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HEAT TRANSFER ENHANCEMENT BY USING RECTANGULAR INSERTS AT DIFFERENT ORIENTATION

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Abstract--*A empirical investigation on heat transfer enhancement has been conducted by using rectangular inserts in turbulent regimes (5000<Re<26000) in order to boost the performance of heat exchanging devices by cutting down the material cost and downsizing surface area for heat transfer. The effect on heat transfer rate, friction factor and heat transfer coefficient in a circular tube has been investigated as 4 rectangular inserts was implanted at different orientations and equal spacing. A copper tube of 26.6 mm internal diameter and 30 mm outer diameter and 900 mm test length was used. Uniform heat flux was applied to the external surface of the tube by wrapping Nichrome wire around the test section and fiber glass over the wire for insulation and water was used as the working fluid. Peripheral surface temperatures of the tube were measured at 5 singular points of the test section by K-type thermocouples. Rectangular boxes of 2 mm thickness were fitted perpendicularly at flow direction at different angular orientations approximating at 0º, 45º, 90º and at equal spacing for 4 inserts. The purpose of using inserts was to scatter the fluid particles which increase heat transfer. But along with the increase in heat transfer rate, pressure drop also increases. This increase in pressure drop increases the pumping cost. Thus, this enhancement project was analyzed in order to find the optimum benefits between increased heat transfer coefficient and higher cost involved due to increased frictional losses. The experimental results conclude that the best outcome was achieved when 4 inserts were used at 45º orientation, where transfer coefficient (h) increased up to 172%, heat transfer rate by 122%, Nusselt number (Nu) up to 29% at the cost of an escalation in friction factor up to 163% compared to the smooth tube which is the nominal correlating to the other statistics.*

Keywords: Heat transfer enhancement, Rectangular insert, Angular orientation, Turbulent flow

1.INTRODUCTION

The study of improvement of heat transfer rate is referred to as heat transfer augmentation, enhancement and intensification. Better performance and/or reduced the size of heat transfer devices can be achieved by heat transfer enhancement techniques. In practice, these techniques can be divided into two groups: active and passive techniques. Passive techniques are generally used for surface and geometrical modifications to the flow channel by introducing inserts or additional devices in which existing flow mechanism is disturbed and heat transfer performance is improved with an increase in friction and pressure drop. In turbulent flow the fluid is chaotic in nature involving crosswise mixing of the main stream. In turbulent flow, the fluid properties such as instantaneous velocity, temperature and pressure are subject to fluctuations. These fluctuations increase the heat transfer rate and resistance to flow. This resistance causes large pressure loss and required pumping power.

A wide range of insert has been introduced when turbulent flow is considered in heat transfer work.

Ahamed et. al. [1] state an experimental investigation that was carried by focusing on heat transfer coefficient and friction factor characteristics of air for turbulent flow in a circular tube fitted with perforated twisted tape inserts. The Reynolds numbers were varied in the range 1.3×10^4 -5.2×10^4 with constant heat flux. With perforation of 4.6% gives the highest heat transfer rate for the same Reynolds number and is around 1.8 times the value of the plain tube at the cost of increase of pumping power by 1.2 to 2.5 times compared to that of smooth tube. The friction factor was high at the inlet of the test section and drops sharply toward the downstream then becomes almost constant. Salam et. al.[2] conducted an experimental investigation for measuring tube-side heat transfer coefficient, friction factor, heat transfer enhancement efficiency of water for turbulent flow in a circular tube fitted with rectangular-cut twisted tape insert. The Reynolds numbers were varied in the range 10000-19000 with heat flux variation 14 to 22 kW/m^2 of smooth tube and 23 to 40 kW/m^2 for tube with inserts. At comparable Reynolds number, Nusselt numbers in tube with rectangular-cut twisted tape insert were enhanced by 2.3 to 2.9 times at the cost of increase of friction factors by 1.4 to 1.8 times compared to that of smooth tube. Heat transfer enhancement efficiencies were found to be in the range of 1.9 to 2.3 and increased with the increase of Reynolds number. Murugesan, et. al. [3] performed numerical simulation to study the fluid flow and heat transfer in a tube with staggered twisted tapes with central holes. In the range of Reynolds number between 6000 and 28000, the modified twisted tape increased the Nusselt number by 76.2% ~149.7% and friction fractor by 380.2~443.8% compared to smooth tube. The holes in the modified tapes reduced the severe pressure loss. It was observed that modified tapes decreased friction factor by 8.0 ~16.1% and enhanced heat transfer by 34.1~46.8%. The performance ratio was higher than 1.0 in the range of Reynolds numbers studied. Ozceyhan, et. al. [4] conducted numerical study which was undertaken for investigating the heat transfer enhancement in a tube with the circular cross sectional rings in the range of Reynolds number 3000 and 50000. Uniform heat flux was applied to the external surface of the tube and air was selected as working fluid. In analysis the Nusselt number was varied from 20 to 90 and friction factor was varied from 0.04 to 0.025 for increasing Reynolds numbers. Overall enhancement of 18% was achieved. Hossain, et. al. [5] analyzed heat transfer enhancement in circular tube with and without inserts for laminar flow in the range of Reynolds number,Re= 1600~2400 by using COMSOL Multiphysics to perform CFD simulation. Here a non-isothermal flow model was considered in which water was taken in the model and copper was considered as material of circular pipe under constant heat flux of 32.087 KW/m^2. Using governing equation of non-isothermal flow together with continuity equation, the dynamic behavior of the flow was described which transport heat. In the simulation four, six, eight inserts were used for experimental length 800mm and they got highest output temperature 319.28 K for four insert while the output temperature was 307.85 when there was no insert in the tube. Other than four inserts, the outlet temperature was decreased. Authors gave an important decision that not only increasing the number of inserts will increase the heat transfer but the distance among the inserts also need to be considered. Anvari et. al. [6] studied forced convective of water in horizontal tubes with conical tube inserts experimentally. The transient flow regime was used for the tests. Experimental results were validated with existing well established correlation. The turbulators were placed in two different arrangements: converging conical ring, referred to as CR array and diverging conical ring, DR array. Two correlations for the Nusselt number based on the experiment were introduced for practical use. It is found that the insertion of turbulators has enhanced the Nusselt number for the DR arrangement up to 521%, and for the CR arrangement up to 355%, although using the turbulators cause a significant increase in pressure drop. Bhuiya et. al. [7] explored the effects of perforated double counter twisted tapes on heat transfer and fluid friction characteristics in a heat exchanger tube. The study was conducted in turbulent flow regime with Reynolds number ranging from7200 to 50,000 using air as the working fluid under uniform wall heat flux boundary condition. Nusselt number, friction factor and thermal enhancement efficiency were increased with decreasing porosity. . In the range of the investigation, heat transfer rate and friction factor were obtained to be around 80 to 290% and 111 to 335% higher than those of the plain tube values, respectively. Based on constant blower power, the highest thermal enhancement efficiency of 1.44 was achieved.

A number of inserts were used when turbulence is considered in heat transfer simulation [5] which concludes that 4 inserts provides optimized result among the set of inserts. This paper refers to result of using 4 inserts practically in laboratory experiment and also how the different angular orientation of inserts influences the result. There were various obstacles in performing this experiment which include: Difficulty of inserting the inserts in the tube; Proper dimensions and orientation of the insert could not be maintained due to practical limitations; Using this kind of insert caused high pressure drop.

2. METHODOLOGY

The test section was a smooth circular tube of copper having 26.6 mm inside diameter, 30 mm outside diameter and 900 mm long as shown in Fig.3.12. Five K-type thermocouples was used in test section [Fig.3.7] which were at equal distance from each other and it was also maintained at a certain distance from the entrance and exit of the pipe. In order to prevent leakage Teflon tape was used at the joining of tubes and after that M-seal was

Fig.1: Designed 4 inserts layout at 45° orientations

Fig.2: Designed isometric cross sectional view of pipe with inserts

used. A constant heat flux had been maintained by wrapping Nichrome wire spirally around the test section and fiber glass insulation over the wire. Mica tape was used to wrap the tube before wrapping it with Nichrome wire and after with insulation. Two thermometers were used at inlet and outlet section for measuring bulk temperatures. Pressure drop was measured of the test section by using manometer. Open loop system of water supply was used. The rate of water flow was measured by Rota meter that was installed at the travel path of inlet water. Two types of temperature were measured during the experiment. One regarding outer surface temperature and other regarding water inlet-outlet temperature. Data was taken for only plain copper tube without insert and with inserts.

Fig.3: Photograph of experimental setup

3. DATA REDUCTION

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

$$Q = mC_p(T_o - T_i) \tag{3.1}$$

Heat transfer coefficient was calculated from

$$h = \frac{Q}{A_s(T_{wi} - T_b)}$$

where $A_s = \pi d_i L$ (3.2)

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

$$T_b = \frac{T_{in} + T_{out}}{2} \tag{3.3}$$

Tube inner surface temperature was calculated from one dimensional radial conduction equation,

$$T_{wi} = T_{wo} - Q\frac{ln\left(\frac{d_o}{d_i}\right)}{2\pi K_w L} \tag{3.4}$$

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

$$T_{wo} = \sum_{k=1}^{5} \frac{T_{wo,k}}{5} \tag{3.5}$$

Theoretical Nusselt number was calculated from Dittus-Boelter equation [8],

$$Nu_D = 0.023\, Re^{\frac{4}{5}}\, Pr^n \tag{3.6}$$

Where n is 0.4 for heating and 0.3 for cooling.

Theoretical friction factor was calculated from Petukhov [9],

$$f = (0.79\, ln\, Re - 1.64)^{-2} \tag{3.7}$$

$$Re = \frac{\rho \vartheta d_i}{\mu} \tag{3.8}$$

$$\Pr = \frac{\mu C_p}{K} \tag{3.9}$$

Experimental Nusselt Number was calculated from,

$$Nu = \frac{hd_i}{K} \tag{3.10}$$

Mean water velocity will be obtained from obtained from,

$$\vartheta = \frac{\dot{m}}{A_f} \tag{3.11}$$

Flow area was obtained from,

$$A_f = \frac{\pi}{4}\, {d_i}^2 \tag{3.12}$$

Friction factor, f can be calculated from,

$$f = \frac{2\Delta P D_i}{\rho L\, \vartheta^2} \tag{3.13}$$

ΔP is the pressure drop across tappings. All the fluid properties were evaluated at bulk temperature.

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4. RESULT AND DISCUSSION

From result it has been found that in case of plane tube (without any insert) heat transfer phenomena are increased by Q = 871.32 ~ 1886.15W from Eq.(3.1), h= 1195.14 ~4662.12 $W/m^{2\,\circ}C$ from Eq.(3.2), Nu_{exp} = 137.08 ~ 414.53 from Eq.(3.10) with the Reynolds number 5057.04 ~ 25251.45 from Eq.(3.8) but costs gradual increase of pressure drop (f_{exp} = 0.280 ~ 0.0421 from Eq.(3.7)). On the other hand, it has been found that when inserts are introduced in the copper tube, heat transfer phenomena are increased compared to the plain tube because of developing axial and radial flow which were responsible for breaking down of water film. But it costs much more pressure drop compared to the plain tube. 4 inserts were introduced in 0º, 45º and 90º orientations. In case of every orientation introduced with inserts, there were huge variation of heat transfer and pressure drop. It has been noticed that in case of 4 inserts at 45º orientation provide higher heat transfer rate and lower pressure drop than 0º and 90º oriented inserts. So we can say that their orientations dominate heat transfer and pressure drop.

At first trial with 4 rectangular inserts the orientation between each inserts were kept to 0º. From result it has been found that heat transfer phenomena are increased (Q = 1568.37 ~ 3862.6 W, h= 1647.65 ~ 10636.88 $W/m^{2\,\circ}C$, Nu_{exp} = 70.68 ~ 456.35) with the Reynolds number (5057.04 ~ 25251.45) but costs gradual increase of pressure drop (f_{exp} = 1.75 ~ 0.105).

With 4 rectangular inserts the orientation between each inserts were kept to 45º. From result it has been found that heat transfer phenomena are increased (Q = 697.05 ~ 3650.77 W, h=1687.15 ~ 11339.31 $W/m^2{}^\circ C$,Nu_{exp} = 72.38 ~ 486.49) with the Reynolds number (5057.04 ~ 25251.45) but costs gradual increase of pressure drop (f_{exp} = 1.78 ~ 0.108).

With 4 rectangular inserts the orientation between each inserts were kept to 90º. From result it has been found that heat transfer phenomena are increased (Q = 1568.37 ~ 4187.35W, h=2070.68 ~ 12678.98 $W/m^2{}^\circ C$,Nu_{exp} = 88.83 ~ 533.96) with the Reynolds number (5057.04 ~ 25251.45) but costs gradual increase of pressure drop (f_{exp} = 1.75 ~ 0.111).

So with the introduction of the inserts the heat transfer efficiency definitely increases. As from previous study by Hossain, et. al. [5] shows that this efficiency is effected by the spacing between each insert and also the number of inserts. But this study further illustrates that the orientation at which the inserts are kept also significantly influence the heat transfer efficiency. The results show that 4 inserts at 45º gives the optimized output.

5. TABLES AND FIGURES

From Fig.4 we see that convective heat transfer coefficient (h) is getting higher value with the increasing of Reynolds number. We see that heat transfer coefficient is higher for 45º orientation than other orientations.

From Fig.5 we see that experimental Nusselt number (Nu_{exp}) is getting higher value with the increasing of Reynolds number and for 4 inserts the 45º orientations give relatively higher value than other sets of insert.

From Fig.6 we see that heat transfer rate (Q) is getting higher value with the increasing of Reynolds number and we see that heat transfer rate (Q) is higher for 45º orientation.

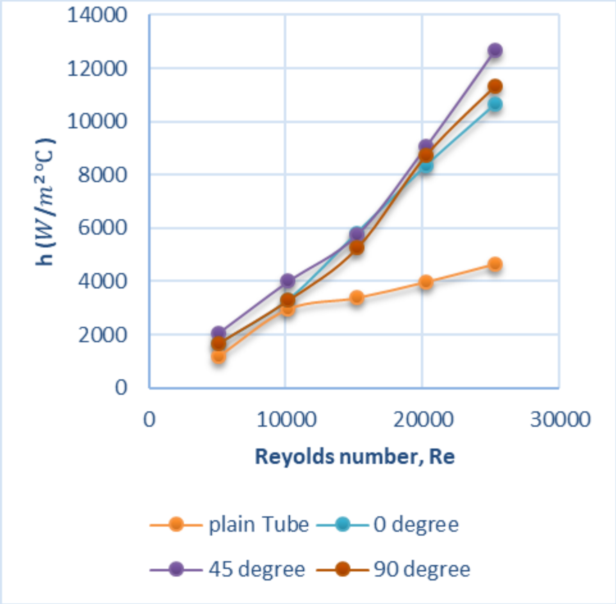

Fig.4: Variation of convective heat transfer coefficient (h) with Reynolds number (Re) for 4 inserts

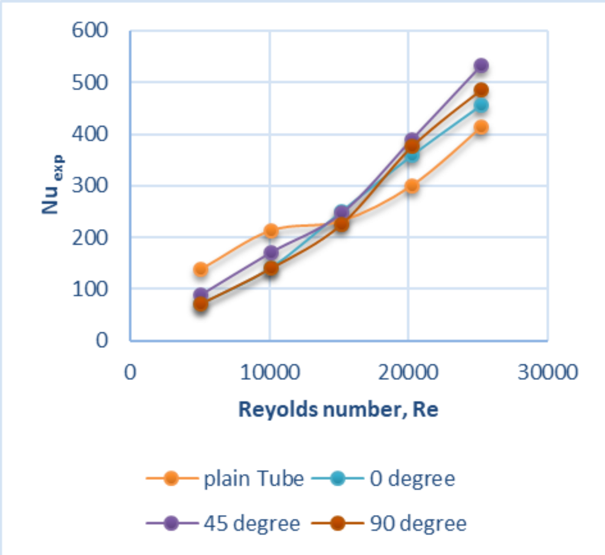

Fig.5: Variation of experimental Nusselt number with Reynolds number for 4 inserts

From Fig.7 we see that there is high cost of pressure drop relative to plain tube. The reasons can be easily understood. We observe drastic variation in the graph.

In Fig.7 we see that pressure drop almost similar for 4 inserts at every introduced orientation though we have got higher heat transfer for 45º oriented inserts in our study. So we can say heat transfer depends not only upon inserts distance but also their orientation

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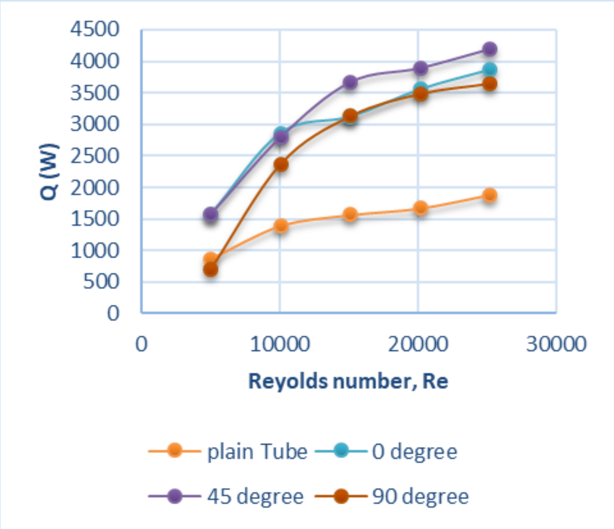

Fig.6: Variation of heat transfer rate (Q) with Reynolds number for 4 inserts

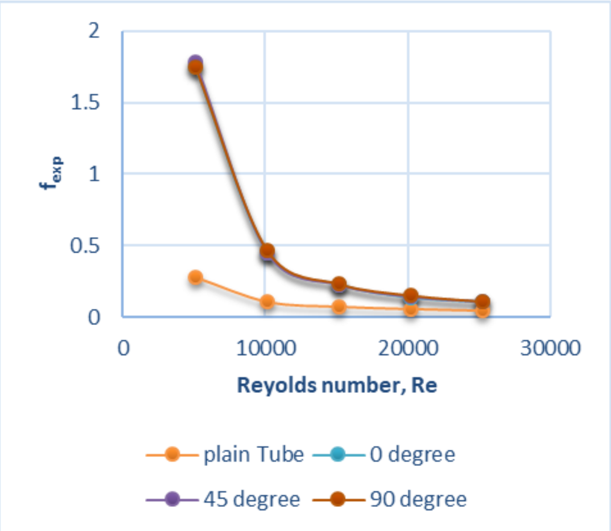

Fig.7: Variation of experimental friction factor (f)with Reynolds number for 4 inserts

6. CONSCLUSIONS

The main purpose of the inserts is making disturbance in flow in order to enhance heat transfer. By the design, there may be confusion with inserts and fins. Due to practical design limitations, these inserts touched the internal surface of the tube and thus must perform heat transfer along the inserts as the fin does. But as our objective is to enhance heat transfer, this fin type activity should be considered as a plus point.

The dimensions of inserts, their angular orientations, spacing that have been introduced in our experiment were difficult to maintain precisely. The thermocouple monitor which has been used does not give fraction. So our study obviously inherits some errors. The effect of inserts supporting wire on this experiment has been neglected though it of course has significant role in heat transfer enhancement as a straight insert.

This empirical investigation on tube side heat transfer enhancement with inserts with different orientations and spacing finds effect on h, Q, f and Nu and concludes that heat transfer is maximum and pressure drop is optimum when 4 inserts is used with 45° orientations. This optimum arrangement increases heat transfer coefficient (h) up to 172%, Nusselt number (Nu) up to 29%, heat transfer rate up to 122%, friction factor up to 163% which is very promising. It is hope that this study will help different persons having interest in heat transfer related region.

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9. NOMENCLATURE

Symbol	Meaning	Unit
T_o	Hot water temperature	(ºC)
T_i	Cold water temperature	(ºC)
T_1	Temperature of thermocouple 1	(ºC)
T_2	Temperature of thermocouple 2	(ºC)
T_3	Temperature of thermocouple 3	(ºC)
T_4	Temperature of thermocouple 4	(ºC)
T_5	Temperature of thermocouple 5	(ºC)
T_b	Bulk temperature	(ºC)
T_{wo}	Outer surface temperature	(ºC)
T_{wi}	Inner surface temperature	(ºC)
m	Mass flow rate	(kg/s)
C_p	Specific heat	(J/kg.K)
Q	Heat transfer rate	(W)
L	Tube length	(m)
d_i	Inner tube diameter	(m)
d_o	Outer tube diameter	(m)
K_w	Thermal conductivity of water	(W/m.K)
F	Friction factor	Dimensionless
Pr	Prandtl number	Dimensionless
R	Reynolds number	Dimensionless
Nu_{exp}	Experimental Nusselt number	Dimensionless