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RESEARCH ARTICLE

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EXPERIMENTAL ANALYSIS OF TURBULENT FLOW HEAT TRANSFER ENHANCEMENT IN AHORIZONTAL CIRCULAR TUBE USING MESH TYPE VORTEX GENERATORS

K.L.SATYAVARMA[1],T.LAKSHMANAKISHORE[2].

[1](PG) Student (M.Tech), Department of Mechanical Engineering, JNTUK-UCEV, vizianagaram, A.P., India.

[2]Assistant professor,Department of Mechanical Engineering,JNTUK-UCEV, vizianagaram, A.P., India

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K.L.SATYAVARMA

ABSTRACT

This work aims at developing a passive vortex generator which would improve the thermal performance of the system. The experimental work is carried outin order to study the heat transfer enhancement in horizontal circular pipe, using a specially designed vortex generators. A vortex generator is like a passive heat transfer enhancement tool having special geometry swirl flow inside the pipe. 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. The results obtained from the setup without mesh inserts is validated with the analytical results. The co-relation between Heat Transfer Coefficient with Mesh diameter, Hydraulic diameter and pitch is derived with experimental and theoretical results.

Keywords:vortex generatormesh diameter, hydraulic diameter, pitch, swirl flow, orifice velocity, air velocity.

INTRODUCTION

In 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.

Literature Review

There has been a considerable amount of research in the area of heat transfer enhancement available both in the form of experimental results and as predictions of the numerical investigation. An overview of the field of heat transfer enhancement is presented in the firstsection.The second section focuses on those investigations which have been performed to evaluate the utility of the vortex generators in augmentation of heat transfer in compact heat exchangers.

Vortex Generation in Fin-Tube Heat Exchangers

Numerically studied heat transfer characteristics [1] and fluid flow structure of fin-and-oval-tube heat exchangers with longitudinal vortex generators (LVGs). For Re(based on the hydraulic diameter) ranging from 500 to 2500, the average Nusselt number for the three-row, fin-and-oval-tube heat exchanger with longitudinal vortex generators increased by 13.6-32.9% over the baseline case and the corresponding pressure loss increased by 29.2-40.6%. The results were analyzed on the basis of the field synergy principle to provide fundamental understanding of the relation between the local flow structure and heat transfer augmentation.[2] It was confirmed that the reduction of the intersection angle between the velocity field and the temperature field was one of the essential factors influencing the heat transfer enhancement. Three geometrical parameters i.e., the placement of LVGs (upstream and downstream), angles of attack (β=15°, 30°, 45° and 60°) and tube-row number (n=2, 3, 4 and 5) were investigated. It was found that the LVG's with placement of downstream, angles of attack β=30°and minimum tube-row number provided the best heat transfer performance.

Mechanism of Heat Transfer Enhancement

Principle and Method

In the tubes ide of a heat exchanger[3], in creasing fluid velocity willead to highershear stress, intensifying fluid disturbance in the boundary flow will result in more dissipation of fluid momentum, and enlarging continuously extended surface will cause more frictional resistance and viscosity dissipation.[4]The flow resistance will be increased by adopting these techniques. If the flow resistance is over large, the fluidvelocity will be come small, which may weaken convective heat transfer between the fluid and the surface. Therefore, reducing flow resistance will be one of the key factors of developing heat transfer enhancement techniques with lower power dissipation.

Principle of Heat Transfer Enhancement in the Core Flow of a Tube:[5] In order to achieve the effect of heat transfer enhancement, amethod that canbe taken into account is inserting heat transfer component in the core flow of a tube. The basic consideration of heat transfer enhancement in this case is to uniform fluid temperature in the core flow and create an equivalent thermal boundary layer with relatively larger temperature gradient near the wall, which helps to enhance convective heat transfer between the fluid and the tubewall. Mean while, the flow resistance will not increase remarkably. The forming of an equivalent thermal boundary layer near the tube wall couldbe in dependent of the velocity boundary layer. [6]For the purpose of reducing flow resistance, the boundary layer with relatively larger velocity gradient should be avoided. Therefore, for improving the overall heat transfer performance or raise the PEC value, the increase in heat transfer may surpass the increase in flow resistance after takingpropermeasurein the coreflow region.To meet this requirement [7] proposed aprinciple ofheat transfer enhancement in the coreflow, which is mainly expressed as [1] strengthening temperature uniformity inthe coreflow; [2]increasing fluid disturbance in the coreflow; [3] reducing surface area of heat transfer component inthe coreflow; [4] decreasing fluid disturbance in the boundary flow. Based on the principle, when developing a technique of heat transfer enhancement in the core flow of atube, we should avoid strongly disturbing thef luid near the wall. In addition, the contact between the heat transfer component and the tubewall should be prevented, and the function of this component with out contacting heat sourceor heat sinkis just to disturb the fluid or uniform the fluid temperature. As the component for heat transfer enhancement in the core flow does not conduct heat from the tubewall, no convective heat transfer occurs at any point between the component surface and the fluid. [8] we can define that surface-based heat transfer enhancement isonetypethatconvective heat transfer occurs between thewallsurface and thefluid;[2] fluid-based heat transfer enhancement is another type that there is no convective heat transfer between the component surface and the fluid, which is so-called heat transfer enhancement in the coreflow.

3 EXPERIMENTAL WORK

3.1Experimental set up:The apparatus consists of a blower unit fitted with a pipe which is connected to the test section located in horizontal orientation. Soldered filament is used as a heating element. Length of the test section is 40cm. Four thermocouples T_1, T_2, T_3 and T_4 at a distance of 5cm, 15cm, 25cm, 35cm from the origin of heating zone are embedded in the walls of the tube and two thermocouples are placed in the air stream, one at the entrance (T_6) and the other at the exit (T_5) of the test section to measure the temperature of flowing air as shown in fig 1.

The test pipe is connected with an orifice to measure the flow of air through the pipe. Input the heater is given through dimmer stat. The velocity of air flow in the tube is measured with the help of orifice plate and the mercury manometer fitted to the board. The inner tube of the heating part which is the test tube with inside diameter 28 mm is made of 2 mm thick copper plate. A heat generating element is wound around this test tubes 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 0-220V, with a maximum current of 10A.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.

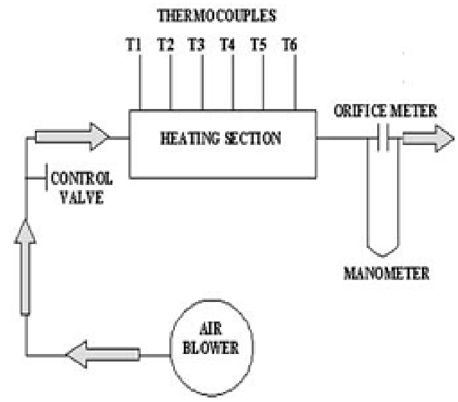

Fig1:line diagram of our experiment

Fig2:The porous media used for the experiments are Iron screens cut out at various diameters (Di) and then inserted on iron rod shown in Figure-.3.

That is, 16 different inserts were obtained by varying the screen diameter and the distance between two adjacent screens (p). Due the 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.

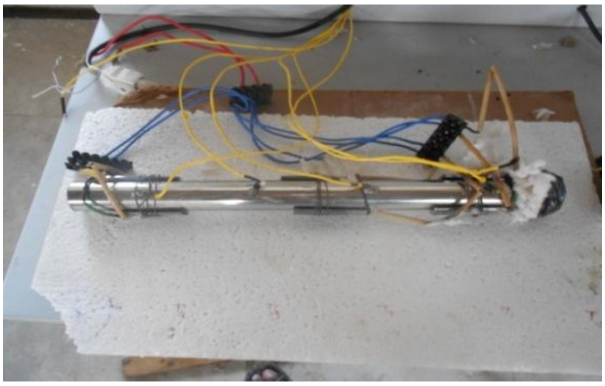

Fig3: photographic view of Experimental set up

3.2 PROCEDURE

Supply is given to the blower motor and the valve is opened slightly. A heat input of 100 W is given to the soldered filament binded on the test section by adjusting the dimmer stat. Thermocouples 1 to 4 are fixed on the test surface and thermocouples 5 to 6 are fixed at both ends of the pipe. The readings of the thermocouples are observed every 5 minutes until they show constant values. Under steady state condition, the readings of all the six thermocouples are recorded. The fluid properties were calculated as the average between the inlet and the outlet bulk temperature. It took 90 minutes to reach steady state conditions. Experiment was carried out at constant heat flux conditions and constant heat input of 100 W at different mass flow rates, with and without mesh inserts.

Sequence of Operations: Experiments carried out first without inserts and then with inserts. Without mesh inserts:Initially the experiment is carried out without any insert. The working fluid airflows through the pipe section with least resistance.

With mesh inserts: The porous media used for the experiments are iron screens as shown in Fig. 3. (Wire diameter nn0.28 mm) cut out at various diameters (Di) and then inserted on iron rods. That is, 16 different inserts were obtained by varying the screen diameter and the distance between two adjacent screens (p). Photograph of one mesh insert is shown in Fig 4. Thermocouple readings from T1 to T6 are taken for all the mesh inserts shown in Table 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. The mass flow rates of air and the heat input are same as that of plain tubes experiment. The experiment with mesh inserts is carried out in four steps. All the steps are explained below.

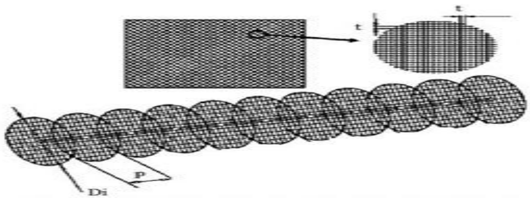

Fig4: Porous medium manufactured from iron screens

Fig5: photographic view of porous medium manufactured from iron screens. Mesh insert diameters along with different pitches

Table1 Experimental values of Heat transfer coefficient and Nusselt number without mesh inserts

Sl.Number	Heat transfer coefficient (h)W/ (m^2K)	Nusselt number for plain tube Nu (hD/k)
1	87.44	88.387

Table2 Theoretical values of Heat transfer coefficient and Nusselt number without mesh inserts

Sl.Number	Heat transfer coefficient (h) W/ (m^2K)	Nusselt number for plain tube Nu (hD/k)
1	108.86	110.86

Table 3 Mesh insert diameters along with different pitches

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

Step 1

In the first step of insertion of meshes, mesh with a diameter 22mm is inserted inside the horizontal tube. Now the experiment is carried out in the same way as that of plain tube. By varying the pitch of the meshes and keeping the diameter constant the experiment is repeated four times .When the meshes of various diameters are inserted into the horizontal tube of test section diameter 28mm, there occurs the turbulence in the flow. As a result the heat transfer coefficient increases when compared to the heat transfer coefficient which occurs in the case of flow through plain tube.Step 2in this step the mesh of diameter 22mm is replaced a diameter of 18mm and inserted into the horizontal tube of test section diameter 28mm. By keeping the diameter constant and varying the pitch of the mesh the experiment is carried out. Temperature values are taken and then experimental and theoretical values are calculated.

Step 3in this step the mesh of diameter 18mm is replaced a diameter of 14mm and inserted into the horizontal tube of test section diameter 28mm. By keeping the diameter constant and varying the pitch of the mesh the experiment is carried out. Temperature values are taken and then experimental and theoretical values are calculated.

Step 4in this step the mesh of diameter 14mm is replaced a diameter of 10mm and inserted into the horizontal tube of test section diameter 28mm. By keeping the diameter constant and varying the pitch of the mesh the experiment is carried out. Temperature values are taken and then experimental and theoretical values are calculated.

Table 4: Experimental values of heat transfer coefficient and Nusselt number with mesh inserts

Sl.Number	Heat transfer coefficient (h) W/ (m^2K),	Nusselt number for plain tube Nu (hD/k)
1	155.09	91.33

Table 5: Theoretical values of heat transfer coefficient and Nusselt number with mesh inserts

Sl.Number	Heat transfer coefficient (h) W/ (m^2K),	Nusselt number for plain tube Nu (hD/k)
1	253.42	149.23

When the experiment is carried out using mesh inserts hydraulic diameter is used instead of mesh diameter in heat transfer calculations. The experimental and Theoretical heat transfer coefficient values are shown below in the table.

4. Heat Transfer Calculations

$T_s = (T_1+T_2+T_3+T_4)/4$ (1)

$T_b = (T_5+T_6)/2$ (2)

Discharge of air, (3)

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

Velocity of air,

$V_a = (Vo * Ao)/A$ (4)

Reynolds number, Re=U D/υ (5)

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

Wetted perimeter= **∏.** (D_i+D) (6)

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

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

h = Q/ (A ($T_s - T_b$) (9)

Nu = hD/K (10)

Eq. (10) gives experimental Nusselt number. Nusselt numbers calculated from the experimental data for plain tube were compared with the correlation recommended by Dittus-Boelter.

Nu = 0.023 $Re^{0.8}Pr^{0.4}$ (11)

Eq. (11) gives theoretical Nusselt number.

RESULTS AND DISCUSSIONS

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

Fig.1 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.

Were r_p is the porous ratio (Di/D).

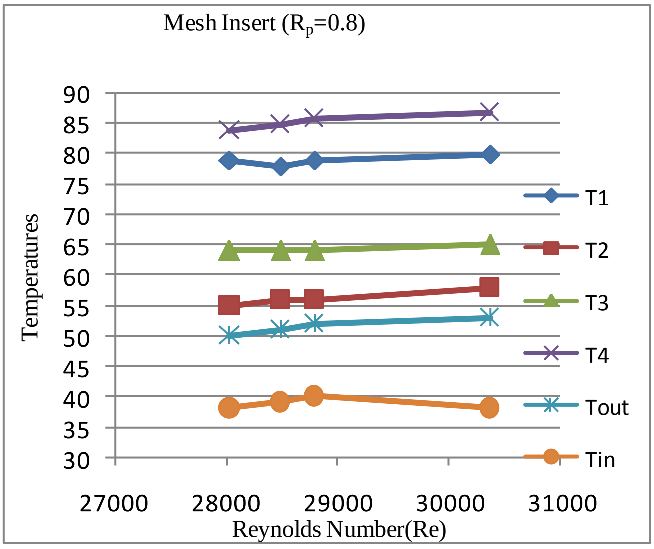

Fig.1 The variation of surface temperature with Reynolds number for mesh insert of diameter 22mm.

Fig.2 shows 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.

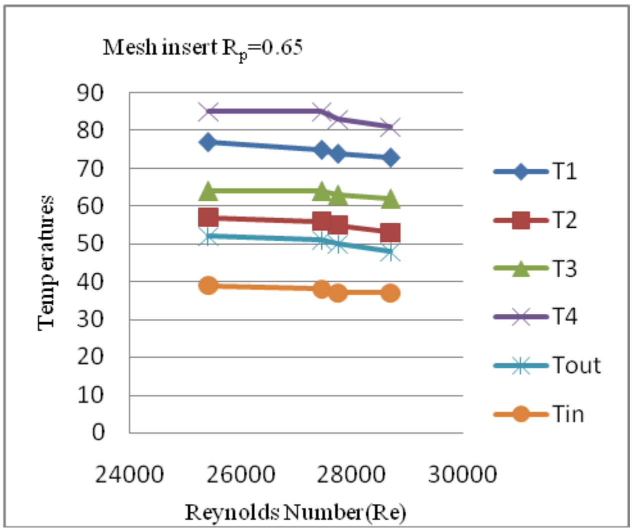

Fig .2: Variation of surface temperature with Reynolds number for meshinsertof18mmdiameter.

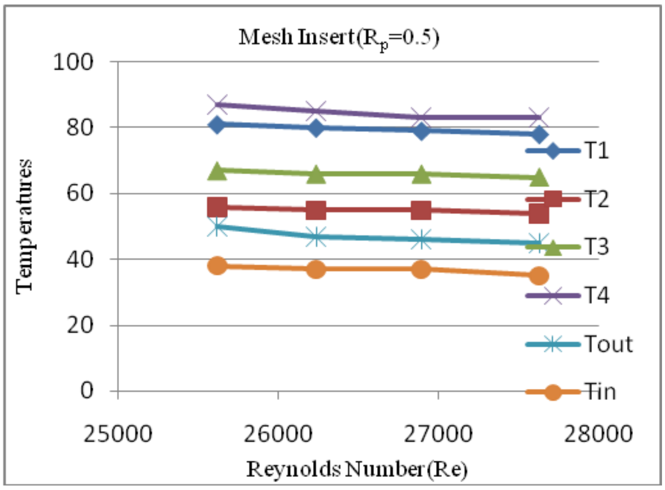

Fig.3: the variation of surface temperature with Reynolds number for the mesh insert of diameter

Fig.3 shows the variation of surface temperature with Reynolds number for the mesh insert of diameter 14mm. As the diameter of mesh insert reduced from 18mm to 14mm and then inserted into the horizontal tube of test section diameter 28mm, turbulence which is created in the tube reduced there by surface temperature also reduced. Were r_p is porous ratio

Fig.4 shows the variation of surface temperature with Reynolds number for the mesh insert of diameter 10mm. As the diameter of mesh insert is further reduced from 14mm to 10mm and then inserted into the horizontal tube of test section diameter 28mm the turbulence which is created in the tube reduced there by the surface temperatures also reduced.r_p is porous ratio.

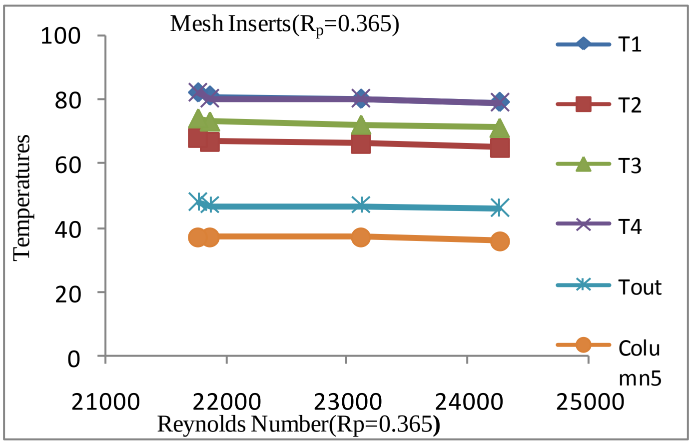

Fig .4: Variation of surface temperature with Reynolds number for mesh insert of 10mm diameter

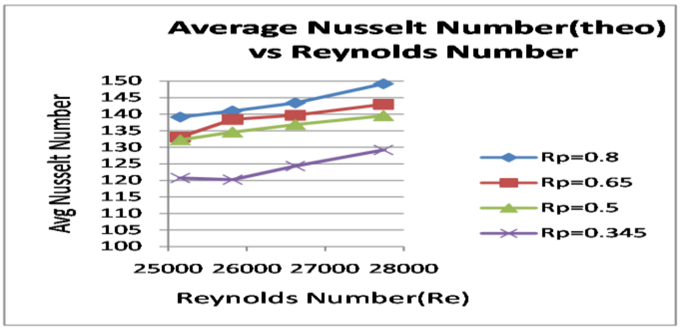

Fig .5: Variation of Experimental Average Nusselt number with Reynoldsnumber

Fig.5 shows the experimental variation of average Nusselt number with Reynolds number for various sizes of mesh inserts. Here it is observed that the mesh insert of diameter 22mm yields the highest value of Nusselt number. This is due to high turbulence which is created inside the horizontal tube with 28mm test section diameter when this mesh of diameter 22mm is inserted.

Fig.6 shows the Theoretical variation of average Nusselt number with Reynolds number for various sizes of mesh insert. Here also it is observed that the mesh insert of diameter 22mm yields the highest value of Nusselt number. But the variation of average Nusselt number with Reynolds number is high when compared to experimental variation.were rp is porous ratio.

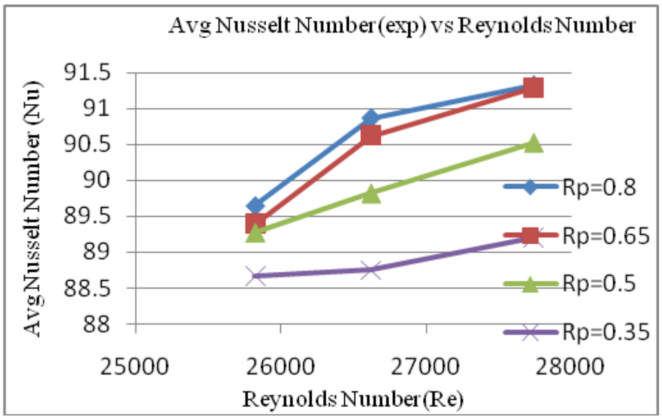

Fig .6: Variation of Theoretical Average Nusselt number with Reynolds number

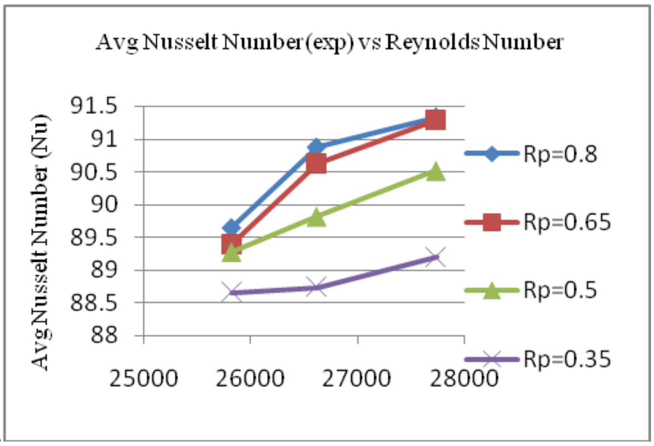

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Fig.7 The variation of theoretical heat transfer coefficient with mesh diameter for different values of pitch. Reynolds number

Fig.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.

Fig.8: Variation of experimental heat transfer coefficient with mesh diameter for different pitch values. Were rp is porous ratio.

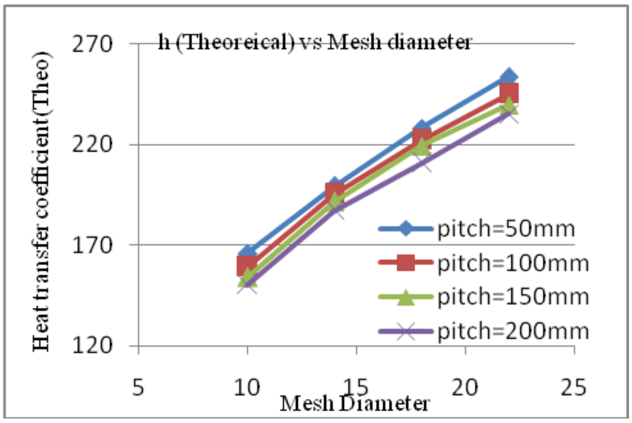

Fig8: Variation of theoretical heat transfer coefficient with mesh diameter for different pitch values

Fig.9 shows the variation of experimental heat transfer coefficient with pitch for various mesh diameters. 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.

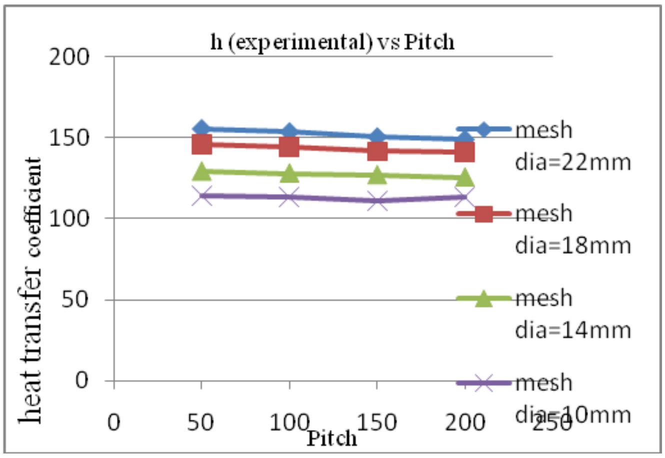

Fig.9: Variation of experimental heat transfer coefficient with pitch for different mesh inserts.

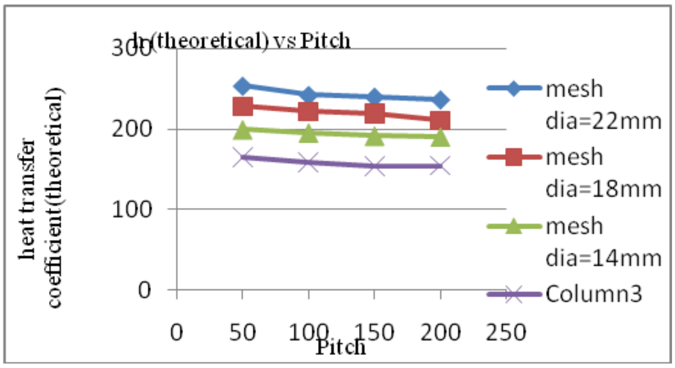

Fig.10 the variation of theoretical heat transfer coefficient with pitch for different values of mesh diameters.

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.

Fig.11 shows the variation of experimental Nusselt number 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 Nusselt number value when compared to the rest of mesh inserts.were r_p is porous ratio.

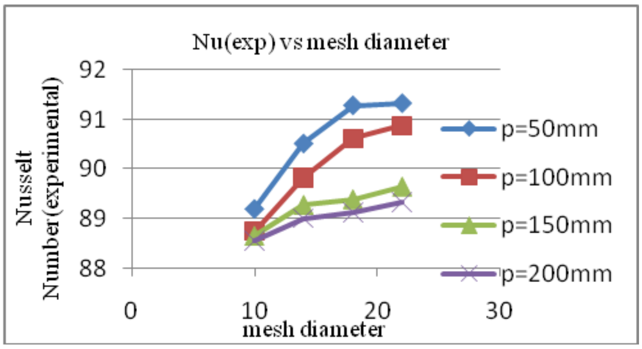

Fig.11: Variation of experimental Nusselt number mesh diameter for different pitch values

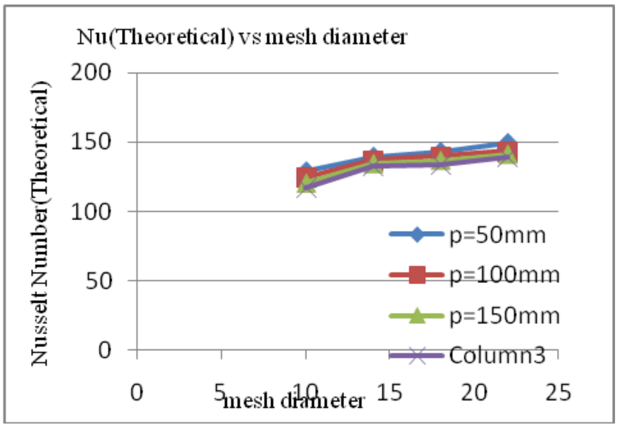

Fig.12: Variation of theoretical Nusselt number with mesh diameter for different pitch values

Fig.13 shows the variation of experimental Nusselt 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 experimental Nusselt number value when compared to the rest of mesh inserts. Were r_p is porous ratio.

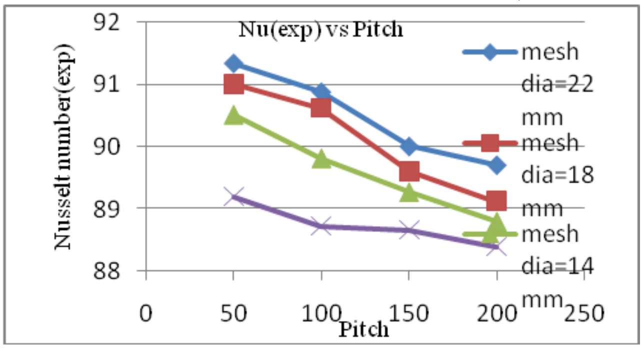

Fig .13:Variation of experimental Nusselt number with pitch for different mesh inserts

Fig.14 shows the variation of theoretical Nusselt 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 theoretical heat Nusselt number value when compared to the rest of mesh inserts.

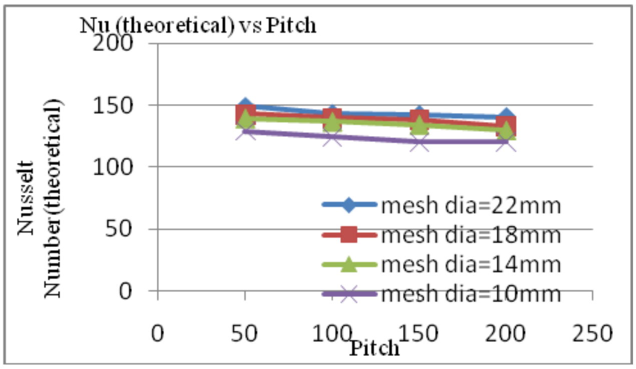

Fig .14: Variation of theoretical Nusselt number with pitch fordifferent mesh inserts

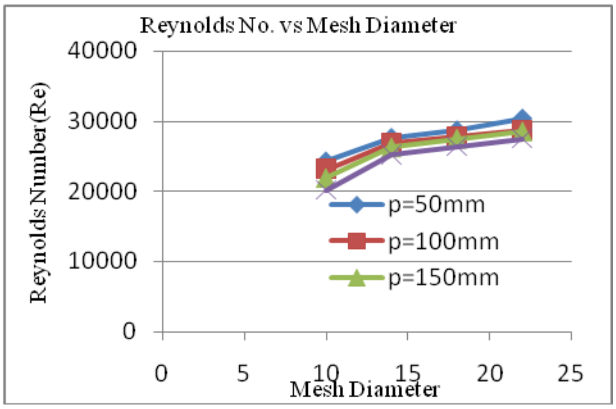

Fig.15 shows the variation of Reynolds number with mesh diameter for different values of pitch. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest Reynolds number value when compared to

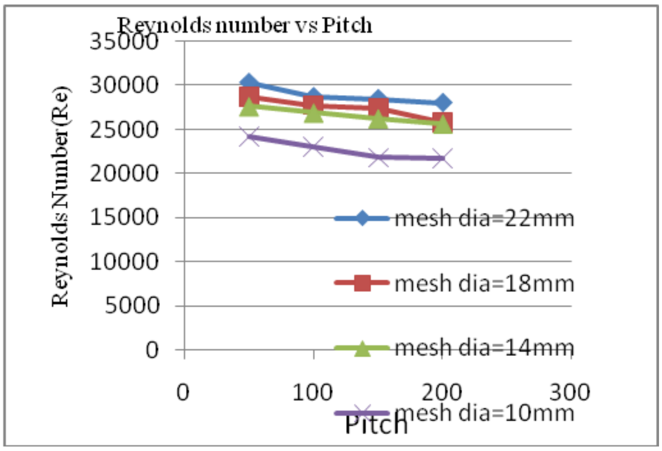

Fig .16: Variation of Reynolds number with pitch for various mesh inserts. Shows the variation of Reynolds number with pitch for different mesh insertsfrom the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest Reynolds number when compared to the rest of mesh inserts. Were r_p is porous ratio.

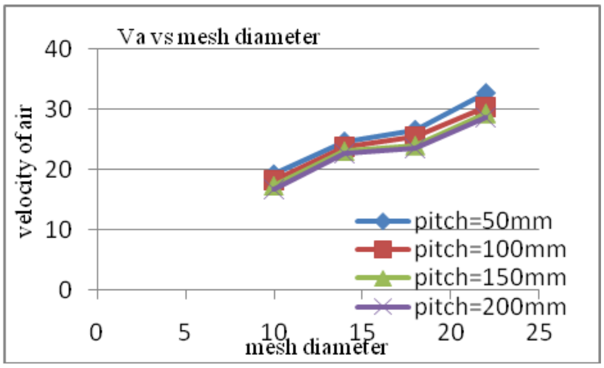

Fig .17: Variation of Velocity of air with Mesh diameter for different pitch values. Shows the variation of Velocity of air with mesh diameter for different pitch values. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest Velocity of air value when compared to the rest of mesh inserts.

Fig.18 shows the variation of Velocity of air with mesh pitch for different mesh inserts. From the figure it is observed that the mesh diameter of 22mm with pitch of 50mm yields the highest Velocity of air value when compared to the rest of mesh inserts.

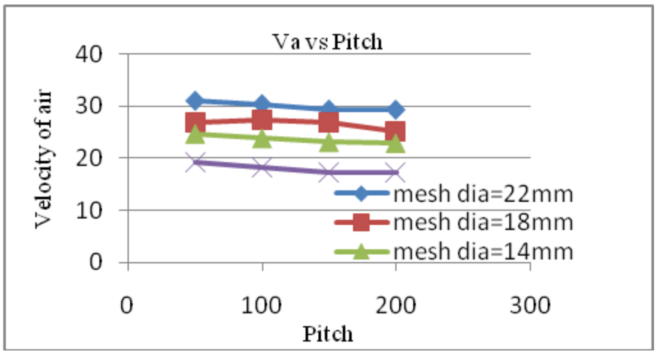

Fig.19 shows the variation of experimental heat transfer coefficient with hydraulic diameter. Heat transfer coefficient is inversely proportional to hydraulic diameter. As pitch and hydraulic diameter increases the value of heat transfer coefficient decreases. From the figure 16.6mm Hydraulic diameter gives highest value of experimental heat transfer coefficient i.e 160.09w/m2

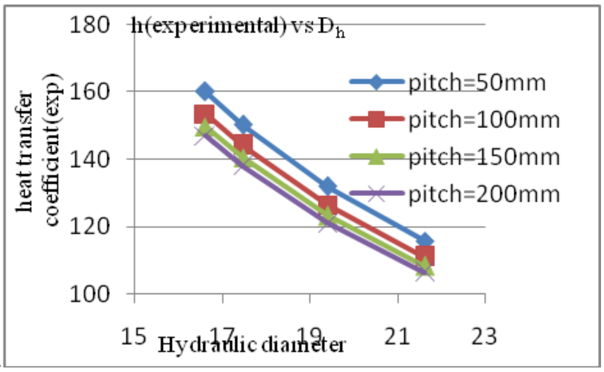

Fig.20 shows the variation of Theoretical heat transfer coefficient with hydraulic diameter. Heat transfer coefficient is inversely proportional to hydraulic diameter. As pitch and hydraulic diameter increases the value of heat transfer coefficient decreases. From the figure 16.6mm Hydraulic diameter gives highest value ofheattransfer coefficient i.e 251.98W/m^2k

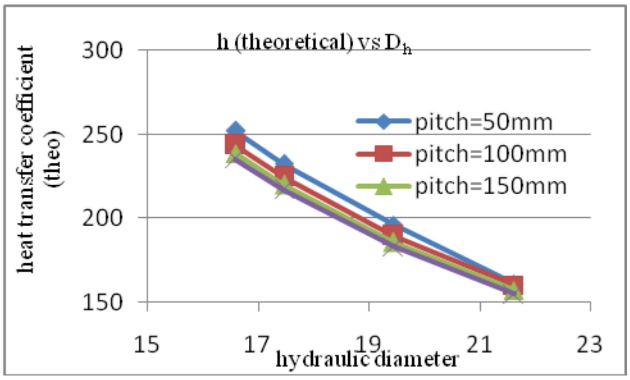

Fig.20: Variation of theoretical heat transfer coefficient with hydraulic diameter

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.

$h = 6274.8\ (d_h)^{-1.221}\ (p)^{-0.0609}$

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.

$h = 26173.39\ (d_h)^{-1.5839}\ (p)^{-0.04942}$

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

Pitch.

When we consider Mesh diameter the relation with heat transfer coefficient is directly proportional to mesh diameter and is inversely proportional to pitch. I.e as mesh diameter increases the heat transfer coefficient also increases. As pitch increases the heat transfer coefficient value decreases.

When we consider Hydraulic diameter the relation with heat transfer coefficient is inversely proportional to mesh diameter and pitch. I.e as hydraulic diameter value increases the heat transfer coefficient is decreases. As the value of pitch increases the heat transfer coefficient value will be decreases.

As mesh diameter increases Reynolds number and velocity of air increases. I.e the mesh diameter is directly proportional to the Reynolds number and as well as velocity of air increases.

As hydraulic diameter decreases Reynolds number and velocity of air increases. I.e the hydraulic diameter is inversely proportional to Reynolds number and Velocity of air.

As pitch increases Reynolds number and velocity of air decreases. The pitch is inversely proportional to Reynolds number and Velocity of air. The variation of pitch with Reynolds number and Velocity of air is same in the cases of with and without mesh inserts.

Effective heat transfer coefficient value is obtained at 22mm mesh diameter with 50mm pitch in the case of mesh diameters.

Effective heat transfer coefficient value is obtained at 16.6mm hydraulic diameter with 50mm pitch value.

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