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Experimental Investigation of Heat Transfer and Fluid Flow Characteristics Using S-shape Inserts through Circular Tube in Turbulent Zone

Rathod Saroj¹, Sable Mahendra²

^{[1][2]}Mechanical Engineering Department, Savitribai Phule Pune University

¹ JSPM Rajarshi Shahu School of Engineering and Research, Narhe, Pune, India

²JSPM Rajarshi Shahu School of Engineering and Research, Tathawade, Pune, India

Abstract— The performance of force convection heat transfer in tubes can be improved by use of inserts. This leads to fluid mixing and redevelopment of thermal boundary layer with resulting enhancement in heat transfer. In present work heat transfer and fluid flow characteristics in circular tube with pipe inserts are investigated experimentally. This paper includes study of heat transfer augmentation in circular tube of inner diameter 34mm and outer diameter 38mm by using deformed ‘S’ shape pipe inserts of 3mm length and 2mm thick. These inserts with different pitches are tested in turbulent zone of Reynolds number 4×10^3 to 20×10^3 . Air is used as working fluid. The comparisons of fluid flow characteristics of plain tube and inserted tube is studied. It is found that the Nusselt number and friction factor of inserted tube increases by 1.001 to 1.736 and by 1.006 to 1.26 than that of plain tube.

Keywords— Friction factor, Heat transfer coefficient, Reynolds number, S shape copper pipe inserts, Turbulent flow

Introduction

I. INTRODUCTION

Heat transfer devices plays an important role in many engineering applications such as air-conditioning, refrigeration, power generation, steel production, chemical industries, metallurgy etc. so increase of heat transfer rate of such devices is very crucial and important task for an engineer. Thermal performance of heat transfer devices can be increased or improved by heat transfer augmentation techniques. These augmentation techniques require low cost, ease of maintenance and can be installed rapidly. Basically main objective of heat transfer augmentation technique is that to obtain high rate of heat transfer with minimum increase in pressure drop. These techniques are classified into three types-active technique, passive technique and compound technique. Active technique requires some external power input source to cause desired flow modification and improvement in heat transfer. Active techniques are complex from design point of view. Augmentation of heat transfer by this method can be obtained by mechanical aids, surface vibration, fluid vibration, electrostatics fields, injection, suction, jet impingement. Passive technique doesn't require any external power source. These techniques generally use surface or geometrical modifications to the flow channel by introducing inserts or additional devices. In this technique high heat transfer coefficient is promoted by disturbing the flow behaviour. These also promote increase in pressure drop. Heat transfer enhancement by this method can be achieved by using treated surface, extended surfaces, displacement enhancement devices, swirl flow devices, coiled tubes, surface tension devices, additives for liquids, additives for gases. Compound technique in which one or more above mentioned techniques are used simultaneously for improving performance of heat exchanger. These techniques include complex design and hence have limited application.

Verities of techniques have been implemented by researchers for heat transfer enhancement in past years for increase in heat transfer rate with low increase in pressure drop which are discussed, Mao-Yu Wen, Ching-Yen Ho et al. [1] studied the characteristics of boiling heat transfer and pressure drop of R-600a in a circular tube with copper porous inserts and found that increase in mass velocity, heat flux and decrease in refrigerant quality and the sizes of inserts (the porosity, mean pore diameters and permeability) increase the heat transfer coefficients for the test tubes (2014). Xiayu Zang, Zhichun Liu et al. [2] studied numerical studies on heat transfer and friction factor characteristics of tube fitted with helical screw tape without core-rod inserts. They have studied H.T. & friction factor characteristics, synergy principle, Entropy generation analysis and overall HT performance. They reported that Avg. overall Heat transfer coefficients in circular plain tubes are enhanced with helical screw-tape of different width by as much as 21.2-35.1% at constant tube side temperature, friction factor are enhanced by as much as 33%-102%. The PEC value of the helical screw-tape insert of different width varies between 1.58 and 2.35. PEC value may reach 1.58 to 2.35 for helical screw tape as Re no. ranges in 4000 to 12000. The overall performance has improved greatly with helical screw tape without core rod insert (2013).

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Weerapun Duangthongsuk, Somchai Wongwises et al. [3] compared the heat transfer performance and friction characteristics between fixed and rotating turbine-type swirl generators fitted in a small circular tube and illustrate that the heat transfer performance of the tube with rotating turbine type swirl generator inserts is 6.3% and 56% higher than that of the fixed and freely rotating turbine type swirl generator and the conventional smooth tubes (2013). M.M.K Bhuiya et al. [4] studied thermal characteristics in heat exchanger tube fitted with triple twisted tape inserts and investigated the influence of triple twisted tapes on heat transfer rate, friction factor and thermal enhancement efficiency. They found that the Nusselt number and friction factor using the triple twisted tape increased up to 3.85 and 4.2 times than that of the plain tube (2013). Subhankar Saha et al. [5] studied enhancement of heat transfer of Laminar flow of viscous oil through a circular tube having integral helical rib roughness and fitted with helical screw tapes. They have studied friction factor, Nusselt no. for circular tube having integral helical rib roughness & fitted with Helical Screw tape insert. They reported that Due to combine enhancement tech. there is 32-49% increase in heat transfer for constant pumping power, also 25-37% reduction in pumping power for constant Heat duty depending upon the rib pitch & rib height. Helical screw tape insert in comb. With integral helical rib roughness performs better than individual enhancement tech (2013). C. Muthusamy, M. Vivar, K. Srihar et al. [6] studied the effect of conical cut-out tabulators with internal fins in a circular tube on heat transfer and friction factor for three pitch ratios 3, 4, 5 and tested with two arrangements as convergent mode and divergent mode. It is found that the D-tabulators arrangement with PR=3 shows maximum heat transfer rate of 315%, thermal performance factor of 2.4 and friction factor of 3.2 times that of plain tube (2013). Bodius Salam, Suman Biswas et al. [7] studied heat transfer enhancement in a tube using rectangular – cut twisted tape insert. They have studied Heat transfer co-efficient, friction factor, Heat transfer enhancement efficiency. They reported that NU_i is enhanced by 2.3 to 2.9 times, friction factor increase by 1.4 to 1.8 times as compare to smooth pipe. Heat transfer enhancement is increased as compare to smooth pipe is 1.9 to 2.3 times than smooth pipe; heat transfer enhancement increase with increase in Reynolds's no (2013).

M.M.K Bhuiya, M.S.U. Chowdhury et al. [8] studied heat transfer performance for turbulent flow through a tube using double helical tape inserts and investigated the influence of double helical tape on Nusselt number and friction factor. The result indicated that the Nusselt number and friction factor were increased up to 305% and 170% respectively than those over the plain tube and the maximum thermal performance was found to be 215% using double helical tape insert with helix angle 9° at high Reynolds number (2012). M.M.K Bhuiya, M.S.U. Chowdhury et al. [9] studied heat transfer enhancement and development of correlation for turbulent flow through a tube with triple helical tape inserts and investigated the influence of triple helical tape on Nusselt number and friction factor and effectiveness. The experiment showed that the Nusselt number, effectiveness and friction factor for the inserts were found to be up to 4.5, 3.45 and 3.0 times, respectively, over the plain tube (2012). A.W. Fan, J.J. Deng, W. Liu et al. [10] studied characteristics of heat transfer, flow resistance and overall thermo-hydraulic performance of turbulent air flow in circular tube fitted with louvered strip through numerical solution. The results show that the Nusselt number is augmented by 2.75-4.05 times that of plain tube. The performance evaluation criteria (PEC) lie in the range of 1.06-2.05, which shows that very good thermo-hydraulic performance (2012). M.M.K Bhuiya, M.S.U. Chowdhury, M. Saha et al. [11] studied heat transfer performance evaluation for turbulent flow through a tube with twisted wire brush with four different twisted wire densities of 100, 150, 200, 250 wires per centimetre by winding. The Nusselt number and friction factor of using the twisted wire brush inserts were found to be increased up to 2.15 and 2.0 times than those over the plain tube (2012).

Sibel Gunes, Veysel Ozceyhan et al. [12] studied the experimental investigation of heat transfer and pressure drop in a tube with coiled wire inserts placed separately from the tube wall. The experiments were performed with constant wire thickness of $a=6$ mm, three different pitch ratios. The results obtained such that the Nusselt number and friction factor increase with decreasing pitch ratio and distance for coiled wire inserts (2010). Pongjet Promvong, Smith Eiamsa-ard [13] studied heat transfer augmentation in a circular tube using V-nozzle tabulator inserts and snail entry and investigated the influences on heat transfer and friction loss characteristics. The values of Nusselt number and friction factor for utilizing both the V-nozzle and the snail entry are found to be higher than that for using V-nozzle alone or the plain tube (2007). Pongjet Promvong, Smith Eiamsa-ard [14] studied heat transfer and turbulent flow friction in a circular tube fitted with conical nozzle tubulators with two arrangements i.e. diverging nozzle and conical nozzle arrangement. The tabulators are inserted with various pitch ratios, PR=2, 4 and 7.0. The heat transfer rate obtained from using both nozzle tubulators are found to be higher than that from the plain tube at a range of 236 to 344% depending upon the Reynolds number and tubulator arrangement (2007).

II. MATHEMATICAL EQUATIONS

The Nusselt number and friction factor are calculated from experimental data at different Reynolds number in turbulent flow region.

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Equations used for calculation of parameters are listed below. The mass flow rate of air is calculated by,

$$\dot{m} = \rho A_c V_a$$

Where,

ρ is the density of air, A_c is the cross sectional area, V_a is the velocity of air. In the test section velocity of air is calculated as

$$V_a = \dot{m} / (\rho A_c)$$

ρ is obtained from bulk mean temperature.

The total heat generated by the heater is calculated as

$$Q_T = V_p I$$

Where,

V_p Is voltage and I is current,

The actual heat supplied Q_1 is calculated as

$$Q_T = Q_1 + Q_L$$

$$Q_1 = Q_T - Q_L$$

Where heat loss is based on insulation and is obtained by measuring average wall temperature and the ambient temperature.

The heat absorbed by the air is calculated as

$$Q_2 = \dot{m} C_p (T_o - T_i)$$

The average heat transfer coefficient is calculated as

$$Q_1 = h A_s (T_s - T_{bm})$$

The average Nusselt number is calculated as

$$Nu = (h D_i) / k$$

The friction factor is obtained as

$$f = \Delta p \cdot 2D_i / V^2 \rho$$

The Reynolds number is obtained as

$$Re = \rho V D_i / \mu$$

Where, μ is dynamic viscosity of fluid.

Prandtl number is given as

$$Pr = \mu C_p / k$$

III. EXPERIMENTAL SETUP

The experimental setup consists of test section, inlet section, outlet section, control panel, U tube manometer and blower. The test section is 1m long and it is insulated with asbestos insulation. It is made up of stainless steel. The inlet section is about 0.75m long and out let section is 0.25m long. The inner and outer diameter of the pipe is 0.034m and 0.038m. The inlet section equipped with orifice meter for measuring mass flow rate of air. Air flow rate through the blower is adjusted for obtaining Reynolds number. 12 K-type thermocouples are used for temperature measurement out of which 2 thermocouples are located at inlet and at outlet, remaining 10 thermocouples are used for measurement of surface temperature. The U tube manometer is used for measurement of pressure drop across test section. The input to the blower and heater is controlled by means of control panel which consists of voltmeter, ammeter, dimmer state, 12 channels digital temperature indicator. Inserts are shown in fig2. These inserts are made up of copper of 3mm length and 2mm thick. These are deformed in 'S' shape by bending the copper wire. Inserts are brazed on the copper rod of 3mm thick then it is inserted into the pipe. The pitches of inserts are varied as 16.7cm, 20cm and 25cm. The experiments are tested with different air flow rates for Reynolds number ranging from 4000 to 20,000. The heat flux is kept constant by adjusting the voltage regulator. During each test the experimental data is recorded after reaching to steady state. After steady state inlet, outlet, surface temperatures, pressure drop across test section and mass flow rate of air are recorded for the calculation of Nusselt number and friction factor.

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Fig. 1 Experimental setup

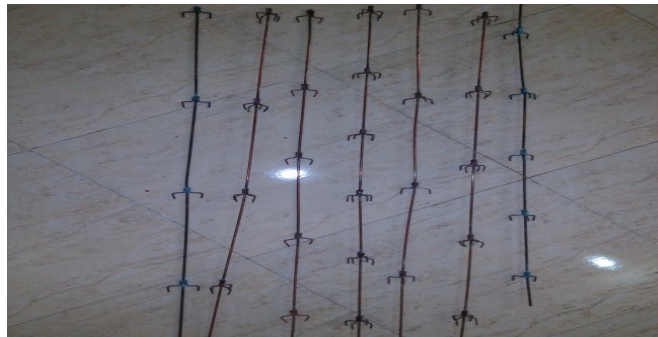


Fig. 2 S-shape copper brazed inserts

IV. RESULT & DISCUSSION

A. Validation of smooth tube

Verification of experimental data of smooth tube for good result is mentioned. The experimental Nusselt number and friction factor characteristics of smooth tube are compared and verified. The Nusselt number and friction factor data obtained from current smooth tube are validated with those from the proposed correlation by Dittus-Boelter for Nusselt number and correlation by Blasius for friction factor. These correlations are given below-

Nusselt number correlation of Dittus-Boelter:

$$Nu = 0.023 Re^{0.8} Pr^{0.4}$$

Friction Factor correlation of Blasius:

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

Heat transfer characteristics are shown in Fig3. From the figure it is seen that Nusselt number increases with increase in Reynolds number.

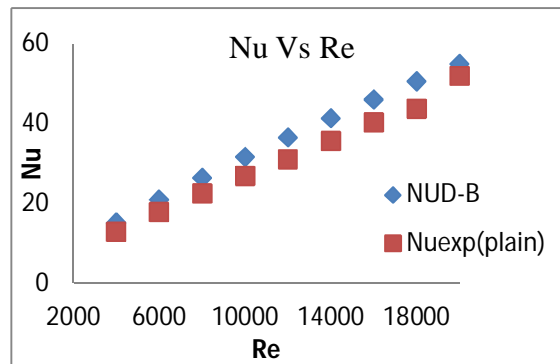


Fig3. Validation of Nusselt number

Friction Factor Characteristics are shown in figure 4. It is seen that Nusselt number increases as Reynolds number increases but in

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case of friction factor as Reynolds number increases friction factor decreases.

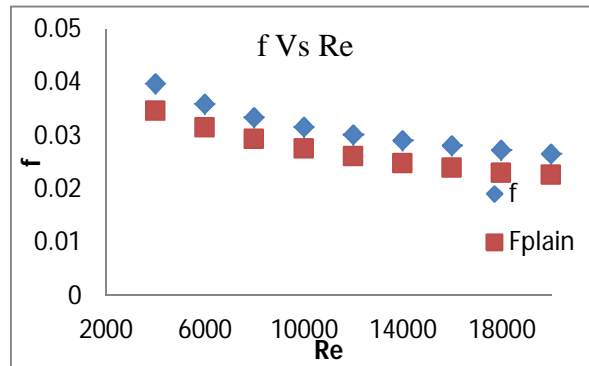


Fig4. Validation of friction factor

B. Effect of heat transfer enhancement

Effects of the small S-shape inserts on heat transfer rate in uniform heat flux are studied. It was found that Nusselt number increases with increase in Reynolds no. Variation of Nusselt number with Reynolds number for 4, 5 & 6 insert for 180°, 120°, 90° S-shape brazed inserts are shown in figure 5.

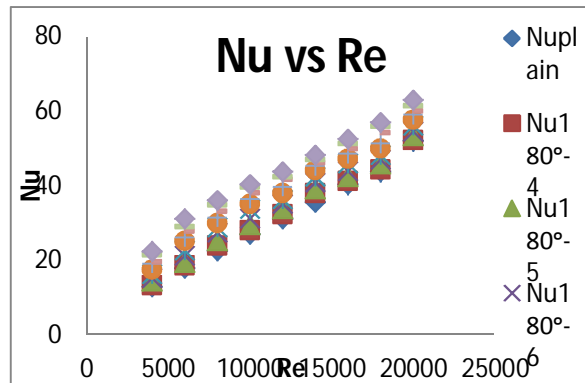


Fig5. Variation of Nusselt number with Reynolds number for 4, 5 & 6 insert for 180°, 120°, 90° S-shape brazed inserts

It was found that for 180° S-shape brazed insert Nusselt No increased by 4-21%, for 120° S-shape brazed insert Nusselt N Nusselt No increased by increased by 7-31% and for 90° S-shape brazed insert Nusselt No is increased by is increased by 17-42% as that of the plain tube.

C. Effect of friction factor

Variation of friction factor (f) with Reynolds number for 4, 5&6 insert for 90°, 120°, 180° S-shape brazed insert is shown in figure 6.

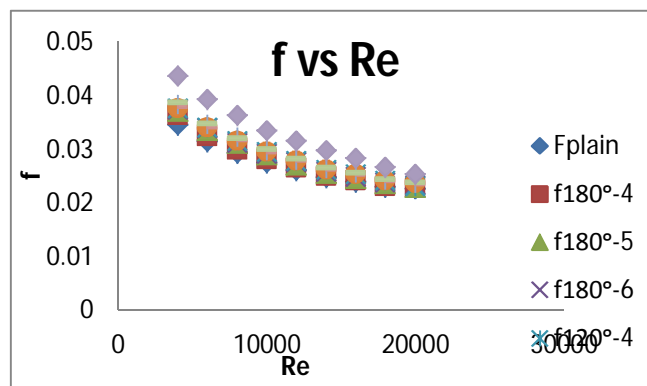


Fig6. Variation of friction factor (f) with Reynolds number for 4, 5&6 insert for 90°, 120°, 180° S-shape brazed insert

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From figure it can be seen that friction factor decreases with increase in Reynolds number. Friction factor values are highest for 6 insert of 90° S-shape brazed insert due to more disturbances created by this type.

D. Performance Evaluation

Figure 7 shows the variation of performance evaluation criteria with Reynolds number for the pipe equipped with S-shape small pipe insert. Figure shows the variation of thermal enhancement factor for 4, 5 & 6 insert for 90°, 120°, 180° S-shape brazed insert. From figure it is shown that thermal enhancement factor decreases with increase in Reynolds number. For all S-shape brazed insert, Thermal enhancement factor are higher than 1 (unity) for the Reynolds number up to 20000.

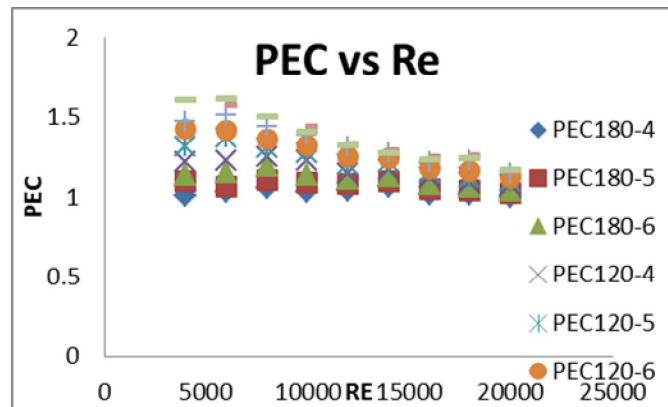


Fig7. Variation of Thermal enhancement factor with Reynolds number for 4, 5 & 6 insert for 90°, 120°, 180° S-shape brazed insert

VI. CONCLUSION

The friction factor and Nusselt number characteristics are investigated experimentally for smooth tube. The result shows that the experimental Nusselt number is $\pm 15\%$ than that of the Dittus-Boelter correlation and friction factor of smooth tube is $\pm 14\%$ than that of the friction factor obtained from Blasius correlation. It is observed that, Nusselt number for S-shape copper insert is higher than plain tube. The Nusselt number for S-shape copper insert is increased by 1.001 to 1.736 times as compare to plain tube for all range of Reynolds number. Friction factor for S-shape copper insert is higher than plain tube for all configurations. Thermal performance factor for 90°, 120° and 180° S-shape copper insert is 1 to 1.74. Thermal performance factor for 90° S-shape copper insert is higher due to higher mixing of hot and Cold fluid. Thermal performance of S-shape copper insert is higher for pitch of 16.7cm. It is due to more turbulence created due to more number of S-shape copper insert.

VII. ACKNOWLEDGMENT

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