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Experimental Investigations in a Circular Tube to Enhance Turbulent Heat Transfer Using Mesh Inserts

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Abstract

The present work focuses on experimental analysis of turbulent flow heat transfer enhancement in a horizontal circular tube by means of mesh inserts. Air is used as the working fluid. Sixteen types of mesh inserts are used with various screen diameters of 22 mm, 18 mm, 14 mm and 10 mm for varying distance between the screens of 50 mm, 100 mm, 150 mm and 200 mm. The horizontal tube was subjected to constant and uniform heat flux. The Reynolds number varied from 15,000 to 30,000. Initially the experiment was carried out without mesh inserts in the horizontal tube and then experiment is carried out using mesh inserts. Then both the results are compared for validation. An optimized relation is derived for both experimental and theoretical values. The optimized relation is derived on the basis of variation of heat transfer coefficient with diameter of mesh inserts and pitch between the mesh insertions.

Keywords: tube, heat transfer, mesh, turbulent flow, pressure drop, augmentation.

INTRODUCTION

In the recent years, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices. This can be seen from the exponential increase in world technical literature published in heat transfer augmentation devices, rowing patents and hundreds of manufacturers offering products ranging from enhanced tubes to entire thermal systems incorporating enhancement technology. Energy and materials saving considerations, space considerations as well as economic incentives have led to the increased efforts aimed at producing more efficient heat exchanger equipment through the augmentation of heat transfer. Among many techniques investigated for augmentation of heat transfer rates inside circular tubes, a wide range of inserts have been utilized, particularly when turbulent flow is considered. The inserts studied included twisted tape inserts, coil wire inserts, brush inserts, mesh inserts, strip inserts etc. The utilization of porous inserts has proved to be very promising in heat transfer augmentation. One of the important porous media characteristics is represented by an extensive contact surface between solid and fluid surfaces. The extensive contact surface enhances the internal heat exchange between the phases and consequently results in an increased thermal diffusivity. Different types of porous materials are extensively studied in forced convection heat transfer due to the wide range of potential engineering applications such as electronic cooling, drying processes, solid matrix heat exchangers, heat pipe, enhanced recovery of petroleum reservoirs etc. However the experimental work carried out in this area is limited.

Experimental studies conducted for heat transfer and pressure drop of laminar flow in horizontal tubes with/without longitudinal inserts (Shou-Shing Hsieh et al., 2003). They reported that enhancement of heat transfer as compared to a conventional bare tube at the same Reynolds number to be a factor of 16 at Re <= 4000, while a friction factor rise of only 4.5.

Hsieh and Kuo (Shou-Shing Hsieh et al., 2003) conducted experimental investigations for the augmentation of tube side heat transfer in a cross flow heat exchanger for turbulent flow of air by means of strip type inserts. They found that longitudinal strip inserts perform better than crossed strip (CS) and regularly interrupted strip (RIS) inserts for high Reynolds number (Shou-Shing Hsieh et al., 2003).

Hsiehand Wu (Shou-Shing Hsieh et al., 2000) conducted experimental studies on heat transfer and flow characteristics for turbulent flow of air in a horizontal circular tube with strip type inserts (longitudinal and Crossed Strip inserts). They reported that friction factor rise due to inclusion of inserts was typically between 1.1 and 1.5 from low Re (=6500) to high Re (=19500) with respect to bare tube. The experimental investigations of Hsieh and Liu (Shou-Shing Hsieh et al., 1996) report that Nusselt numbers were between four and two times the bare values at low Re an high Re respectively.

Bogdan I.Pavel (Bogdan I. Pavel et al., 2004) experimentally investigated the effect of metallic porous inserts in a pipe subjected to constant and uniform heat flux at a Reynolds number range of 1000-4500. The maximum increase in the length-averaged Nu number of about 5.2 times in comparison with the clear flow case and a highest pressure drop of 64.8Pa were reported with a porous medium fully filling the pipe.

Devarakonda Angirasa, 2001 performed experiments that proved augmentation of heat transfer by using metallic fibrous materials with two different porosities namely 97% and 93%. The experiments were carried out for different Reynolds numbers (17,000-29,000) and power inputs (3.7 and 9.2 W). The improvement in the average Nusselt number was about 3-6 times in comparison with the case when no porous material was used.

Fu (Fu. H.L et al., 2001) experimentally demonstrated that a channel filled with high conductivity porous material subjected to oscillating flow is a new and effective method of cooling electronic devices.

Mehmet Sozen (Mehmet Sozen et al., 1996) numerically studied the enhanced heat transfer in round tubes filled with rolled copper mesh at Reynolds number range of 5000-19,000. With water as the energy transport fluid and the tube being subjected to uniform heat flux, they reported up to ten fold increase in heat transfer coefficient with brazed porous inserts relative to plain tube at the expense of highly increased pressure drop.

Paisarn Naphon (Paisarn Naphon et al., 2006) had experimentally investigated the heat transfer characteristics and the pressure drop in horizontal double pipes with twisted tape insert. The results obtained from the tube with twisted insert are compared with those with out twisted tape.

Liao. Q (Liao. Q et al., 2001) carried out experiments to study the heat transfer and friction characteristics for water, ethylene glycol and ISOVG46 turbine oil flowing inside four tubes with three dimensional internal extended surfaces and copper continuous or segmented twisted tape inserts within Prandtl number range from 5.5 to 590 and Reynolds numbers from 80 to 50,000. They found that for laminar flow of VG46 turbine oil, the average Stanton number could be enhanced up to 5.8times with friction factor increase of 6.5fold compared to plain tube.

Betul Ayhan Sarac (Betul Ayhan Sarac et al., 2007) conducted experiments to investigate heat transfer and pressure drop characteristics of a decaying swirl flow by the insertion of vortex generators in a horizontal pipe at Reynolds numbers ranging from 5000 to 30000. They observed that the Nusselt number increases ranging from 18% to 163% compared to smooth pipe. Experimental investigation on heat transfer and friction factor characteristics of circular tube fitted with right-left helical screw inserts of equal length and unequal length of different twist ratios was done by (Sivashanmugam, et al., 2007). They observed that heat transfer coefficient enhancement for right left helical screw inserts is higher than that for straight helical twist for a given twist ratio. A maximum performance ratio of 2.97 was obtained by helical screw inserts. Heat transfer, friction factor and enhancement efficiency characteristics in a circular tube fitted with conical ring turbulators and a twisted-tape swirl generator were investigated experimentally by Promvonge (Promvonge, et al., 2007). Air was used as test fluid. Reynolds number varied from 6000 to 26000. The average heat transfer rates from using both the conical-ring and twisted tape for twist ratios 3.75 and 7.5, respectively are found to be 367% and 350% over the plain tube.The effect of two tube insert wire coil and wire mesh on the heat transfer enhancement, pressure drop and mineral salts fouling mitigation in tube of a heat exchanger were investigated experimentally (Pahlavanzadeh H. et al., 2007) with water as working fluid. The heat transfer rate averagely increased by 22-28% for wire coil and 163 -174% for wire mesh over a plain tube value depending on the type of tube insert, density of wire torsion and flow velocity. Pressure drop also increased substantially by 46% for wire coil and 500% for wire mesh. As Bogdan I. Pavel (Bogdan I. Pavel *et al*., 2004) carried out their work in a pipe with porous inserts in laminar and turbulent region with Reynolds number ranging from 1000-4500, the present work has been done similar lines but in turbulent region (Re number range of 7,000-14,000) as most of the flow problems in industrial heat exchangers involve turbulent flow region.

EXPERIMENTAL PROCEDURE

The apparatus consists of a blower unit fitted with a pipe in horizontal orientation. Nichrome bend heater encloses the test section to a length of a 40cm. Four thermocouples are embedded on the walls of the tube and two thermocouples are placed in the air stream, one at the entrance and the other at the exit of the test section to measure the temperature of flowing air shown in Figure-1.

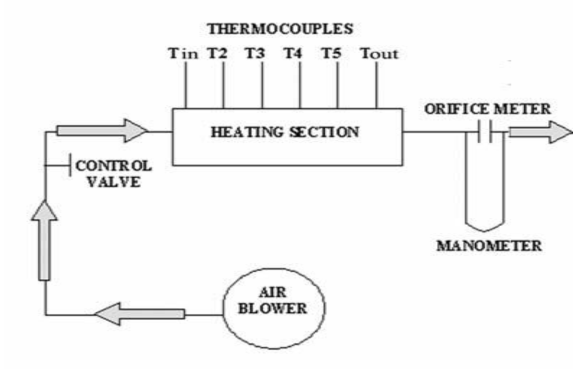


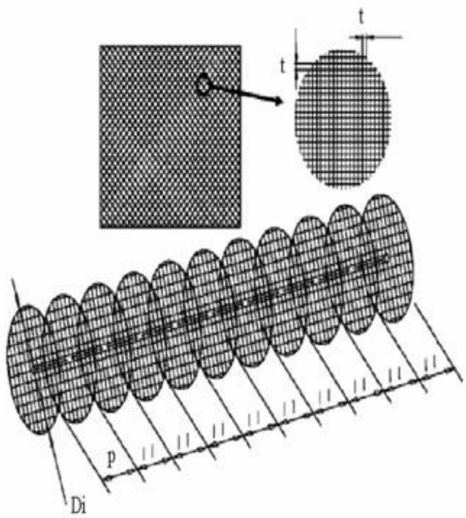
Figure-1. Experimental Setup.

The test pipe is connected with an orifice to measure the flow of air through the pipe. Input to heater is given through dimmer stat. The velocity of airflow in the tube is measured with the help of orifice plate and the water manometer fitted on the board. The inner tube of the heating part which is the test tube with inside diameter 27.5mm is made of 3.2 mm thick copper plate. A heat generating element is wound around this test tube so that the required heat input is given. The thermocouples (J-type) with accuracy ±0.4% are installed and drilled into the backside of the tube wall. Display unit consists of voltmeter, ammeter, dimmer stat and temperature indicator. Heat input can be varied by changing the voltage and current which are in turn altered by the dimmer stat position. The circuit was designed for a load voltage of 0-220 V, with a maximum current of 10 A. Outlet of the test pipe section is connected to an orifice meter and a manometer so that the pressure drop, mass flow rate of air can be measured.

The fluid properties were calculated as the average between the inlet and the outlet bulk temperature. It took 90mins to reach steady state conditions. Experiment was carried out at constant heat flux conditions and constant heat input of 40 W at different mass flow rates, with and without the inserts. In this, we assume that the air flowing through the circular tube to be hydro dynamically and thermally fully developed turbulent flow.

The porous media used for the experiments are Copper screens (wire diameter 0.28 mm) cut out at various diameters (D_i) and then inserted on copper rods as shown in Figure-2.

That is, 16 different inserts were obtained by varying the screen diameter and the distance between two adjacent screens (p). Due to insertion of the mesh inserts the hydraulic diameter reduces and the velocity in the pipe increases resulting in enhanced heat dissipation from the heating section.



SEQUENCE OF OPERATIONS

Experiments are carried out first without inserts and then with inserts.

Without inserts

Initially the experiment is carried out without any insert. The working fluid air flows through the pipe section with least resistance.

With inserts

In this, the mesh inserts with different diameters and pitches are taken as shown in Table-1. The mass flow rates considered for the constant heat input of 40 W in terms of water level difference in U-Tube water manometer are 2 inch, 3 inch, 4 inch and 5 inches(0.0047 to 0.0055(Kg/sec) of air).

PROCEDURE TO INSERT THE INSERTS

Photographic view of the inserts is shown in Figure-3. Each insert is taken and inserted into the test section axially. It is taken care that the strip doesn't scratch the inner wall of the pipe and get deformed. The presence of the insert in the pipe causes resistance to flow and increases turbulence.

Mesh Diameter (D_i)	**Pitch** (p)
22mm	50mm
18mm	100mm
14mm	150mm
10mm	200mm

Table-1. Shows the Mesh insert diameters along with different pitches.

2.4 Heat Transfer Calculations

Ts = (T2+T3+T4+T5)/4 (1)

Tb = (T1+T6)/2 (2)

Discharge of air,

$$d=C_d\sqrt{2gh(\rho m-\rho a)}\Big/\sqrt{\rho a(1-\beta)} \quad (3)$$

Velocity of air,

$$Va = {(Vo * Ao)}/{A} \quad (4)$$

Reynolds number, Re=U D/□ (5)

(To calculate Re while using mesh inserts, Dh instead of D is used)

Wetted perimeter= Π.(Di+D) (6)

Dh= 4g.(A_{fr}/Wetted perimeter)) (7)

Q =m.Cp.(T1-T6) (8)

h = Q/(A (Ts - Tb)) (9)

Nu = hD/K (12) (10)

Table 2: Mesh insert diameters along with different pitches.

Mesh insert Number	Diameter	Pitch	Rp
1	22	50	0.8
2	22	100	0.8
3	22	150	0.8
4	22	200	0.8
5	18	50	0.65
6	18	100	0.65
7	18	150	0.65
8	18	200	0.65
9	14	50	0.5
10	14	100	0.5
11	14	150	0.5
12	14	200	0.5
13	10	50	0.34
14	10	100	0.34
15	10	150	0.34
16	10	200	0.34

Table 3: Experimental and Theoretical values of heat transfer coefficient and Nusselt number for mesh diameter 22mm with various pitch values

Pitch (mm)	Experimental		Theoretical	
	H	Nu	H	Nu
50	155.09	91.33	253.42	ss149.23
100	153.09	90.87	241.72	143.41
150	150.07	89.64	239.51	142.49
200	148.57	88.58	236.17	140.82

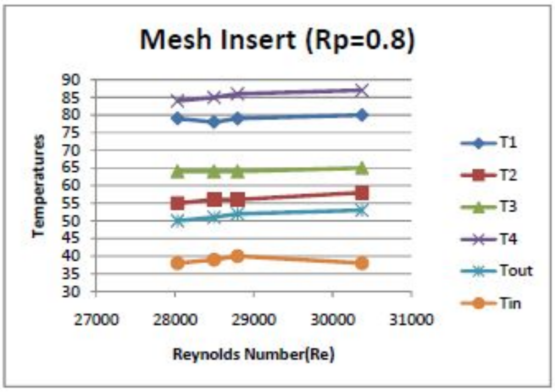
Table 4: Experimental and Theoretical values of heat transfer coefficient and Nusselt number for mesh diameter 18mm with various pitch values.

Pitch (mm)	Experimental		Theoretical	
	H	Nu	h	Nu
50	146	91.28	228.09	143
100	144.24	90.61	222.08	139.71
150	141.49	89.38	219.15	138.44
200	140.89	89.11	210.49	133.14

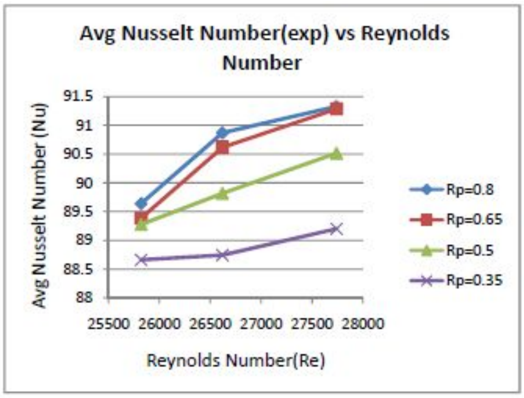
Results and discussions

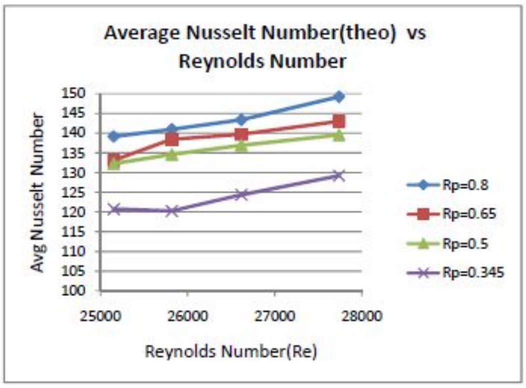
Experimentally determined Nusselt number values for plain tube (without mesh insert) are compared with Dittus-Boelter correlation.

Figure 2.0 shows the variation of surface temperature with Reynolds number for mesh insert of diameter 22mm. As the mesh is inserted into the horizontal tube of test section diameter 28mm turbulence is created thereby increasing the surface temperature.

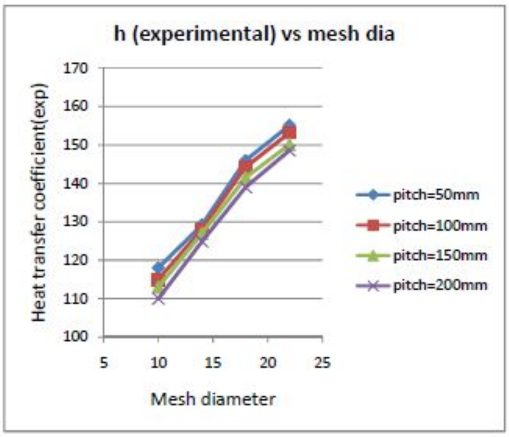


The variation of surface temperature with Reynolds number for mesh insert of diameter 18mm. As the diameter of mesh insert is reduced from 22mm to 18mm and inserted into the horizontal tube of test section diameter 28mm, the turbulence which is created in the tube reduced there by reducing the surface temperatures.

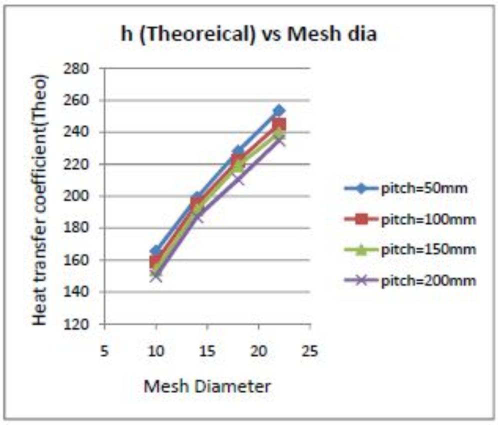




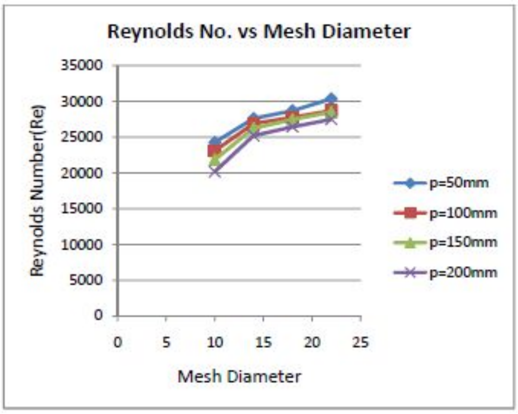
Our present work main objective is to enhance the heat transfer by inserting mesh inserts into the horizontal tube of test section diameter 28mm. Figure 2.6 shows the variation of experimental heat transfer coefficient with mesh diameter for different values of pitch. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest experimental heat transfer coefficient value when compared to the rest of mesh inserts. Figure 2.7 shows the variation of theoretical heat transfer coefficient with mesh diameter for different values of pitch. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest theoretical heat transfer coefficient value when compared to the rest of mesh inserts.



The variation of theoretical heat transfer coefficient with pitch for different values of mesh diameters. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest theoretical heat transfer coefficient value when compared to the rest of mesh inserts.



The variation of Reynolds number with pitch for different mesh inserts. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest Reynolds number when compared to the rest of mesh inserts.



From the above all discussions it is found that experimental and theoretical value of heat transfer coefficient is varying with mesh diameter and pitch. When the meshes are inserted into the horizontal tube hydraulic diameter is considered instead of mesh diameter. Hence heat transfer coefficient is varying with hydraulic diameter. Finally an optimized relation is derived in both experimental and theoretical cases.

Experimental Case

h = 54.9705 (d) 0.42247 (p)-0.05899

From the above relation heat transfer coefficient is directly proportional to mesh diameter and inversely proportional to pitch.

h = 6274.8 (dh)-1.221 (p)-0.0609

From the above relation heat transfer coefficient is inversely proportional to hydraulic diameter and pitch.

Theoretical Case

h = 55.7442 (d) 0.5503 (p)-0.0469

From the above relation heat transfer coefficient is directly proportional to mesh diameter and inversely proportional to pitch.

h = 26173.39 (dh)-1.5839 (p)-0.04942

From the above relation heat transfer coefficient is inversely proportional to hydraulic diameter and pitch.

Conclusion

Heat transfer coefficient depends on the parameters-mesh diameter, hydraulic diameter and pitch. When we consider Mesh diameter the relation with heat transfer coefficient is directly proportional to mesh diameter and is inversely proportional to pitch. When we consider Hydraulic diameter the relation with heat transfer coefficient is inversely proportional to mesh diameter and pitch. As mesh diameter increases Reynolds number and velocity of air increases. As hydraulic diameter increases Reynolds number and velocity of air increase. As pitch increases Reynolds number and velocity of air decreases. Effective heat transfer coefficient value is obtained at 22mm mesh diameter and 16.6mm hydraulic diameter with 50mm pitch value.

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