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Experimental investigation of forsed convection augmentation with insert in tube

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ABSTRACT

An experimental investigation was carried for measuring tube side heat transfer coefficient, friction factor of water for turbulent flow in a circular tube fitted with insert. A circular tube of 25 mm internal diameter and 32 mm of outer diameter and 500 mm test length was used. A uniform heat flux boundary condition was created by band type heater fitted around test section. Outer surface temperature of the tube was measured by 8 different points of the test section by thermocouples. Two thermometers were used for measuring the bulk temperature. The experiments were conducted by insertion of inserts for flow blockage in the range of 20, 30, 40 and 50% with different heat flux condition. During the test water at 30°C was passed through the test tube which was controlled under uniform wall heat flux condition. The Reynolds number was varied from 3000 to 8000 with heat flux variation 4.936 kW/m^2 to 19.723 kW/m^2. According to experimental results heat transfer rate in the tube fitted with insert are augment compared to that in plain tube. The result also reveals the higher flow blockage offer highest heat transfer rate with the largest pressure loss.

Keywords— **Heat transfer enhancement, Passive technique, Insert, Flow blockage**

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I. INTRODUCTION

Increase in heat exchanger's performance can lead to more economical design of heat exchanger which can help to make energy, material & cost savings related to a heat exchange process.

The need to increase the thermal performance of heat exchangers, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as heat transfer augmentation. These techniques are also referred as heat transfer enhancement or intensification. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a tube/ heat exchanger.

Existing enhancement technique can be broadly classified into three different categories

A. Active method:

This method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. Some examples of active methods include mechanicals aids, surface vibration, fluid vibration, electro-static fields. It finds limited application because of the need of external power in many practical applications.

B. Passive method:

Passive techniques do not require any direct input of external power. They generally use geometrical or surface modifications to the flow channel by incorporating inserts or additional devices. Except for the case of extended surfaces, they promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour.

C. Compound method:

When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement.

In general the performance of turbulator strongly depends upon their geometries. Bodius Salam [1] experimentally studied Heat transfer enhancement in a tube using rectangular-cut twisted tape insert. Eiasma ard [2] reported the effect of twist ratio and space ratio on heat transfer and pressure drop. Ramin k [3] reported the effect of stationary insert on heat transfer coefficient and friction factor.

S. Naga Sarda [6] experimental report the effect of varying width insert on thermal performance of heat exchanger tube. S K saha, Hyder eren [5 to 7] reports the effect of compound technique like combination of twisted tape and wire coil, wire coil and spring, dimple tube and twisted tape etc. were performing better than single passive technique.

The present work proposes circular ring as a turbulator for heat transfer enhancement device. The turbulators of various flow blockages (20%, 30%, 40% and 50%) were arbitral selected for test.

Nomenclature

A	heat transfer surface area, m−2
Cp	specific heat capacity of water, kJ kg−1 K−1
d	inner diameter of insert, m
D	inner diameter of pipe, m
DR	diameter ratio=d/D
f	friction factor
f_t	Theoretical friction factor
f_e	Experimental friction factor
fo	Friction factor ratio
h	mean heat transfer coefficient, Wm−2 K−1
I	current, A
K	thermal conductivity of water, Wm−1 K−1
L	length of the test tube, m
m	mass flow rate, kg s−1
Nu	Nusselt number
Nu_t	Theoretical Nusselt number
Nu_e	Experimental Nusselt number
Nuo	Nusselt number ratio
Q	heat transfer rate, W
p	pitch length of insert, m
ΔP	pressure drop, Pa
PR	pitch ratio=p/D
Pr	Prandtl number
Re	Reynolds number
T	temperature, K
t	thickness of the test tube, m
μ	Viscocity, N s m-2
u_m	mean axial velocity, m s−1
V	voltage, V
$\dot{v}$	volumetric flow rate, m3 s−1
Greek symbols	
ν	kinematic viscosity, Ns m−2
η	thermal performance factor
ρ	fluid density, kg m−3
Subscripts	
b	bulk
i	inlet
s	surface
o	outlet
p	plain tube
t	turbulator
w	wall

II. GEOMETRY

The details of the circular ring for flow blockage are as shown in fig. 1. The circular ring inserts are made of aluminium with 6mm thickness. The inner diameter of pipe and the outer diameter of inserts were fixed at 25 mm while inner diameter of inserts were varied 23.72 mm, 22.36 mm, 20.92 mm, 19.37 mm and 17.68 mm for 20%, 30%, 40%, 50% flow blockage. In the experiment circular ring inserts were placed in the test tube with pitch equal to 50 mm. The test section is made of mild steel with 25 mm inner diameter, 500 mm in length and 3.5 mm thickness.

III.EXPERIMENTAL SETUP

The schematic diagram of experimental setup is as shown in fig. 2. The experimental setup consists of storage tank, pump, bypass valve, rota-meter, band type heater, digital manometer and radiator. A 0.5 HP pump connected to collecting tank of 0.05m3 capacity is used to circulate the working fluid through the test section. The outer diameter of test section was enclosed with band type heater of 1000W rating. The heater is enclosed with glass-wool insulation to minimize convective heat loss to the surroundings. The heater was connected to 250 V, 4 A main. Ten K-type thermocouples are fixed at different locations, eight on the surface of the tube wall and the other two are located at the inlet and outlet to measure the temperature of the water. A rota-meter of 10 L/min capacity was provided to measure the water flow rate. A digital manometer was used to measure the pressure drop across the tube. A radiator was located in between test section and storage tank.

Initially, water was pumped from the tank to test section through rota-meter. The flow rate was varied for different data and kept constant during experiment. The flow rate was varied from 3 L/min to 6 L/min. Take readings for different heat flux condition to test section with inserts and without inserts

IV. DATA REDUCTION

$$aQ = VI \quad \text{(Energy supplied)}$$

Heat transfer rate by the heater to water was calculated by measuring heat added to the water. Heat added to water was calculated by,

$$Q = mC_p(T_o - T_i)$$

Heat transfer coefficient was calculated from,

$$h = \frac{q}{T_s - T_b}$$

And heat flux was obtained from,

$$q = \frac{Q}{A}$$

Where,

$$A = \pi DL$$

The bulk temperature was obtained from the average of water inlet and outlet temperatures,

$$T_b = \frac{T_i + T_o}{2}$$

Tube outer surface temperature was calculated from the average of eight local tube outer surface temperatures,

$$T_s = \sum_{i=1}^{5} \frac{T_{wo}}{5}$$

The standard correlations include Dittus-Boelter and Blasius for the fully developed turbulent flow in circular tube.

Nusselt number correlation:

Dittus-Boelter correlation: $Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.3}$

Friction factor correlation:

Blasius correlation:

$$f = 0.316 \mathrm{Re}^{-0.25}$$

$$\mathrm{Re} = \frac{\rho u_m D}{\mu}, \ \mathrm{Pr} = \frac{\mu C_p}{K}$$

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

Friction factor, f can be calculated from,

$$f = \frac{\Delta P}{\left(L/D\right)\left(\rho u^2{}_m/2\right)}$$

Δp is the pressure drop across tapping. All the fluid properties were evaluated at bulk temperature.

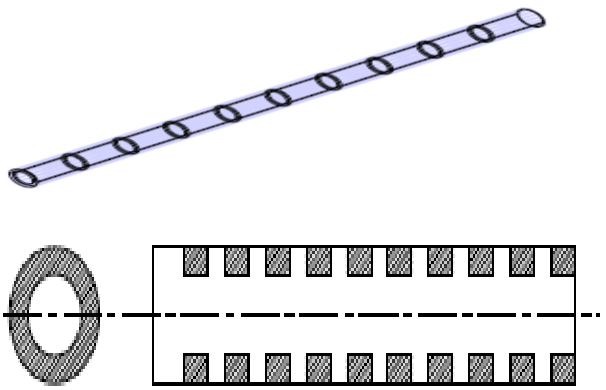

Fig.1. Test section with circular-ring for blockage

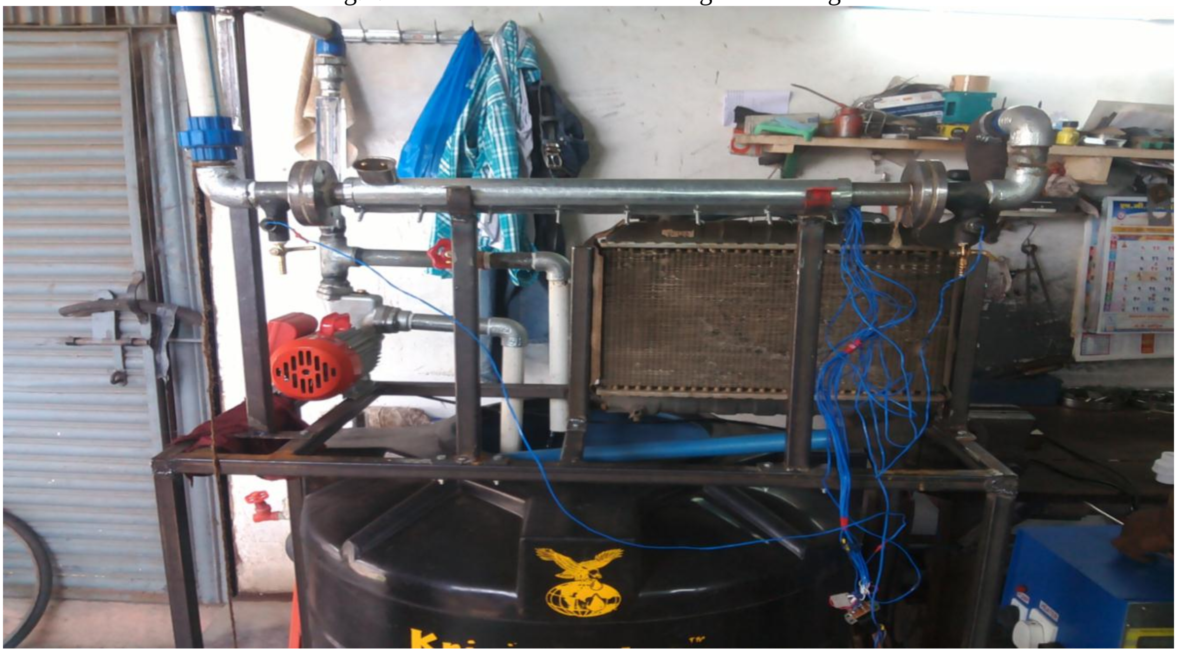

Fig.2. Schematic diagram of experimental setup

V.RESULTS

Effects of tube fitted with circular ring provided different flow blockages like 20%, 30%, 40%, and 50% on the heat transfer enhancement and pressure drop are reported in the present paper. Heat transfer and friction data for the plain tube are collected first. These data are taken to check the validity of the setup and measurement technique over the range of Reynolds number 3000 to 7000. Fig.3. Shows the comparison between experimental Nusselt number for plain tube and theoretical Nusselt number which is calculated by Dittus-Boelter correlation:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3}$$

Fig.4. Shows the comparison between friction factor of plain tube and theoretical friction factor which is calculated by Blasius correlation:

$$f = 0.316\,\mathrm{Re}^{-0.25}$$

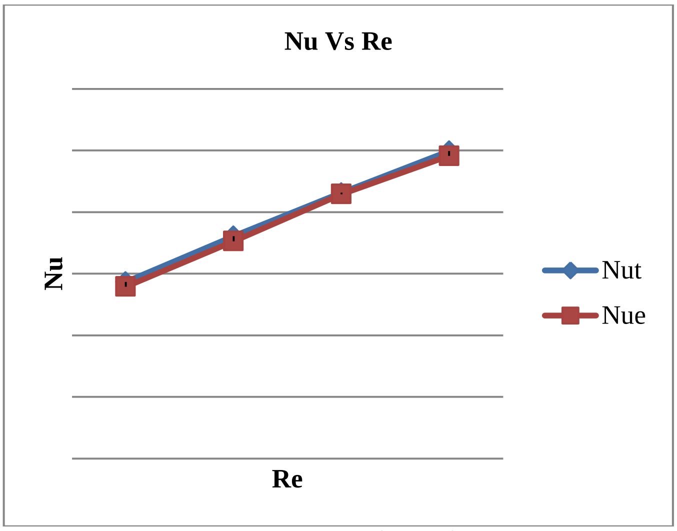

Fig.3. Nusselt number data verification for plain tube

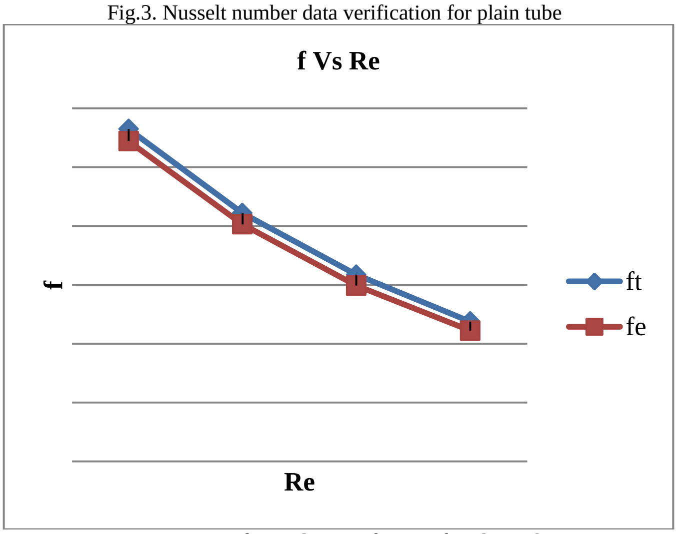

Fig.4. Friction factor data verification for plain tube

A. Effect of flow blockage on heat transfer:

Introducing circular ring is in tube with different blockage like 20%, 30%, 40% and 50%. These circular rings are effect on heat transfer enhancement and Nusselt number. Fig.5. Shows the comparison between Reynolds number and Nusselt number, which is different for plain tube and different blockage. Fig.6. shows comparison between heat transfer coefficient and Reynolds number.

At a given Reynolds number, Nusselt number and heat transfer coefficient in the tube equipped with circular ring is higher than that of plain tube. The experimental results reveal that the circular ring provides higher blockage generate stronger turbulence intensity and higher heat transfer rate than smaller blockage. Nusselt number ratio is the ratio of theoretical to experimental Nusselt number. Fig.7. Shows Nusselt number ratio slightly decreases with rise of Reynolds number for all circular rings

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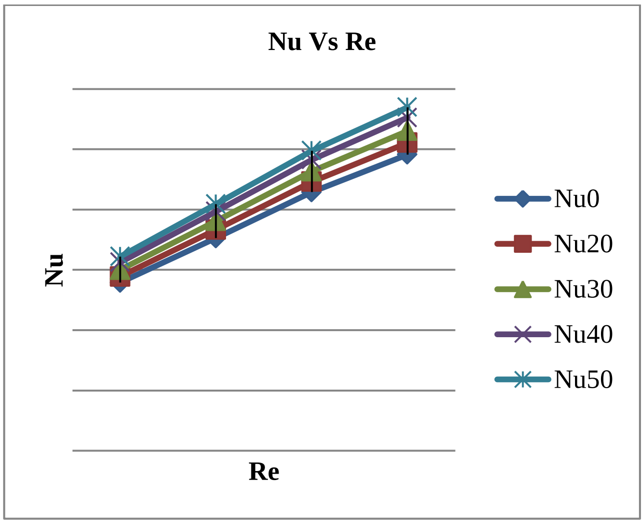

Fig.5. Nusselt number versus Reynolds number

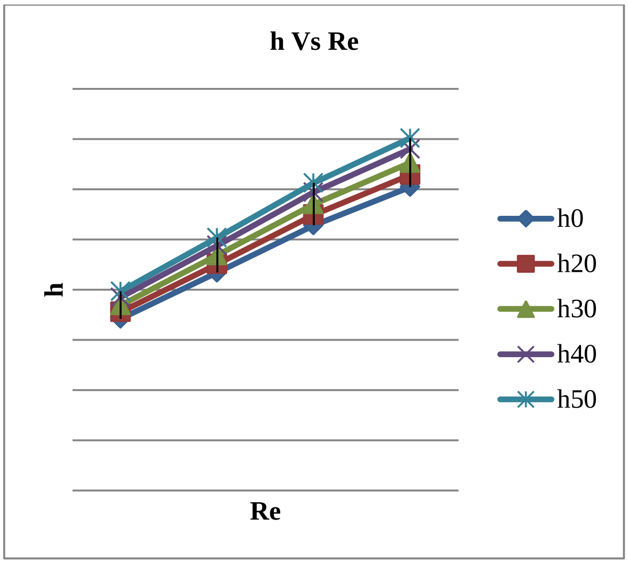

Fig.6. Heat transfer coefficient versus Reynolds number

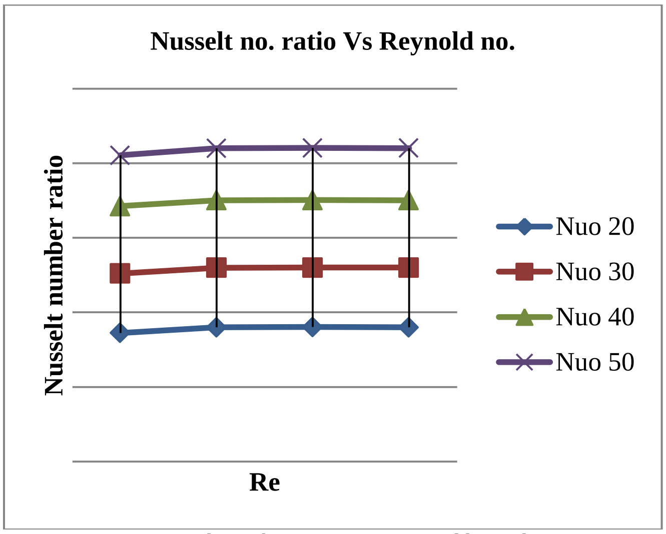

Fig.7. Nusselt number ratio versus Reynolds number

B. Effect of flow blockage on friction factor

Effect of the circular ring provided different blockage (20%, 30%, 40% and 50%) on friction factor is presented. Fig.8. shows the comparison between Reynolds number and friction factor. Friction factor is different for plain tube and different blockages. Fig.9. shows the comparison between Reynolds number and pressure drop. Pressure drop is different for plain tube and different blockage. At a given Reynolds number, friction factor and pressure drop in the tube equipped with circular ring was higher than that of plain tube. The experimental results reveal that the circular ring provides higher blockage generate higher pressure drop than smaller blockage. Friction factor ratio is the ratio of theoretical to experimental friction factor. Fig.10 shows Friction factor ratio slightly increases with rise of Reynolds number for all circular rings.

C .Thermal performance factor

To check the potential of insert for real application, it is required to account enhanced heat transfer and friction loss caused by insert. Enhanced heat transfer ratio is defined as ratio of Nusselt number in the tube with insert to that in the tube without the device. Friction factor ratio is defined as ratio of friction factor in the tube with insert to that in the tube without the device. For thermal performance factor both ratio are required with constant pumping power. Thermal performance factor is defined as

$$\frac{\left(Nu_t / Nu_P\right)}{\left(f_t / f_P\right)^{1/3}}$$

Fig.11 shows the graph between thermal performance factor and Reynolds number. Thermal performance factor are below than unity especially at high Reynolds number regime. Friction factor ratio increases with Reynolds number and Nusselt number ratio decreases with Reynolds number. Hence friction factor ratio dominant over Nusselt number ratio so thermal performance factor less than unity.

But at low Reynolds regime, Thermal performance factor above than unity are found at 50% flow blockage indicating the potential of these circular ring as energy saving device

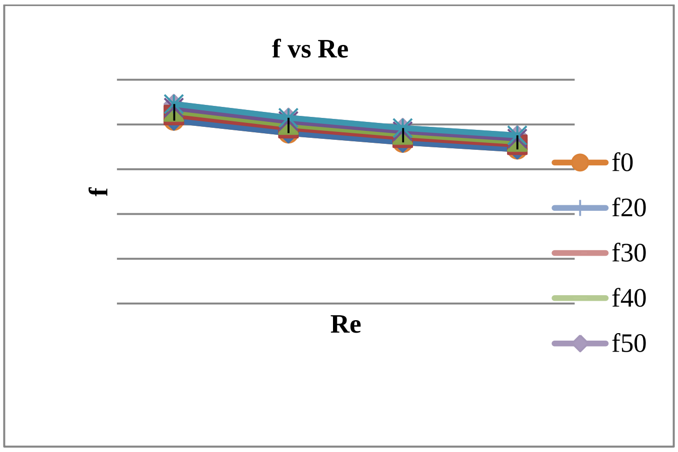

Fig.8. Friction factor versus Reynolds number

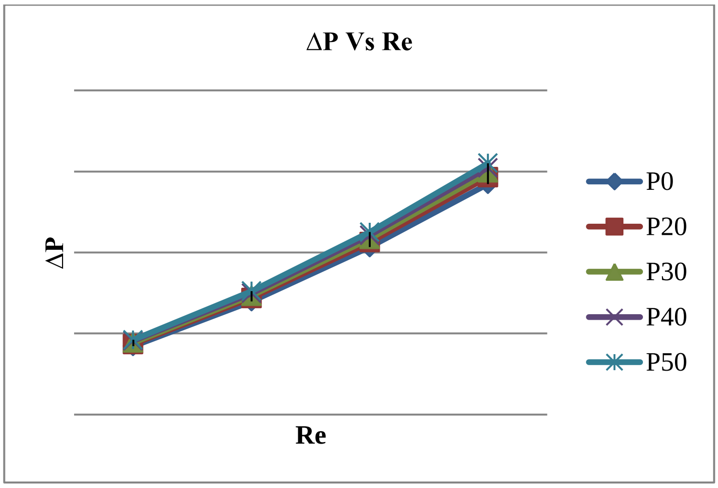

Fig.9. Pressure drop versus Reynolds number

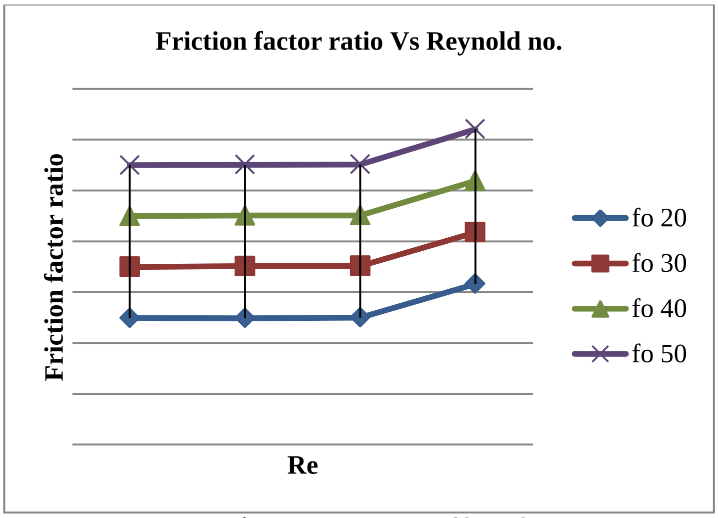

Fig.10. Friction factor ratio versus Reynolds number

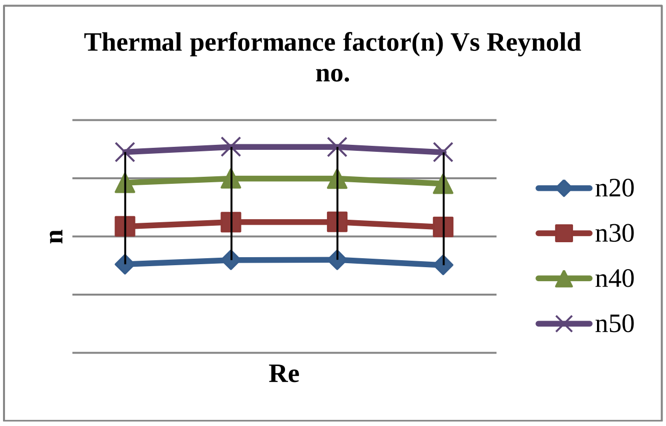

Fig.11. Thermal performance factor versus Reynolds number

VI.CONCLUSION

Heat transfer enhancement in tube fitted with insert is reported in this paper. By using this circular ring insert for flow blockage which affect on heat transfer enhancement and friction factor is reported in this paper. The following conclusion are done by using these insert are

1 Experimental setup is valid by comparing the result obtained by correlation and experimental result.

2 At the given Reynolds number, Nusselt number (heat transfer rate) in the tube equipped with circular ring is higher than that in plain tube.

3 As the Reynolds number is increases Nusselt number increases and friction factor decreases.

4 As the flow blockage increases from 20 to 50% the heat transfer rate and friction factor also increases.

5 As the Reynolds number increases, friction factor ratio slightly increases and Nusselt number ratio slightly decreases.

6 The maximum thermal performance factor is obtained by the use of circular ring with 50% blockage.

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