EXPERIMENTAL INVESTIGATIONS IN A CIRCULAR TUBE TO ENHANCE TURBULENT HEAT TRANSFER USING MESH INSERTS

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ABSTRACT

The present work shows the results obtained from experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube by means of mesh inserts with air as the working fluid. Sixteen types of mesh inserts with screen diameters of 22mm, 18mm, 14mm and 10mm for varying distance between the screens of 50mm, 100mm, 150mm and 200mm in the porosity range of 99.73 to 99.98 are considered for experimentation. The Reynolds number is varied from 7000 to 14000. Correlations for Nusselt number and friction factor are developed for the mesh inserts from the obtained results. It is observed that the enhancement of heat transfer by using mesh inserts when compared to plain tube at the same mass flow rate is more by a factor of 2 times where as the pressure drop is only about a factor of 1.45 times.

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, growing 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). Hsieh and 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 and 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.

Angirasa (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.

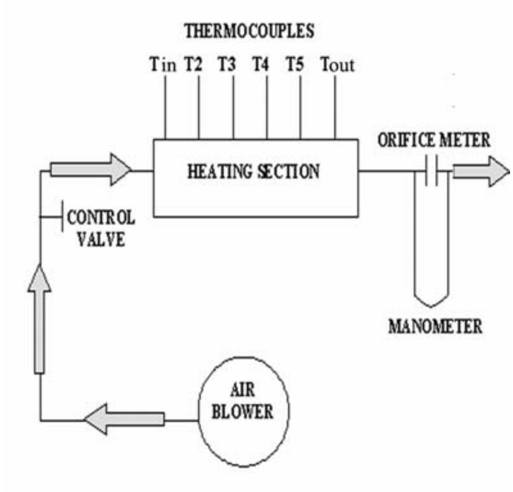

Figure-1. Experimental Setup.

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.

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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.

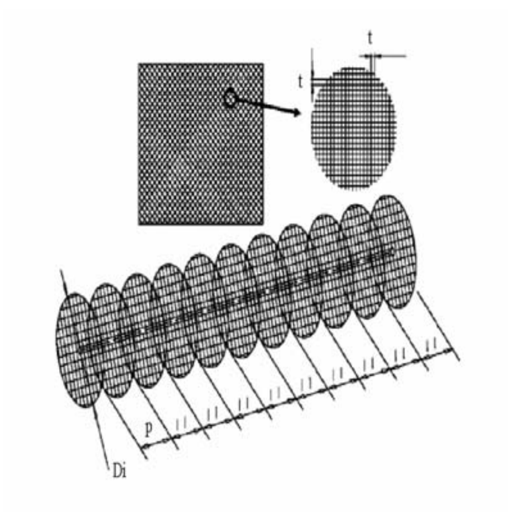

Figure-2. Porous medium manufactured from copper screens.

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).

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

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

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.

Figure-3. Photographic view of mesh inserts.

EXPERIMENTAL UNCERTAINTY

Table-2 and 3 shows the mesh inserts characteristics and uncertainty in the measurements of instruments respectively.

The Nusselt number obtained from experimental work is compared with the value obtained using Dittus-Boelter equation (theoretical) (Figure-4). The experimental uncertainty is found as 10% for Nusselt number.

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Table-2. Mesh inserts characteristics.

Mesh insert No.	D_i (mm)	P (mm)	R_p	ε (%)
1	22	50	0.8	99.7298
2	22	100	0.8	99.8649
3	22	150	0.8	99.9090
4	22	200	0.8	99.9324
5	18	50	0.65	99.8191
6	18	100	0.65	99.9095
7	18	150	0.65	99.9397
8	18	200	0.65	99.9548
9	14	50	0.5	99.8906
10	14	100	0.5	99.9453
11	14	150	0.5	99.9635
12	14	200	0.5	99.9726
13	10	50	0.364	99.9442
14	10	100	0.364	99.9721
15	10	150	0.364	99.9814
16	10	200	0.364	99.9861

Table-3. Accuracy and uncertainty of measurements.

Instruments	Accuracy (%)	Uncertainty
Rota meter (water flow rate, kg/s)	0.2	± 0.01
Thermocouple type, J type (°C)	± 0.4	± 0.10

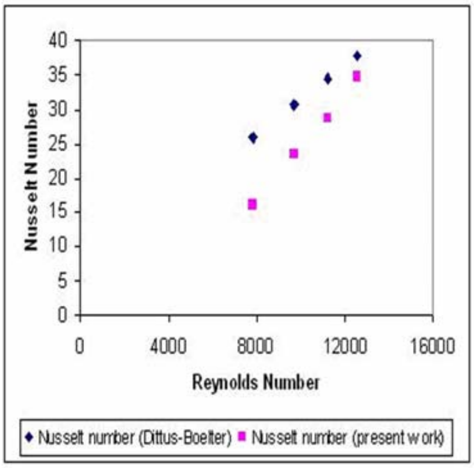

Figure-4. Comparison of Nusselt number (Dittus-Boelter and present work) for a plain tube.

HEAT TRANSFER CALCULATIONS

The average heat flux from the tube wall to the fluid is defined in terms of surface area.

$$Q = h\,A(Ts - Tb) + \sigma A \varepsilon_n (Ts^4 - T_b^4) \quad (1)$$

Where $Q = m_a * C_p * (Tout - Tin)$

It took approximately 45minutes to obtain a steady state for each run.
Heat transfer coefficient h is calculated using equation (1)

$$Nu = \frac{h\,D_H}{K} \quad (2)$$

The local values for Nu, Pr and Re were calculated on the basis of air properties corresponding to bulk fluid temperature. Experimental data obtained for Nusselt number in fully developed axial flow is compared with the correlations from the literature.
Nusselt numbers calculated from the experimental data for plain tube were compared with the correlation recommended by Dittus-Boelter

$$Nu = 0.023\,Re^{0.8}\,Pr^{0.4} \quad (3)$$

F, friction factor for smooth tube is given as

$$F = (1.82 \log_{10} Re - 1.64)^{-2} \quad (4)$$

EXPERIMENTAL RESULTS AND DISCUSSIONS

Experimentally determined Nusselt number values for plain tube (with out mesh insert) are compared with Dittus-Boelter correlation. It is seen that the experimental results are in good agreement with aforementioned studies. The same experiments were repeated with mesh inserts of varying diameters 22mm, 18mm, 14 mm and 10mm. Figure-5 shows the variation of temperatures recorded in a (a) plain tube, (b) Mesh insert 2, (c) Mesh insert 5, (d) Mesh insert 9 and (e) Mesh insert 13.

A careful examination of cases in Figure-5 (a), (b), (c), (d) and (e) when a mesh insert is placed in the test section compared to the plain tube case shows that as the mesh diameter (R_p) decreases, turbulence created in the tube decreases thereby causing an increase in surface temperature. As Reynold's number increases, higher heat transfer rates are observed regardless of the value of porosity (ε).

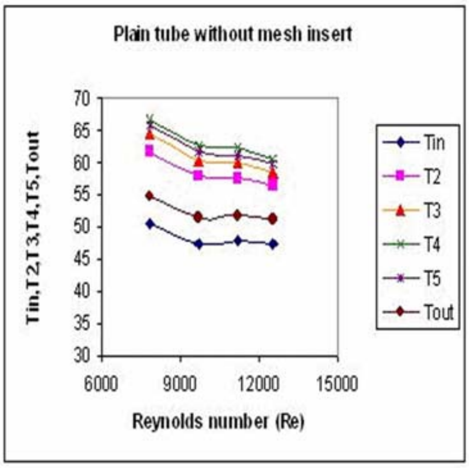

Figure-5 (a)

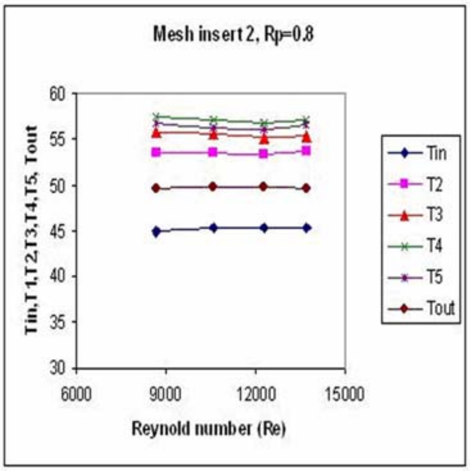

Figure-5 (b)

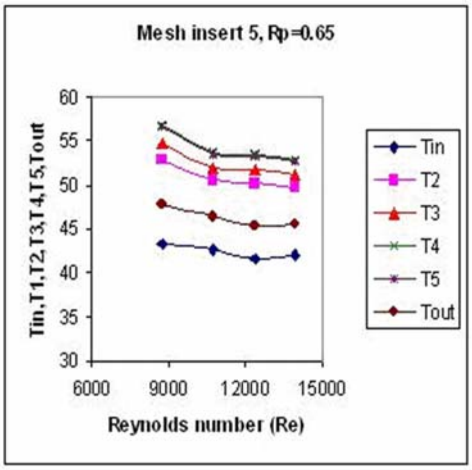

Figure-5 (c)

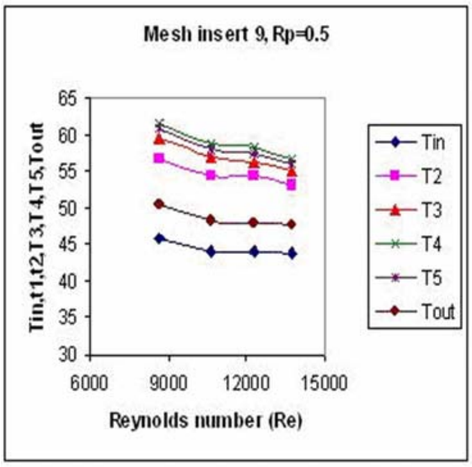

Figure-5 (d)

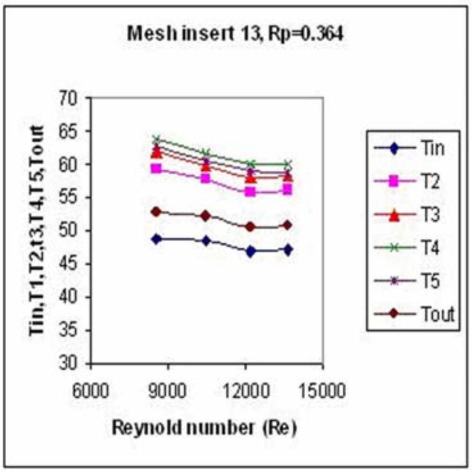

Figure-5 (e)

Figure-5. Temperatures recorded (a) plain tube (b) Mesh insert 2 (c) Mesh insert 5 (d) Mesh insert 9 (e) Mesh insert 13.

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Figure-6 shows the length averaged Nusselt number for plain tube and for mesh inserts. It is observed that R_p = 0.8 (22mm mesh diameter) yields the highest value of Nusselt number. This may be due to more obstruction caused to airflow leading to more turbulence. The enhancement of heat transfer by employing mesh inserts is to be balanced by the penalty arising from the increased pressure drop.

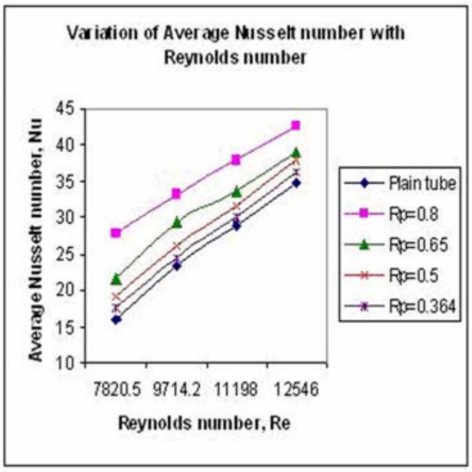

Figure-6. Variation of Average Nusselt number with Reynolds number for mesh inserts (Rp= 0.8, 0.65, 0.5 and 0.364).

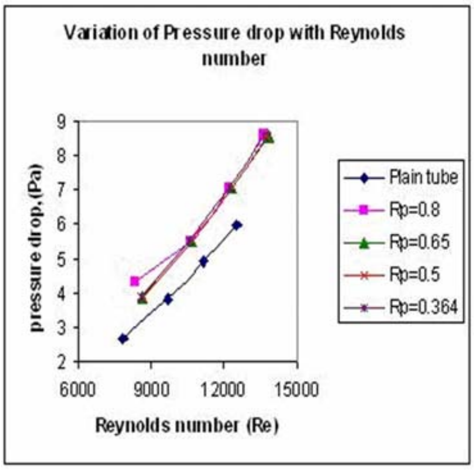

Figure-7 (a). Variation of pressure drop with Reynolds number for mesh inserts (Rp= 0.8, 0.65, 0.5 and 0.364).

Figure-7 (a) presents the pressure drops for different mesh inserts at different Re. As expected, largest pressure drop occurs to the case that offers the highest thermal performance i.e. R_p = 0.8.

Figure-7 (b) shows the average Nusselt number vs. Reynolds number and Figure-7(c) shows the average Nusselt number vs. porosity.

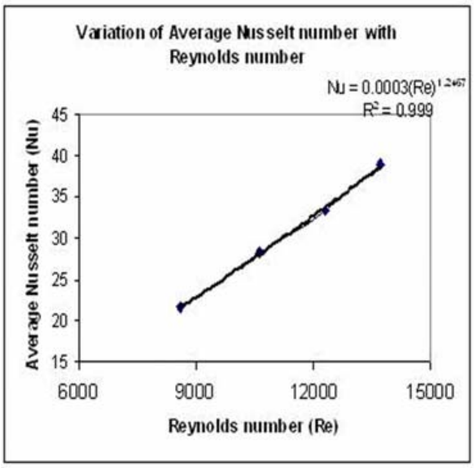

Figure-7 (b).Variation of average Nusselt number with Reynolds number.

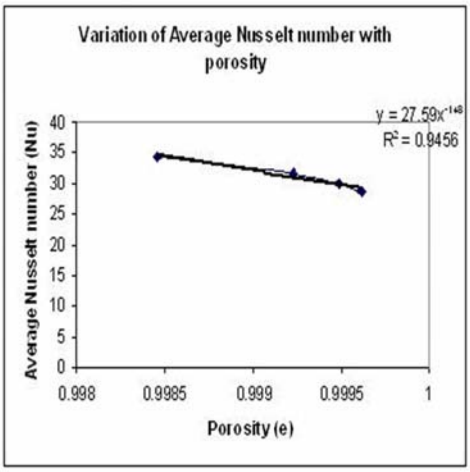

Figure-7 (c). Variation of Average Nusselt number with porosity.

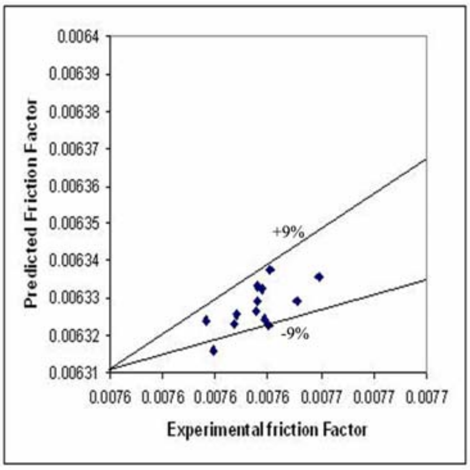

Figure- 8. Comparison between predicted and experimental friction factor.

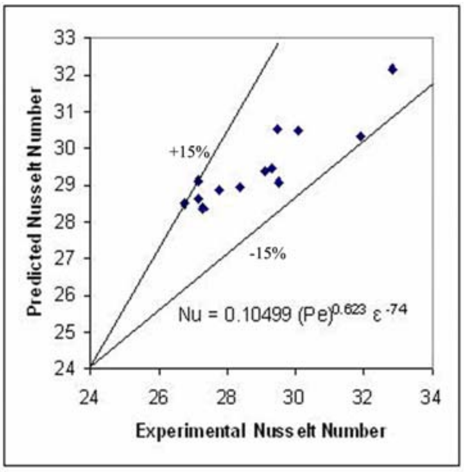

Figure-9. Comparison between predicted and experimental Nusselt number.

The Nusselt number and friction factor obtained from the present experiments are compared with those calculated from the proposed correlations as shown in Figure-8 and 9 respectively. It is observed that the majority of the data falls within ±15% for Nusselt number and ± 9% for friction factor.

CONCLUSIONS

The study presents an experimental investigation of the potential of mesh inserts to enhance the rate of heat transfer occurring between the surface of a pipe heated with a constant and uniform heat flux and the air flowing inside. The following conclusions can be drawn:

- For a constant diameter, further enhancement in heat transfer can be attained by using a porous insert with smaller porosity.
- The maximum increase in Nusselt number of approximately 2 times was obtained by fully filling the pipe at the expense of highest pressure drop of 8.6 Pa.

Correlations for Nusselt number and friction factor are developed for the mesh inserts in terms of Peclet number (Pe) and porosity (ε),

$$Nu = 0.10499\,(Pe)^{0.623}\,\varepsilon^{-74}$$
$$f = 0.02606\,Re^{-0.13215}\,\varepsilon^{-2.0128}$$

The correlations are comparable with the results obtained by Pavel and Mohamad (Bogdan I. Pavel *et al.*, 2004).

The increase in pressure drop by increasing R_p values is not as significant as the increase in Nusselt number with R_p. The optimum value obtained for $R_p = 0.8$ and p = 50mm. The Nusselt number is 1.86 times the value of that of plain tube and the maximum pressure drop obtained is 8.6 Pa.

Nomenclature

D_i	diameter of mesh insert, (m)
D_H	hydraulic diameter (4A/P), m
f	friction factor
Nu	Nusselt number, ($h.D_h/k$)
h	Heat transfer coefficient for air(W/m^2 K)
P	distance between two screens, (m)
K	Thermal conductivity of air(W/m K)
A	Surface area of test section (m^2)
Δp	pressure drop (N/m^2)
Pe	Pecelt Number
Pr	Prandtl Number
Q	heat gained by air (W)
Re	Reynolds number, ($u.D_H\rho/\mu$)
Pr	Prandtl number
R_p	ratio of porous material ($2r_p/D_p$)
C_p	Specific heat of air(J/kg K)
Tin, Tout	air temperature at inlet and outlet, respectively, (°C)
T2, T3, T4, T5	tube wall temperatures, (°C)
T_s	average surface temperature of the working fluid, (°C)
T_b	bulk temperature, (°C)
ε_N	emissivity of tube material (copper)
ε	porosity
T	thickness of the wire mesh, mm
m_a	mass of air, kg/sec
σ	Stefan-Boltzmann's constant

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