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HEAT TRANSFER ENHANCEMENT IN A CIRCULAR TUBE BY USING RECTANGULAR INSERTS AT FIXED ORIENTATION FOR DIFFERENT SPACING

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Abstract- *An experiment has been conducted with rectangular inserts in a circular tube in turbulent region (5000<Re<26000) for heat transfer enhancement. Trends of Heat transfer rate, heat transfer coefficient and friction factors has been studied for inserts at different spacing at a fixed orientation. Rectangular inserts were positioned perpendicularly to the flow directions at 45*^{0} *and at equal spacing of different lengths from each other for 4, 6, 8 and 10 inserts. The purposes of the inserts were to accelerate the mixing of fluid particle which substantially increases the heat transfer. However, pressure drop accompanies and compounds the pumping cost. Thus, this empirical study investigated to find the optimum profits between increased heat transfer and added frictional losses. The best results were found from the experiment was for 4 inserts where heat transfer rate increased by 122% and leads to proliferation in friction factor up to 163% compared to smooth tube.*

Keywords: Heat transfer enhancement, Rectangular insert, Insert spacing, Turbulent flow

1. INTRODUCTION

Reducing size and boosting effectiveness of heat exchanging devices has become a desideratum for technological advancement in heat transfer and many other fields for that study of heat transfer enhancement has emerged as an important topic to researcher. Heat transfer enhancement is achieved by two methods i.e., passive and active. The active enhancement techniques have been studied widely which requires addition of external power to bring about desired flow modification. On the other hand, passive method exploits different ingenious geometrical modification 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. The inserts studies include coil wire inserts, twisted tape inserts with and without perforation, rectangular or U-cut twisted tape inserts, strip inserts etc.

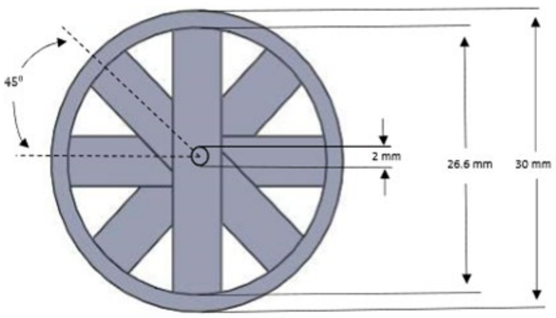
Heat transfer coefficient and friction factor characteristics of air for turbulent flow in a circular tube fitted with perforated twisted tape inserts were experimentally investigated by Ahamed et al. [1]. Maximum heat transfer rate was found with perforation of 4.6% and 1.8 times more than for smooth tube. Pumping power increases up to 1.2 to 2.5 times of smooth tube. Rectangular-cut twisted tape inserts [2] has also demonstrated good heat transfer enhancement capability. Heat flux variations were 23 to 40 Kw/m^2 for inserts with Reynolds number ranging from 10000-19000. Heat transfer efficiency increases 1.9 to 2.3 times the plain tube. Anvari et al. [3] studied forced convective of water in horizontal tubes with conical tube inserts experimentally. Diverging conical arrangement enhanced Nusselt number up to 521% and converging conical ring increases 355%, although pressure drop was significant. Conical ring coupling with twisted tape inside a circular tube for Reynolds numbers from 6000 to 26000 [4] has shown to increase Nusselt number up to 4 to 10%, maximum heat transfer rate 367% contrast to the conical-ring for twist ration of 3.75. Numerical study on various complex geometry for heat transfer enhancement also can be found in the literature. Numerical investigation of staggered twisted tapes with central holes [5] has exhibited amplification of Nusselt number by 76.2% ~ 149.7% and friction factor by 380.2 ~ 443.8% compared to smooth tube. Ozceyhan, et al. [6] 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 to 50000. Overall 18% enhancement was achieved applying uniform heat flux to the external surface in the range of Reynolds number 3000 and 50000.

Although, complex geometries similar to the above discussion demonstrate descent heat transfer enhancement characteristics, however these geometries pose other nuisance associated with them. Simpler geometry needs to be addressed to tackle conundrums related to pressure drop, heat exchanger size, and material cost. Literature shows that rectangular inserts is a noble candidate to offer solution to these problem. Investigation on the heat transfer enhancement characteristics of rectangular inserts can be found in literature nonetheless not complete. CFD simulation of rectangular inserts inside a circular tube [7,8,9] indicated that heat transfer enhancement competence doesn't proliferate linearly with number of inserts and spacing between inserts needs to be taken into account. Another experimental analysis on rectangular inserts [10] has proven that orientation of inserts also effects the enhancement capability. Optimum heat transfer enhancement provided with minimum friction factor has found to be for 45^0 orientations.

The scope of the present work is to empirically determine the spacing between the inserts in a circular tube for which maximum heat transfer enhancement and minimum power consumption can be achieved.

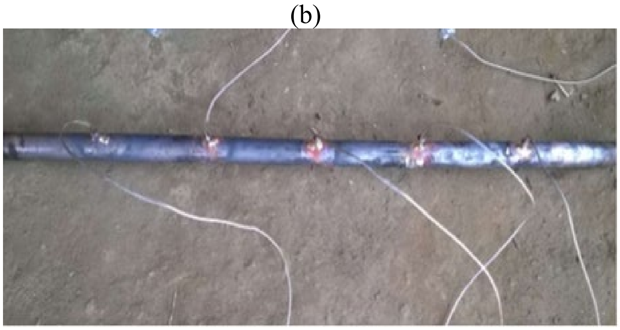
2. EXPERIMENTAL SETUP

A 900 mm long copper tube of 26.6 internal diameter and 30 mm outer diameter was used as the test section. A constant heat flux had been maintained by wrapping nichrome wire around the test section and fiberglass insulation over the wire. The inserts were perpendicular to the flow direction and were oriented at 45^0 relatives to each other shown in Fig. 1(a) and 1(b) at equal spacing of different lengths from each other for 4, 6, 8 and 10 inserts which were supported by a wire of 2 mm diameter which passes through the center of each inserts. The inserts were rectangular shape and made of aluminum of length 26.5 mm, width 5mm, thickness 2 mm. Outer surface temperature of the tube was measured at five points of the test section by maintaining equal distance from one point to another by k-type thermocouples and two thermometers were used at inlet and outlet section shown in Fig. 1(c) for measuring bulk temperatures. Pressure drop of the test section was measured by using manometer. Open loop water supply system was used and the rate of water flow was measured by a rotameter that was installed at the travel path of inlet water. Initially, water was delivered to the system by a pump and flow rate was 5 to 25 L/min and was controlled by a regulating rotameter. Constant voltage was applied to the heater by a voltage regulator to regulate the applied heat to the nichrome wire.



(a)

(b)



(c)

(d)

FIGURE 1. (a) Cross sectional view of the designed tube with inserts at 45° orientation (b) 6 inserts layout at 45° orientations (c) Photograph of copper tube after the thermocouple brazing (d) Photograph of the experimental setup

Temperature was recorded when thermometer and thermocouples exhibited a steady reading. Pressure drop was recorded from the manometer. Data was taken for plain tube first than with different number of inserts.

3. METHODOLOGY

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

$$Q = \dot{m} C_p (T_o - T_i) \tag{1}$$

Heat transfer coefficient was calculated from,

$$h = \frac{Q}{A_s (T_{wi} - T_b)} \tag{2}$$

Where, $A_s = \pi d_i L$

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

$$T_b = \frac{T_i + T_o}{2} \tag{3}$$

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

$$T_{wi} = T_{wo} - Q\frac{\ln(\frac{d_o}{d_i})}{2\pi K_w L} \tag{4}$$

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

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

Theoretical Nusselt number was calculated from Dittus-Boelter equation [11],

$$Nu = 0.023\,\mathrm{Re}^{\frac{4}{5}}\,\mathrm{Pr}^{n} \tag{6}$$

Where, n is 0.4 for heating and 0.3 for cooling

Theoretical friction factor was calculated from Petukhov [12],

$$f = (0.79\ln \mathrm{Re} - 1.64)^{-2} \tag{7}$$

$$\mathrm{Re} = \frac{\rho v d_i}{\mu} \tag{8}$$

$$\mathrm{Nu} = \frac{h d_i}{K} \tag{9}$$

Mean water velocity will be determined from,

$$v = \frac{\dot{m}}{A_s} \tag{10}$$

Flow area was obtained from,

$$A_f = \frac{\pi}{4} d_i^2 \tag{11}$$

Friction factor, f can be calculated from,

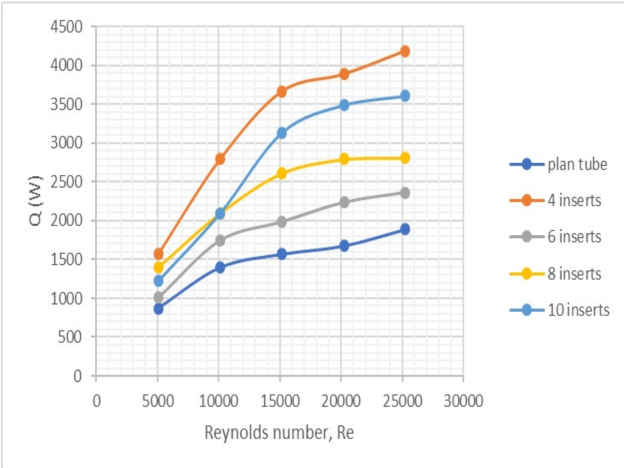
$$f = \frac{2\Delta P d_i}{\rho L v^2} \tag{12}$$

Where, ΔP is the pressure drop across tapings. All the fluid properties were evaluated at bulk temperature.

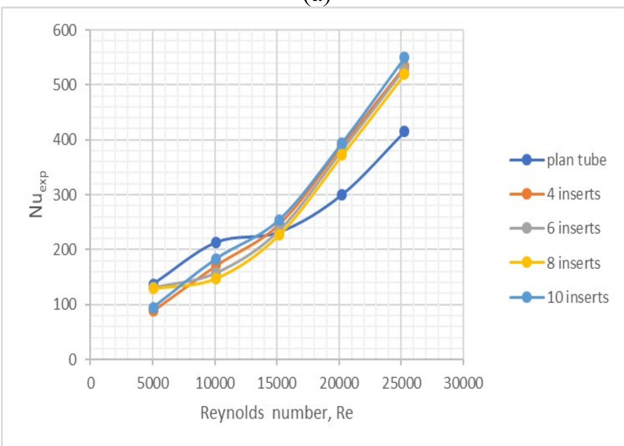
4. RESULTS AND DISCUSSION

In Fig. 2(a) we have plotted the total heat transfer rate Q in the tube without and with different inserts arrangement for different Reynolds numbers. It is conspicuous from the figure that heat transfer rate increases with Reynolds number. However, 4 inserts provide higher heat transfer rate and which is substantially greater than 6, 8, and 10 inserts. Total heat transfer rate Q = 1568.37 to 4187.35 was found with Re= 5057.04 to 25251.45 for 4 inserts.

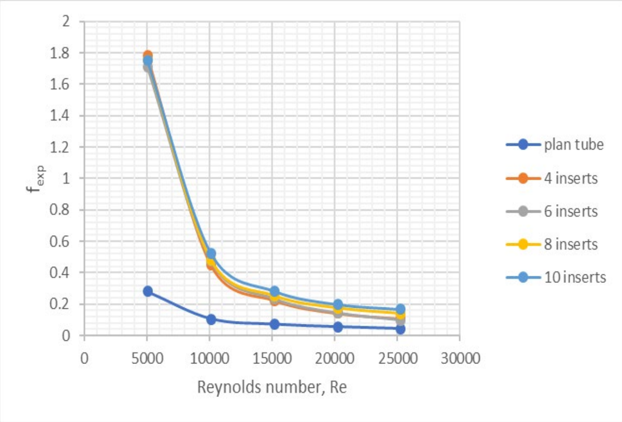
Figure 2(b) shows that Nusselt number increases with Reynolds number and this trend conforms to the theoretical relation. Addition of inserts into the tube accelerated Nu number relative to plain tube by breaking the boundary layer and developing radial and axial flow, thus, instigating further mixing of fluids. Nu number is maximum and approximately equal for 4 and 10 inserts and their maximum difference is 7% for Re= 10114.06.



(a)



(b)



(c)

FIGURE 2. Variation of (a) heat transfer rate (Q) with Reynolds number (b) experimental Nusselt number with Reynolds number (c) experimental friction factor with Reynolds number

For Re=10114.06, experimental Nu number seems to be maximum for plain tube and this discrepancy with other Re number results might be due to error associated with the experiment. To find the optimum spacing we have to consider the friction factor for different inserts arrangement. Figure 2(c) illustrates experimental friction factor with its corresponding Reynolds number. Friction factor decreases with Reynolds number and minimum friction factor is found to be for plain tube as plain tube causes less pressure drop. Introduction of inserts causes further pressure drop and increases friction factor which is palpable from the fig. 2(c). Among all inserts arrangement, 4 inserts provide less friction factor for all Reynolds number when compared to other inserts. Four inserts experimental friction factor f_{exp} =1.71 to 0.098 and 10 inserts f_{exp}=1.75 to 0.168 for Reynolds number Re= 5057.04 to 25251.45.

So, addition of inserts increases heat transfer enhancement. Hossain, et al. [7] previous numerical study shows that heat transfer efficiency is influenced by the spacing between each insert. This experimental investigation confirms that spacing between inserts have effect on heat transfer enhancement and empirically investigate which spacing delivers optimum results. Although, 10 inserts give maximum Nu number still heat rate and friction factor along with Nu number consideration shows that 4 inserts give the optimized output.

5. CONCLUSION

Heat transfer enhancement effects of inserts with 45^0 orientations for different spacing arrangement on a circular tube were investigated in turbulent regime (5000<Re<25000). 4, 6, 8 and 10 inserts were used as an adjunct into a circular tube to conduct the experiment. Heat transfer coefficient, Nusselt number, Heat transfer rate and friction factor data was extracted and juxtaposed to determine number of inserts consequently the spacing between the inserts for optimum results. The effect of inserts supporting wire on this experiment has been neglected though it has a significant role in heat transfer enhancement as a straight insert. However, why 4 inserts provide better results cannot be answered from this experiment and further investigation is necessary to discover the causes.

Following percentage increase was recorded relative to plain tube for the optimum spacing (4 inserts) of inserts from the experiment. Heat transfer coefficient (h) up to 172%, Nusselt number (Nu) increases up to 29%, heat transfer rate up to 122%, friction factor up to 163%

6. ACKOWLEDGEMENT

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8. NOMENCLATURE

Symbol	Meaning	Unit
T_i	Inlet temperature	(K)
T_o	Outlet temperature	(K)
T_b	Bulk temperature	(K)
T_{wi}	Surface temperature of the tube wall (inner side)	(K)
T_{wo}	Surface temperature of the tube wall (outer side)	(K)
L	Length	(m)
d_i	Diameter (inner side)	(m)
d_o	Diameter (outer side)	(m)
A_s	Cross sectional area	(m^2)
$\dot{m}$	Mass flow rate	(Kg)
ρ	Density	(Kg/m^3)
μ	Dynamic viscosity	(Pa.s)
k_W	Thermal conductivity of tube wall	(W/m.k)
C_p	Constant pressure specific heat capacity	(J/Kg.K)
v	Velocity	(m/s)
P	Pressure	(Pa)
h	Heat transfer coefficient	($W/m^2 \cdot K$)
Q	Heat transfer rate	(Kw)
Nu	Nusselt Number	
Re	Reynold number	
Pr	Prandlt number	
f	Friction fator	