselt number
$P_r$ = Prandtl number

The overall enhancement efficiency is expressed as the ratio of the Nusselt number of an enhanced tube with conical ring insert to that of a smooth tube, at a constant pumping power is introduced by Webb [10].

$$PEC = \eta = \frac{\text{Nu with/Nu w/o}}{(\text{fwith|fw/o})^{1/3}} \quad (12)$$

The actual photographs of serrated plate inserts of various pitches and thickness are shown below from Fig. (a) to (d).

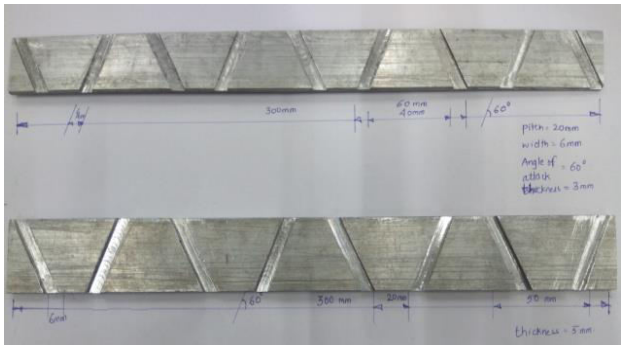


**Fig. (a)**Aluminium inserts of 6mm width slot for 3mm and 5mm thick plate.

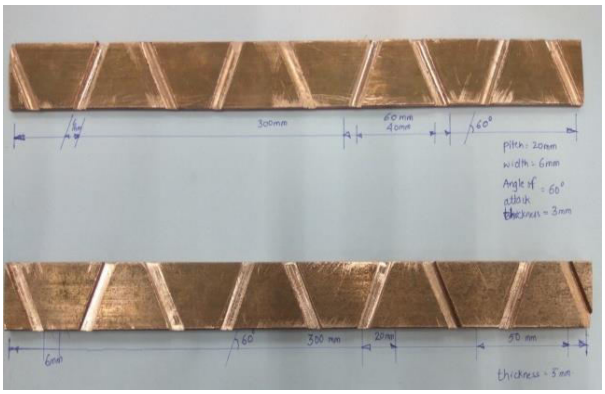


**Fig. (b)** Copper inserts of 6mm width slot for 3mm and 5mm thick plate.

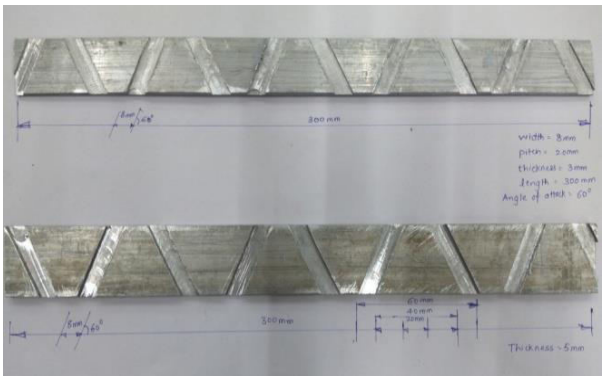


**Fig.(c)** Aluminium inserts of 8mm width slot for 3mm and 5mm thick plate.

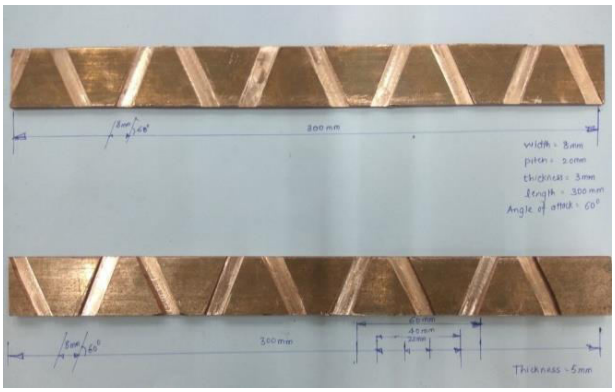
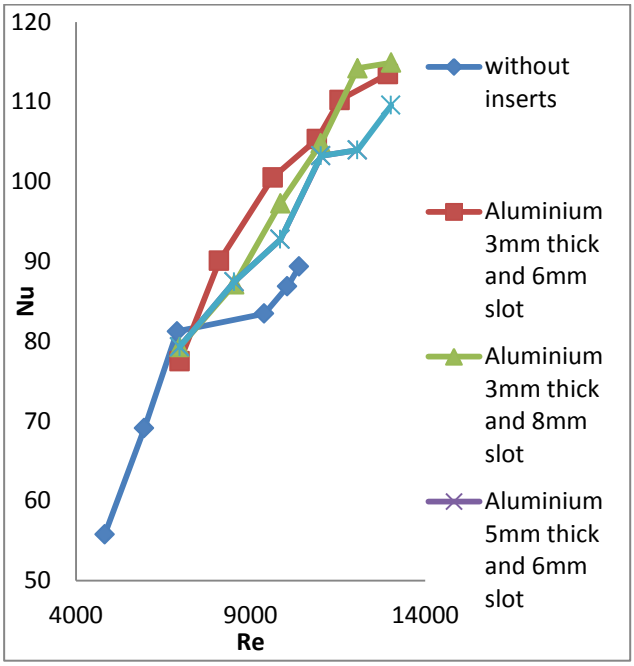


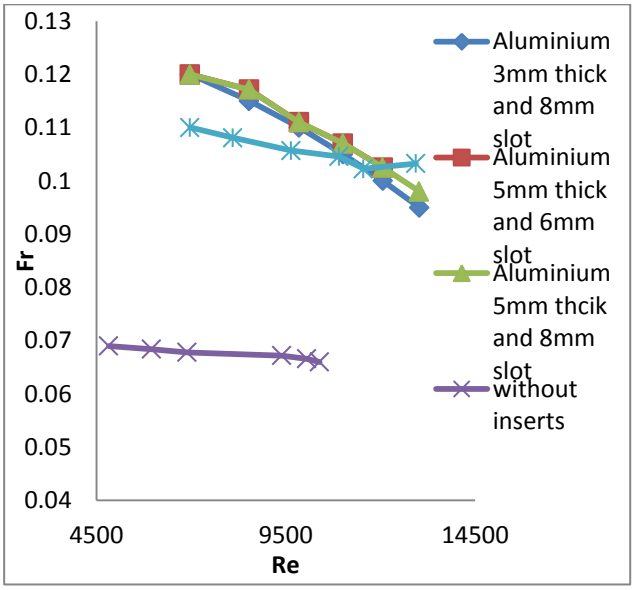
**Fig. (d)** Copper inserts of 8mm width slot for 3mm and 5mm thick plate.

## 6. Results and Discussions



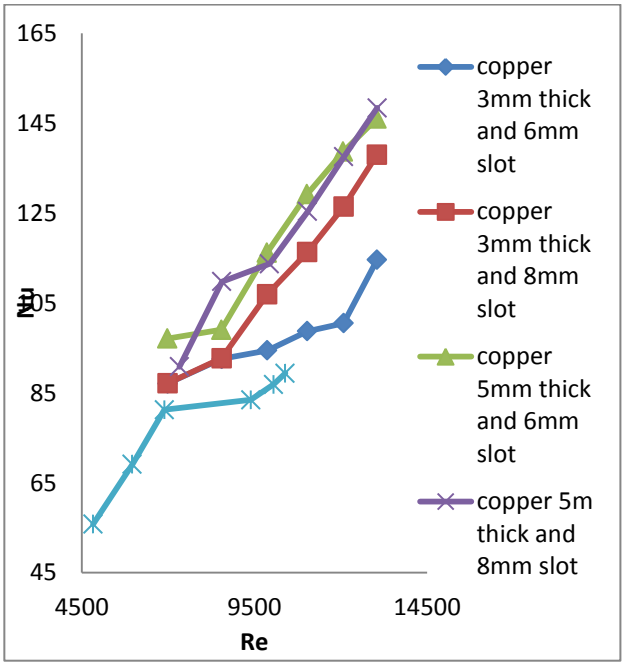

**Graph 1** Comparison Nu Vs Re for Al Inserts

From above graph 1, Nusselt number Vs Reynolds number it is observed that aluminium plate 3mm thick and 8mm slot has greater nusselt number in range of 79 to 115 approx from 7000Re to 13000Re hence it is found to be optimum.



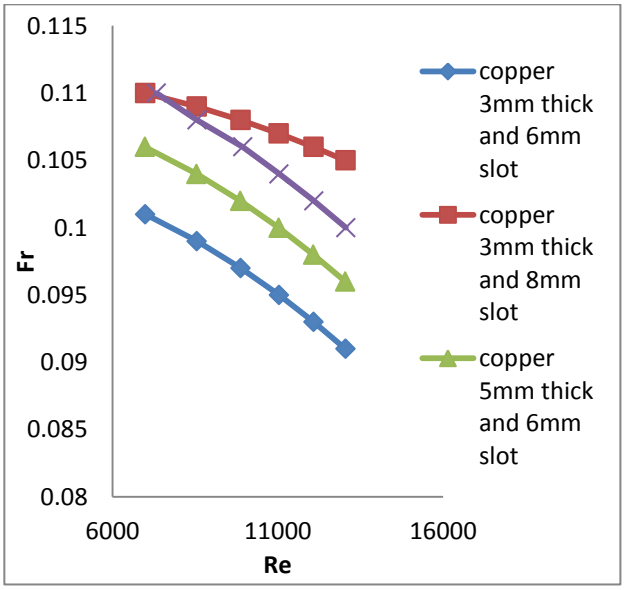

**Graph 2** Comparison Fr Vs Re for 'Al' inserts

From above graph 2, Friction factor Vs Reynolds number it is observed that aluminium plate 3mm thick and 8mm slot has lesser friction factor in the range of 0.12 to 0.095 with approx. 7000Re to 13000Re hence it found optimum.



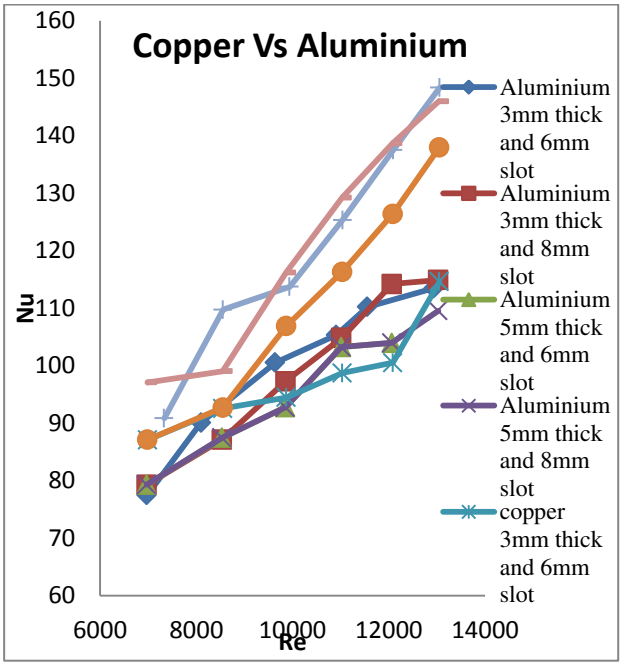

**Graph 3** Comparison of Nu Vs Re for Cu Inserts

From above graph 3, Nusselt number Vs Reynolds number it is been observed that plate of copper 5mm thick and 8mm slot has greater nusselt number in the range of 90 to 150 with approx 7000Re to 13000Re hence found optimum.



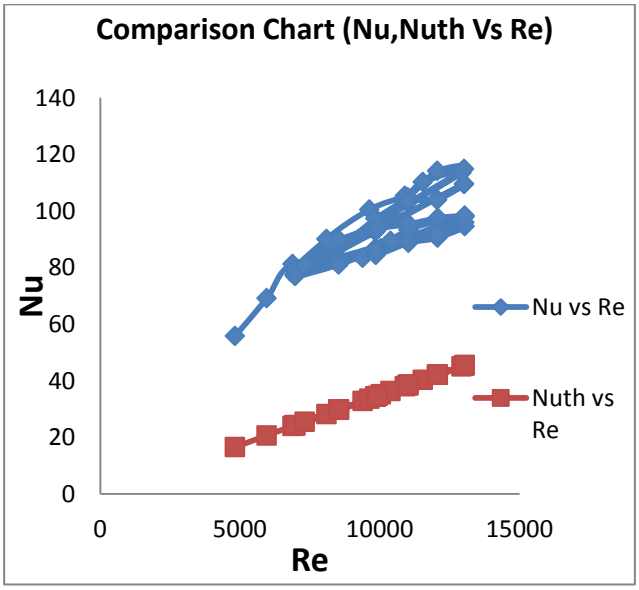

**Graph 4** Comparison of Fr Vs Re for 'Cu' Insert

From above graph 4, Friction factor Vs Reynolds number it is been observed that plate of copper 3mm thick and 6mm slot has lesser friction factor in the range of 0.10 to 0.091 with approx 7000Re to 13000Re hence found optimum.




**Graph 5** Comparison of Nu Vs Re for 'All' Inserts

From above graph 5, Nusselt number Vs Reynolds number it is been observed that plate of copper 5mm thick and 8mm slot has greater nusselt number in the range of 90 to 150 with approx 7000Re to 13000Re hence it is found to be optimum.




**Graph 6** Comparison of Nu, Nuth Vs Re for 'All' Inserts

From above graph 6, Nusselt number Vs Reynolds number it is been observed that greater nusselt number in case of experimental results by neglecting friction or the various minor losses as compared to the theoretical Nusselt number.

## 7. Future Scope

There is always scope for improvements in present work. So, serrated plates are not an exception. This may leads to better design of serrated plates with different thickness which will be helpful for serrated plates to be used as a commonly used different inserts. Other possible modifications are as listed below: We can reduce number of thickness to width distance to get more optimum value .Experiments are also possible with varying the thick of serrated plates. We can try some modification in the geometry of serrated plates with the help of CFD analysis. We can try the same geometry with different materials and compare the results with the existing material.

## 8. Conclusion

The results obtained from the experiments which are discussed above are concluded below to find an optimum insert as compared to the other inserts to be used for this experimental set up.

We observe that with an increase in the Reynolds number (Re) ranging from 6700 to 13000, the heat transfer coefficients increases for Serrated Plates whereas the friction factor decreases.

The 6mm slot and 5mm plate thickness copper shows higher Nusselt number of 60% to 76% and lesser friction factor by 39% as compared to without inserts.

Comparison of Nusselt number experimental and Nusselt number theoretical with Reynolds number shows that, greater nusselt number in case of experimental results by neglecting friction or the various minor losses as compared to the theoretical Nusselt number.

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