Experimental Study on Effect of Passive Devises on Heat Transfer and Pressure Drop in a Round Tube Using Circular Ring Inserts, Fitted With Screen Mesh

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Abstract
This paper discusses the effect of the passive devices i.e. Circular-Ring Insert (CRT) fitted with screen mesh on heat transfer and pressure drop in round tube. The experimental set up is developed and experiments are performed by insertion of CRTs with various geometries, including diameter ratios of DR=di/do = 0.6724, 0.8275, 0.9310 and different pitch ratios of PR = p/Do =6.25, 8.33 and 12.5. During the test atmospheric condition air is passed through the test tube which is controlled under uniform wall heat flux condition. The Reynolds number is varied from 8000 to 15,000. The mesh insert is fitted on circular ring insert. Inserts are manufactured of different diameter ratios e.g. 0.6724, 0.8275, 0.9310. By changing pitch ratios e.g. 6.25, 8.33 and 12.5 experiments are performed. For insert having DR =0.6724 and PR=6.25, highest heat transfer rates are achieved along with largest pressure loss. Insert having DR=0.8275 and PR=12.5 does not have good potential for heat transfer. Insert of DR= 0.9310 and PR=6.25, has thermal performance factor above 1 while insert of DR=0.9310, PR=8.33 it has thermal performance factor approximately equal to 1 which shows that the insert of DR=0.9310 have good potential to enhance heat transfer.

Keywords
Heat Transfer Enhancement, Screen Mesh Insert, Pressure Drop

I. Introduction

To reduce the size and cost of heat exchangers, heat transfer enhancement technology has been developed. There are 3 enhancement techniques namely passive technique, active technique and compound Technique. Passive techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. Active techniques require external power for their operation. Compound technique include more than one enhancement techniques in combination is used. Tube insert devices are the examples of passive technique. Tube insert devices include twisted tape inserts; wire coil inserts, mesh inserts, ring insert etc. The ring insert reduces the hydraulic diameter and acts as an extended surface. Mesh insert are made of a matrix of thin wire filaments. The selection of these tube inserts depends on two factors: performance and cost. In fact, twisted tape inserts and wire coil inserts are more widely applied than the others. This is probably because the extended surface insert suffers from relatively high cost.

Hosni Mulaweh [1] used orifices or other temporary restrictions to induce turbulence. As opening is much smaller than the rest of the annulus; liquid velocity increased which was associated with rapid expansion which resulted into transition from laminar to turbulent flow.

S.Naga, A.V.Sita et.al [2] performed experiments that proved augmentation of heat transfer by using screen mesh insert with porosity between 99.73% to 99.98%. Experiments were carried out for different Reynolds numbers (7000-14000). The improvement in average Nusselt number was about 2.15 times, while pressure drop increased by 1.23 times compared to that of plain tube.

P. Promvonge, S. Eiamsa-ard [3] experimentally investigated the effect of circular ring turbulator (CRT) on heat transfer and pressure loss in a round tube. Air used as working fluid. The heat transfer enhancement was obtained, which was 57% to 195% more as compared to plain tube without passive insert.

Veysel Ozceyhan, Sibel Gunes, et.al [4] numerically investigated the effect on rate of heat transfer by circular ring separated from wall. The experiments were performed by attaching the ring inserts separated from the tube wall. By attaching the inserts near the wall helped in preventing the formation of contamination. The circular cross sectional rings were preferred.

In present work I am going to use two different passive devices i.e. circular ring insert and mesh insert in combination. In present study mesh insert is fitted on circular ring insert. Inserts are manufactured of different diameter ratios like 0.6724, 0.8275, and 0.9310. By changing pitch ratios values as 6.25, 8.33 and 12.5 experiments are performed.

II. Experimental Set Up and Procedure

A. Experimental Set up

The Set up required for present work is shown in fig. 1.

Test Section is made up of Cu material. Cu is selected as it has high thermal conductivity value. The test section is made of copper tube with 63 mm inner diameter, 1500 mm in length (L) and 2 mm in thickness (t). Heater coil is wound around test section to give constant heat flux. Insulation is provided in order to reduce heat loss. The details of the insert i.e. circular-ring with wire mesh are demonstrated in fig. 2. The circular-ring turbulators are made of aluminum with 5 mm thickness. The outer diameter of the turbulators (D_o) is fixed at 58 mm while inner diameter is varied i.e. 0.6724 Di, 0.8275 D_i, 0.9310 Di. Mesh is inserted in circular ring as secondary insert.

To measure temperature along test section 15 K type thermocouples are placed uniformly along circumference of Cu tube. As length of tube is 1500 mm, each thermocouple is placed at 100 mm distance. To measure temperature of inlet air and outgoing air RTD's are used. Thickness of Cu tube is 2 mm, so heat transfer through wall of tube neglected. Temperature indicating unit is used to indicate value of temperature at various locations. Range of temperature indicating unit is from -10^0 to 600^0. To measure flow rate orifice flow meter is used. The diameter of orifice is 2.85cm. To measure pressure difference, two U tube manometers are used. Water is used as manometric fluid.

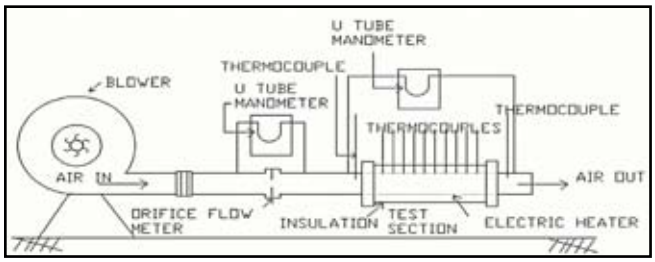

Figure 1: General layout of experimental test set up.

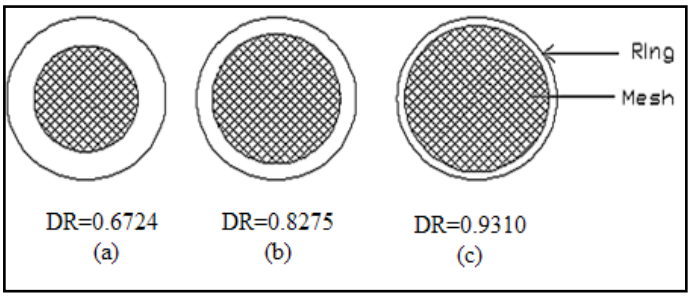

Fig. 2 (a, b, c): Various Inserts used in Experimentation

B. Experimental Description

The inlet bulk air at 32°C from blower is directed through the orifice meter and passed to the heat transfer test section. The air flow rate is measured by an orifice meter, while pressure drop is measured with U-tube manometers. The air flow rates from the blower is adjusted so that Reynolds number of the bulk air varied from 8500 to 15 000. During the experiments, the bulk air is heated by an adjustable electrical heater wrapping along the test section which having 2 kW capacity. Both the inlet and outlet temperatures of the bulk air from the tube are measured by K-type thermocouple at 16 stations. For each test run, necessary record of temperature, volumetric flow rate and pressure drop of the bulk air at steady state conditions in which the inlet air temperature is 32°C are maintained

The following test conditions are considered while performing above experimentation which are given in Table 1 while accuracy table for instruments used is given in Table 2.

Table 1: Test Conditions

Sr. No	Test conditions
1	Without any inserts, varying mass flow rate by keeping input power constant.
2	Put insert of DR=0.6724 and PR =6.25 in tube, Change mass flow rates.
3	Repeat test condition 2, for different PR= 8.33, 12.5
4	Change insert with DR=0.8274, 0.9310, for different PR perform experimental work with changing mass flow.

Table 2: Accuracy Table

Sr.No.	INSTRUMENT	UNIT	ACCURACY
1	Ammeter	A	-
2	Voltmeter	V	±0.1 V
3	Temperature Indicator	0C	± 1 0c
4	U tube manometer	mm of water	± 1 mm

III. Formulae and Standard Equations

A. Heat Transfer Coefficient (h)

Under steady state conditions heat transfer rate of hot air (Q_{air}) is same as that of heat transfer from wall by convection ($Q_{convection}$)

$$Q_{air}=Q_{convection} \quad (1)$$

Now Q air is calculated as

$$Q_{air} = \dot{m}c_{pa}\,(T_o - T_i) \quad (2)$$

Also Q convection is calculated as follow

$$Q\ convection = h.A.\left(\bar{T}_w - T_b\right) \quad (3)$$

Where $\bar{T}_w$ is average wall temperature calculated from locations and it is given by

$$\bar{T}_w = \sum(T_w)/15 \quad (4)$$

While T_b is average of temperature of air i.e.

$$T_b = (T_o + T_i)/2 \quad (5)$$

The average heat transfer coefficient and Nusselt number (Nu) is calculated as follow

The average heat transfer coefficient (h) is found as

$$h = \dot{m}c_{pa}\,(T_o - T_i)/A.\left(\bar{T}_w - T_b\right) \quad (6)$$

$$Nu = hD/k \quad (7)$$

Reynolds number (Re) is calculated as

$$Re = UD/\vartheta \quad (8)$$

B. Friction Factor (f)

As the air is flowing through the annular passage of the test tube and the wire mesh, thereby it suffers some friction.

This value of friction factor is obtained using the following equation [3].

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho\,\frac{U^2}{2}\right)} \quad (9)$$

C. Thermal Performance Factor (η)

The value of Thermal performance factor is obtained using the following equation [3].

$$\eta = \frac{\left(\frac{Nu_t}{Nu_p}\right)}{\left(\frac{f_t}{f_p}\right)^{1/3}} \quad (10)$$

IV. Results and Discussions

Fig. 3 and Fig. 4 represent confirmation of present experimental set up with standard equations. The natures of graphs are as follow.

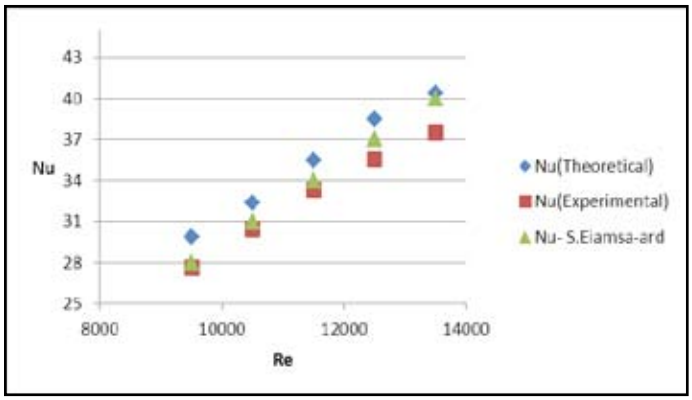

Fig. 3: Validation of Nusselt Number for Plain Tube Without Inserts

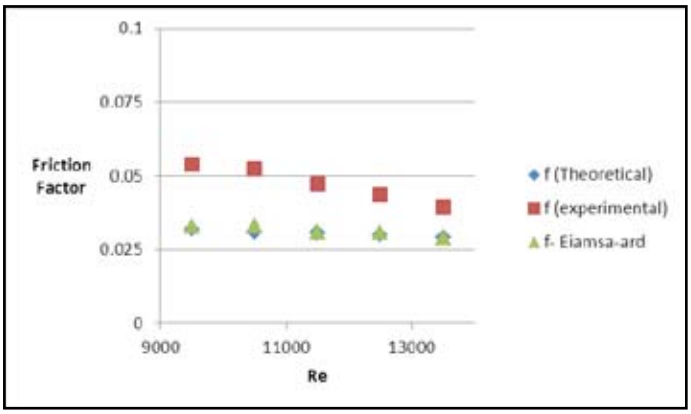

Fig. 4: Validation of Friction Factor for Plain Tube Without Inserts

Fig. 5 is plotted which give variation of Nusselt number with Reynolds number (Re) for different DR and PR inserts having following nature.

Fig. 6 is plotted which give variation of Friction factor with Reynolds number (Re) for different DR and PR inserts having following nature.

Fig. 7 is plotted which give variation of Thermal performance factor with Reynolds number (Re) for different DR and PR inserts having following nature.

To evaluate the potential for real application of the heat transfer enhancement device, both enhanced heat transfer and friction loss caused by the device must be considered.

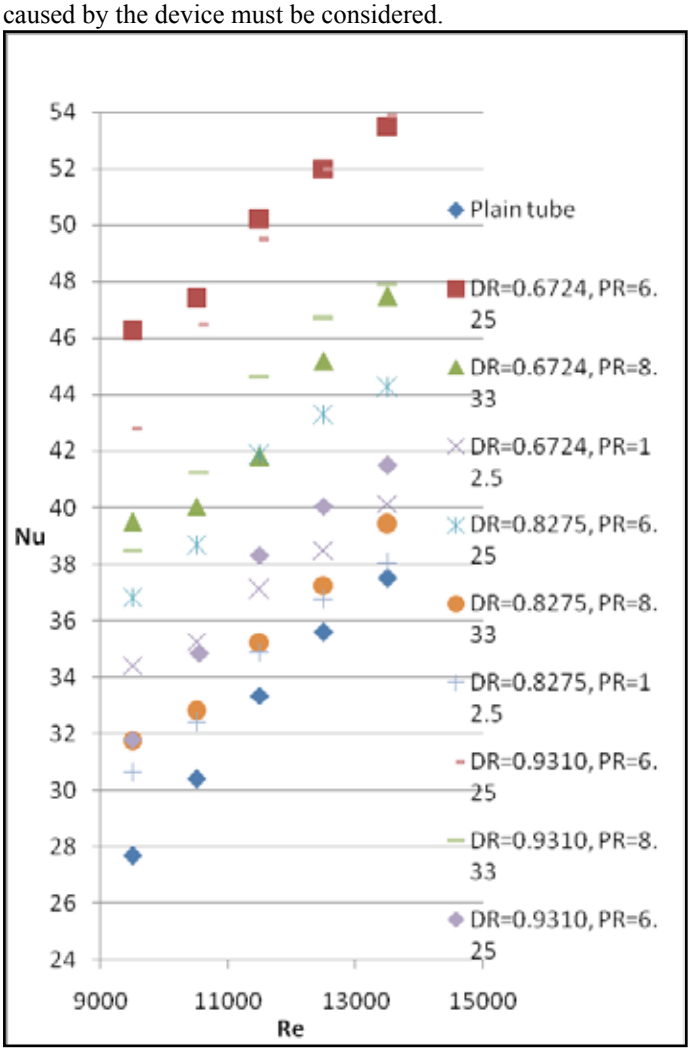

Fig. 5: Variation of Nusselt Number with Reynolds Number for Plain Tube Without Inserts and Tube With Inserts

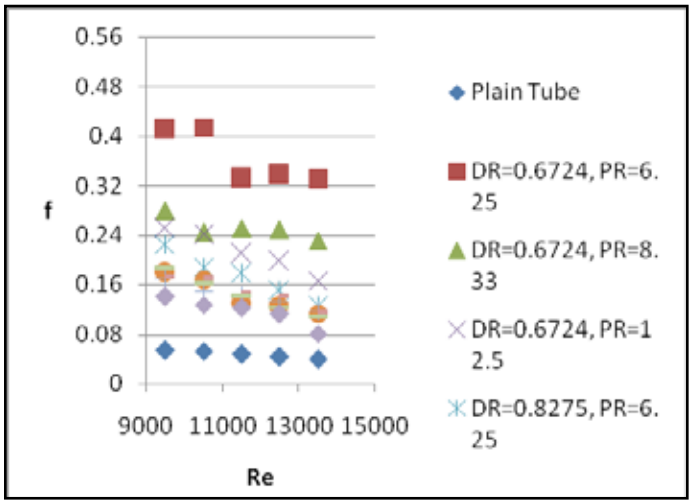

Fig. 6: Variation of Friction Factor With Reynolds Number for Plain Tube Without Inserts and Tube With Inserts

Generally, enhanced heat transfer is defined as ratio of Nusselt number in the tube with enhancing device to that in the tube without the device (Nu_t/Nu_p). Similarly, enhanced friction factor ratio is defined as ratio of friction factor in the tube with enhancing device to that in the tube without the device (f_t/f_p).

Both mentioned ratios are applied for the thermal performance criteria under the constraint of a constant pumping power ($\dot{v}\Delta P_p=\dot{v}\Delta P_t$), yields the thermal performance factor as $(Nu_t/Nu_p)/(f_t/f_p)^{1/3}$.

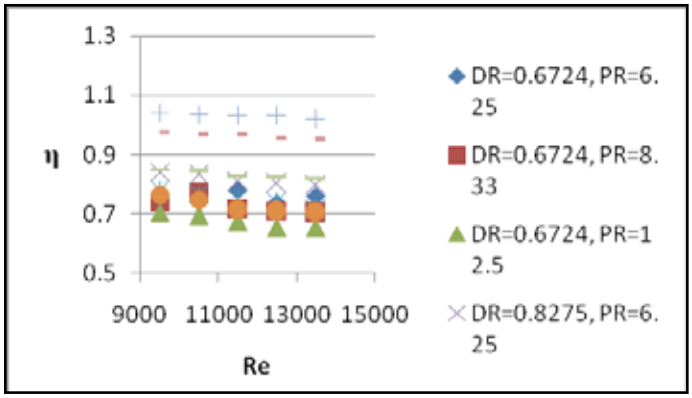

Fig. 7: Variation of Thermal Performance Factor With Reynolds Number for Plain Tube Without Inserts and Tube With Inserts

VII. Conclusion

1. Heat transfer rate increases at lower pitch ratios i.e. 6.25, 8.33.
2. Validation of Nu and friction factor without inserts for plain tube shows experimental results are in close match with the previous investigators results for heat transfer in round tube under forced convection.
3. For DR=0.9310 Nu values increased more than for DR=0.8275, this is because turbulence generated by screen mesh is more in case of inserts having DR=0.9310.
4. For all cases, Nusselt number increases and friction factor decreases with increasing Reynolds number. The highest Nusselt number and friction factor is obtained in the case of PR = 6.25
5. The Nusselt number and friction factor increase with decreasing ring spacing (PR). As expected inserts of DR=0.6724, PR=6.25 introduces more pressure drop than the other types.

ISSN : 2249-5762 (Online) | ISSN : 2249-5770 (Print)

6. The maximum thermal performance factor 1.04 obtained for insert having DR=0.9310, PR=6.25.
7. Heat transfer enhancement around 90% to 130% as compared to plain tube observed. For insert of DR=0.6724, heat transfer enhancement is 130%, for insert of DR=0.9310, heat transfer enhancement is 110% while for insert of DR =0.8275, it is only 90%.

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