Proceedings of the
International Conference on Mechanical Engineering 2007
(ICME2007) 29- 31 December 2007, Dhaka, Bangladesh

ICME07-TH-05

HEAT TRANSFER IN TURBULENT FLOW THROUGH TUBE WITH PERFORATED TWISTED TAPE INSERT

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ABSTRACT
Enhancement of heat transfer in tube flow has practical applications. Inserts of various geometry in a circular tube enhance heat transfer. An experimental investigation has been carried out for turbulent flow in a circular tube with perforated twisted tape insert. In this work, mild steel twisted tape inserts with holes of different diameters termed as perforated twisted tapes are used. Heat transfer and pressure drop characteristics are examined in the tubes for the turbulent flow. The flow of air through tubes is varied thereby varying the Reynolds number. Heat transfer and pressure drop data are generated for Reynolds numbers ranging from 1.3×10^4 to 5.2×10^4. Air velocity, its temperatures, tube wall temperatures, pressure drops are measured for smooth/plain tube. The same variables are measured after inserting twisted tapes in the tube. Heat transfer coefficients, Nusselt numbers, pumping power and heat transfer effectiveness are calculated using measured variables for the smooth/plain tube, as well as for tube with the inserts. Heat transfer rate is found to increase with corresponding increase in friction factor and pumping power for tube with the twisted tape inserts. For the same Reynolds number, the heat transfer coefficient in the tube with the twisted tape inserts is found to increase upto 5.5 times of that for the plain tube. Whereas, the pumping power in the tube with the twisted tape insert is found to increase upto 1.8 times of that for the plain tube. It is also found that the heat transfer effectiveness in a tube with the twisted tape insert is found to increase upto 4.0 times compared to the value for the plain tube.

1. INTRODUCTION
Energy and material saving considerations as well as economic incentives have led to comprehensive efforts to design high performance heat exchanger. The efficiency and economic competitiveness of industrial processes depend, to a great extent, on the performance of heat exchangers. The performance of heat exchanger can be improved by using special surface geometry such as finned surfaces, integrated rough surfaces and various types of inserts. Improvement in performance of the heat exchanger may result reduction of its size and subsequently its costs. For a heat exchanger of fixed size, improvement in performance may increase the heat transfer rate or decrease the temperature difference between the process fluids allowing for more efficient utilization of thermodynamic availability. In practice, an enhanced surface must provide the desired heat transfer rate and meet the required flow rate and pressure drop constraints.

Sieder and Tate [1] derived an empirical correlation for turbulent flow of forced convection through circular as well as non-circular tubes. The correlation shows that higher heat transfer occurs in finned tubes than plain tube.

Date [2] developed a numerical model for predictions of heat transfer for twisted tape insert in fully developed laminar flow at constant temperature with constant properties. His analysis for loosely fitting tape with helix angle 34.9 degree and Prandtl number, Pr=1 shows that the Nusselt number increases with increasing Reynolds number, contrary to laminar flow in plain tubes, which is independent of Re and Pr.

Friction and Nusselt number data were generated and semi-empirically evaluated by Gupte and Date [3] for twisted tape generated helical flow in annuli. Results have been obtained for radius ratios of 0.41 and 0.61 and twist ratios of ∞, 5.302, 5.038, and 2.059. The experimental results show that for the same twist ratio (y), the increase in pressure drop exceeds the increase in heat transfer irrespective of the radius ratios (r*). For y =5.302, r* = 0.41 and at the same Reynolds number based on hydraulic diameter, the increase in pressure drop and heat transfer coefficient over an empty annulus are 90 percent and 60 percent respectively. At lower twist ratios both the percentages are even greater. Again the semi-analytical expressions developed predict the Nusselt numbers with good agreement for y = ∞ and y = 5 .302 and 5.038.

Agarwal and Raja Rao [4] experimentally determined isothermal and non-isothermal friction factors and mean

Nusselt numbers for uniform wall temperature during heating and cooling of Servotherm oil (Pr=195-375) in a circular tube (Re=70-4000) with twisted tape inserts (twist ratio, y=2.41-4.84). Isothermal friction factors were found to be 3.13-9.71 times higher than that of the plain tubes. The Nusselt numbers were found to be 2.28-5.35 and 1.21-3.70 times higher than that of the plain tubes in forced convection based on constant flow rate and constant pumping power, respectively, for the minimum twist ratio.

Uddin [5] studied pressure drop characteristics and heat transfer performance of air through an internal rectangular finned tube. He observed that the heat transfer coefficient based on inside diameter and nominal area was in the range of 1.5 to 1.75 times higher than that of the plain tube. When compared with a plain tube at constant pumping power an improvement of 4% was obtained.

Aloke [6] studied experimentally heat transfer performance of a T-section internal fin in a circular tube. He found that for finned tube, friction factor was 3.0 to 4.0 times and pumping power was 3.5 to 4.5 times higher than those of plain tube for the Reynolds number range from 2.0 x 10^4 to 5.O x 10^4. Heat transfer coefficient for finned tube was about 1.5 to 2.0 times higher than that of the plain tube within the same Reynolds numbers.

Saha and Dutta [7] studied laminar swirl flow through a tube fitted with twisted tape where a large Prandtl number viscous fluid was used as working fluid. On the basis of constant pumping power and constant heat duty, short length twisted-tapes placed at inlet showed better performance than the full- length twisted tapes.

Mohammad [8] numerically investigated the heat transfer augmentation for flow in a pipe or a channel partially or fully filled with perforated material placed at the core of the channel. It was shown that partially filling the channel with perforated substrates can reduce the thermal entrance length by 50% and increase the rate of heat transfer from the walls. Alhough the perforated materials contributes to the pressure drop along the channel, an optimum thickness of about 60% of the channel height was found to offer a substantial increase in the Nusselt number at the expense of reasonable pressure drop.

Pavel and Mohamad [9] experimentally investigated the potential of perforated inserts to enhance the rate of heat transfer at constant heat flux where the fluid was air. They found that porosity and diameter of the pipe have a positive influence upon heat transfer and negative impact on pressure drop. The highest increase in the Nusselt number of approximately 5.28 times(compared to smooth tube) was obtained by fully filling the pipe at the expense of the highest pressure drop of 64.8 Pa. In comparison with fully filling the pipe, a partial filling has the advantages of a same increase in the Nusselt and a smaller increase in the pressure drop.

Sarkar et al. [10] studied the heat transfer in turbulent flow through tube with wire-coil inserts. For the same Reynolds number heat transfer coefficient for tubes with wire- coil inserts varied from 1.2 to 2.0 folds than that of the plain tube. But the friction factor increased from 1.5 to 4.0 times than that of the plain tube. Garcia et al.[11] studied the enhancement of heat transfer with wire-coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers. In the turbulent flow wire-coils cause higher pressure drops. But in the laminar region tube with wire coil behave like plain tube and transition takes place at low Reynolds number (Re=700). At costant pumping power,wire coil insert shows an increase of heat transfer rate at Reynolds number below 30000 over the plain tube.

Ahmed et al. [12] studied the heat transfer for turbulent flow through a circular tube with twisted tape inserts having different twist ratios. They found higher heat transfer rate in the entrance region for lower twist ratio. The average heat transfer rate found in that work was 3 times higher than that of the plain tube.

So far a very little research work has been reported in literature on heat transfer in turbulent flow through tube with perforated twisted tape inserts. Pores in the twisted tape may cause good mixing of the working fluid during its flow in the downstream. So, the perforated twisted tape insert may be a probable way for heat transfer enhancement in heat exchanger.

The main objectives of this research are, firstly, to study turbulent heat transfer and fluid flow characteristics in a tube with perforated twisted tape insert, secondly, analyze the heat transfer and pressure drop phenomena and to compare the result with the data of plain tube giving an idea of augmentation of heat transfer and increase of pressure drop and thirdly comparison of the results of this experiment with those of previous works.

2. EXPERIMENTAL FACILITY AND PROCEDURE

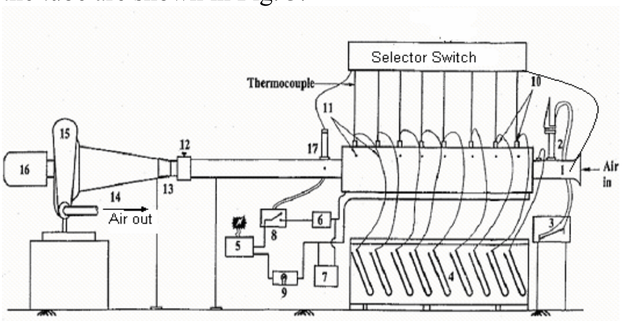
The experimental facility and procedure of collecting the heat transfer data for plain tube and tube with perforated twisted tape inserts are described in this section.

2.1. Experimental Facility

The experimental facility consists of an inlet section, a test section, an air supply system (electric blower) and a heating arrangement. The schematic diagram of the experimental facility is shown in Fig. 1.

2.2. Test section

A cross-sectional view of the tube, with perforated twisted tape insert is shown in Fig. 2. The semi cylindrical plain tube made of two halves are clamped together by flanges at the ends having 70 mm inside diameter and 1500 mm length. The mild steel perforated twisted tapes of different pore diameters are in turn inserted within the plain tube. The length of the tape is 1500 mm and the width is 55 mm. The twist ratio of the insert is 4.55. After placing the tape axially in one half of the tube, then the two halves of the plain tube are clamped. In order to prevent leakage, the putting is pressed into the joint of the tube. Then the plain tube in the test section is covered with mica sheet to isolate electrically the tube. A layer of glass fiber is put on the mica sheet. Nichrome wire (of resistance 1.2 ohm/m) used as an electric heater is spirally wound uniformly with spacing of 16mm around the tube. Then mica sheet, glass fiber tape, heat insulating tape, and asbestos tape are sequentially put over the Nichrome wire heater coil. These protected the radial heat losses. The test section is installed in the test facility with the help of the bolted flanges with gaskets of asbestos (of thickness 3.5 mm) to prevent the heat flow in the longitudinal direction and to prevent leakage of air. All the layers of insulation over the tube are shown in Fig. 3.



1. Inlet section
2. Traversing pitot tube
3. Inclined tube manometer
4. U-tube manometer
5. Variable voltage
6. Ammeter
7. Voltmeter
8. Temperature controller
9. Heater on off lamp
10. Pressure tappings
11. Thermocouples
12. Flow control valve
13 Flexible pipe
14. Diffuser
15. Blower
16. Motor transformer
17. Traversing thermocouple

Fig 1: Schematic diagram of the experimental facility

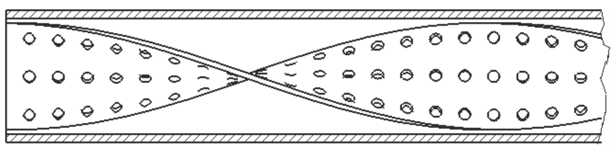


Fig 2: Cross-section of the test section with perforated twisted tape inserts

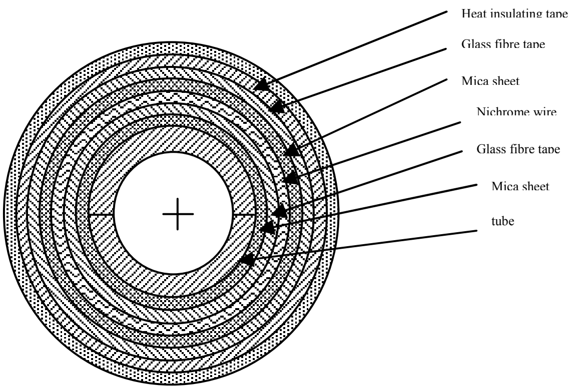


Fig 3: X-section of the test section with insulations

In the similar fashion, seven test specimens are made with seven different perforated twisted tapes. But the twist ratio of the tapes are constant (y=4.55).Twist ratio is the ratio of half of the pitch to the width of the tape. The pore diameter varies from 3 mm to 9 mm in 1mm steps. The distance between two adjacent holes is axially 15 mm and transverse wise 20mm (Fig. 4). Porosity of the twisted tapes is calculated by dividing the total pore area in a tape to the tape area (including holes). Porosity of the inserts varies from 2.5% to 20.8%.

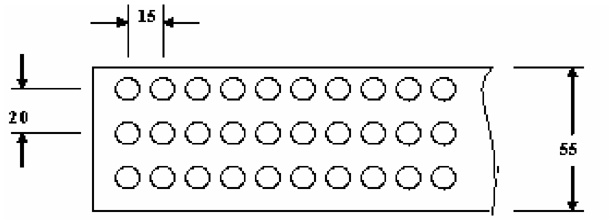


Fig 4: Schematic diagram of the hole position in the plate

2.3 Experimental Procedure

At first the blower is switched on and allowed to run for about five minutes to make the transient characteristics die down. The flow of air through the test section is set to a desired value and kept constant with the help of the flow control valve. Then the electric heater is switched on. The electric power is adjusted (if necessary) with the help of a regulatory transformer or variac. First the variations in wall temperatures at all locations are observed and the test run continues until constant values are attained at all the eight locations. Then the bulk outlet air temperature is measured. Steady state condition is assumed when the outlet air temperature does not fluctuate over 30 minutes time. At the steady state condition, thermocouple readings are recorded with the help of selector switch and at the same time, manometer readings are recorded from the inclined tube manometers.

After each experimental run, the air flow is changed with the help of the flow control valve thereby changing the Reynolds number. At that time, electrical power supply is kept constant. After waiting for steady state condition, data are recorded in the similar way.

3. RESULTS AND DISCUSSION

In this paper, friction factors and heat transfer characteristics in the turbulent flow through the tube with and without perforated twisted tape inserts are analyzed. Seven mild steel twisted tapes of different pore diameters (3 – 9mm) are used and hence porosity of twisted tape inserts varies from 2.5% to 20.8%. Porosity of insert, Rp = 0 indicates the twisted tape without perforation and Rp= $\propto$ indicates the plain tube. The twist ratio (y) of the tape is 4.55 and the central distance between the two adjacent pores is fixed as 15mm (axially) and 20 mm (transverse wise). All the necessary variables have been calculated from the generated data. These data and information are presented and analyzed in this section. Data for the twisted tape insert without holes for twist ratio y= 4.25 are available in the report of, BRTC-2005, BUET which is used to compare with the results of the present work.

3.1 Temperature Distribution

The bulk temperature is the representative of the total energy of the flow at any particular location. The local bulk temperature, $T_b(x)$ at any location, x for the plain tube can be expressed as:

$$mC_p \times \{T_b(x) - T_i\} = q_s W_s X$$

$$\Rightarrow T_b(x) = T_i + \frac{q_s W_s X}{mC_p} \ldots\ldots\ldots\ldots\ldots Eq(1)$$

The local wall and bulk fluid temperatures at different axial locations for plain tube and tube with perforated twisted tape inserts for a particular Reynolds numbers are shown in Fig. 5. It shows the variation of wall and fluid bulk temperature along the axial locations at different porosity of insert for a given Reynolds number (Re=29670). The axial position of any point is non-dimensionalized by the total length of the test section (L). The wall temperatures are recorded directly during experiment. But the local fluid bulk temperatures are calculated using Eq (1). Considering uniform heat flux, the variation of the fluid bulk temperature is assumed as linear through the test section. It is revealed from Fig. 5 that higher wall temperatures occurs in the plain tube for a given axial location. But the bulk fluid temperature is lower in the plain than that in the tube with inserts. At porosity R=4.6%, the wall temperature is lower than that of the tube with other inserts. Tube with twisted tape insert (non-perforated) has higher wall temperature than that of the tube with perforated twisted tape inserts.

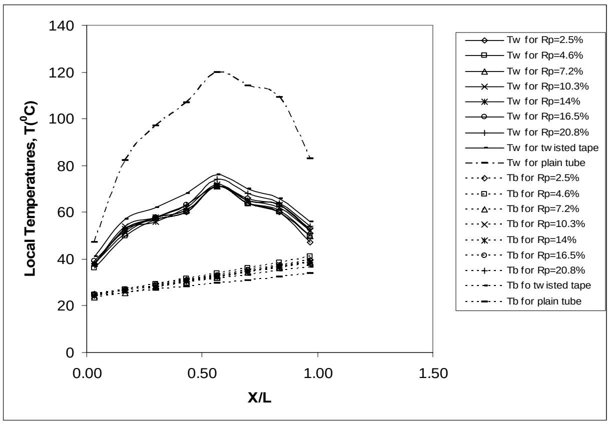


Fig 5: Variation of wall and bulk temperatures along the axial distance for different porosity of insert at Reynolds number around 29670.

Variation of tube wall and fluid bulk temperatures along the axial locations in the test section with perforated twisted tape inserts at different Reynolds number are presented in Fig. 6 and Fig. 7. Fig. 8 shows the variation of tube wall and fluid bulk temperatures along the axial locations for the plain tube at different Reynolds numbers.

From the Fig. 6, 7 & 8, it is observed that the wall temperature increases along the axial position for a given Reynolds number and reaches its maximum at X/L=.577. Then the tube wall temperature drops slightly at the downstream due to end effect. It may be noted that for both cases the wall temperatures are lower at the entry and the exit partly because of the conduction losses (end effect). For both the cases the wall temperatures decrease with the increase in Reynolds number. Higher Reynolds number indicates higher flow rate of fluid and it is possible to take away more heat from the wall. Wall temperatures in the tube with perforated twisted tape insert at any location is lower than that of plain tube for a given Reynolds number. As the plain tube has lower wetted perimeter and less contact area with the working fluid so its ability to transfer heat is low.

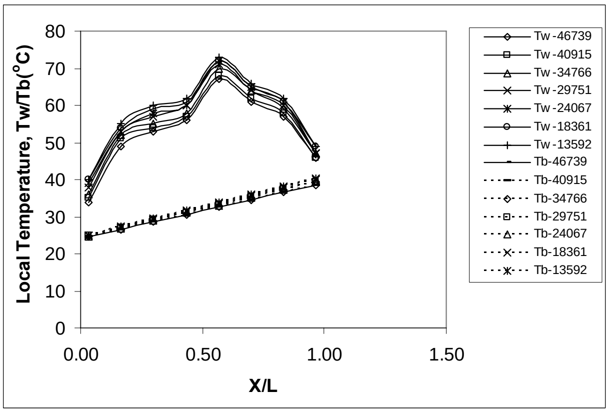


Fig 6: Variation of wall and bulk temperature along axial distance for the tube with the perforated twisted tape having porosity, R_p=2.5 %.

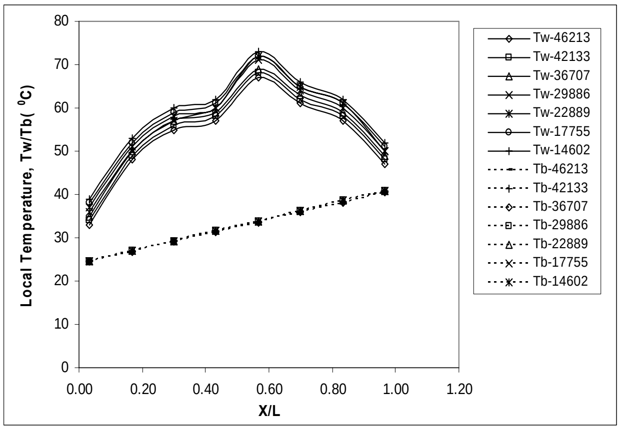


Fig 7: Variation of wall and bulk temperature along axial distance for the tube with the perforated twisted tape having porosity, R_p=4.6 %.

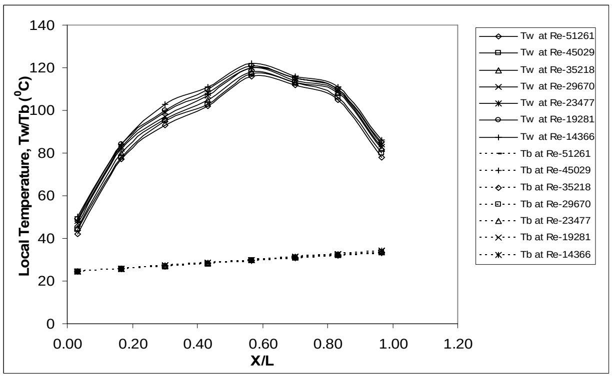


Fig 8: Variation of wall and bulk temperature along axial distance for smooth tube.

The calculated bulk fluid temperatures based on average 'q' along the axial locations of the tube with and without perforated twisted tape inserts at different Reynolds number obviously increase linearly. Because air passes through the heated tube along the length and takes away heat from the tube wall. So, it increases along the tube for both cases (plain tube and tube with the tape). At lower Reynolds number, the bulk fluid temperature is higher. At lower Reynolds number, air flows slowly over the tube surface in the test section as well as the inserts. So, it gets enough time for being heated. Thus the bulk fluid temperature is higher at a particular location at lower Reynolds number. But at higher Reynolds number, faster moving of air gets insufficient time for being heated. From the figures, it is also clear that for the plain tube the bulk fluid temperature is lower than that of tube with the twisted tape inserts at any location for a particular Reynolds number.

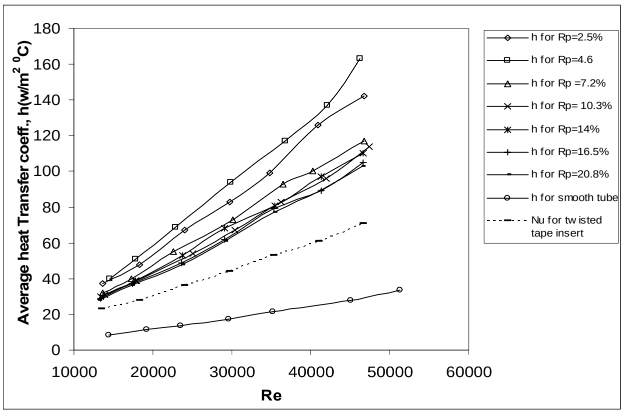


Fig 9: Variation of heat transfer coefficient with porosity of the twisted tape inserts at different Reynolds numbers.

3.2 Effect Of Reynolds Number And Porosity Of The Twisted Tape On Heat Transfer Coefficient

Variation of average heat transfer coefficient with perforated twisted tape inserts at different Reynolds numbers are shown in Fig. 9**.** The figure shows that average heat transfer coefficient increases with the increase of Reynolds number for plain tube and the tube with the inserts. At high Reynolds number, mixing of the fluid occurs and more heat is taken away from the tube. So, the temperature difference decreases. But the heat transfer coefficient increases. Again from the figure it can be noted that average heat transfer coefficient depends on porosity of twisted tape inserts. And average heat transfer coefficient is observed as maximum for the tube with twisted tape inserts having porosity, R_p=4.6%. For porosity, R_p=4.6%, the heat transfer coefficient increases upto 5.5 times compared to that of plain tube. The heat transfer coefficient decreases below and above 4.6% of porosity because the area decreases with the increase of porosity. Lower porosity causes less fluid pass to through the hole. At high speed fluid can not pass through the small hole.

Fig. 10 shows the variation of average heat transfer coefficient for different inserts like perforated twisted tape inserts (porosity, Rp=4.6%), twisted tape insert, longitudinal strip insert and wire- coil –inserts. From the figure, it is observed that the heat transfer coefficient for the perforated twisted tape insert is approximately 1.7 times higher than that of the twisted tape insert (report of BRTC-2005, BUET). Heat transfer coefficient for the perforated twisted tape insert is 2.0 times higher than that of the longitudinal strip insert by Sarkar et al.[13]. It is also 1.9 times higher than that of wire-coil-inserts by Sarkar et al.[10].

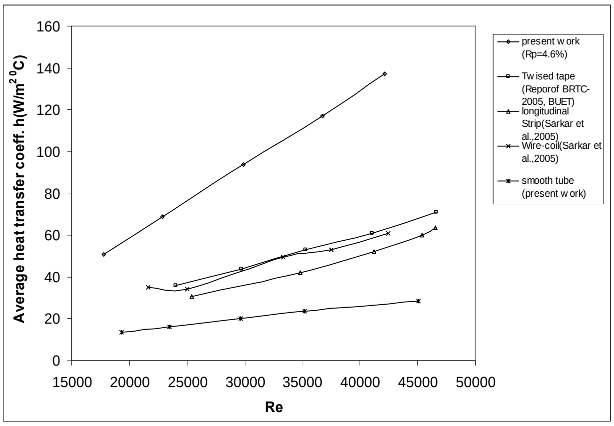


Fig 10: Variation of the average heat transfer coefficient for different inserts (porous and nonporous insert) at different Reynolds numbers.

3.3 Effect of Reynolds Number and Porosity of The Twisted Tape on Nusselt Number

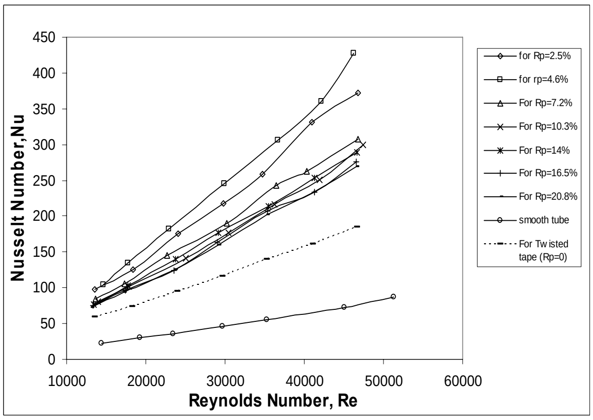


Fig 11: Variation of the Nusselt number with porosity of the twisted tape insert at different Reynolds numbers.

Fig. 11 shows the variation of average Nusselt number with porosity of the twisted tape inserts at different Reynolds number. Average Nusselt number increases with the increase of Reynolds number. Again from the figure, it is clear that the average Nusselt number also depends on porosity of twisted tape inserts. The average Nusselt number based on this local Nusselt number for tube with perforated twisted tape insert varies from 4.0 to 5.5 folds in comparison to the plain tube. Average Nusselt number is maximum for porosity of twisted tape insert, Rp= 4.6%. Below and above Rp= 4.6% of porosity of the twisted tape insert, Nusselt number decreases. So tube with perforated twisted tape insert is advantageous over the plain tube.

3.4 Effect of Reynolds Number and Porosity of the Twisted Tape on Heat Transfer Rate

Fig. 12 shows the variation of heat transfer rate for porosity of twisted tape inserts at different Reynolds number. The figure indicates that the heat transfer rate increases with the increase in the Reynolds number for all the cases. At higher Reynolds number, there is intensive mixing of air, which increases the heat transfer rate. It is also observed that the heat transfer rate is higher for perforated twisted tape inserts than that of the plain tube. Perforated twisted tape inserts create swirl flow in the test section. Heat transfer rate for tube with the perforated twisted tape insert increases upto 1.8 folds in comparison to the plain tube. The heat transfer rate also changes with the porosity of the twisted tape inserts. The heat transfer rate is maximum when the porosity of twisted tape insert is 4.6%. It decreases with the increase of porosity as well as decrease of porosity of the twisted tape inserts. High porosity causes less swirling effect. Hence the heat transfer rate is decreased in the tape of higher porosity. Low porosity fails to produce the necessary turbulence. At the same Reynolds number, tube with perforation 4.6% shows higher heat transfer rate than the other inserts within the range of porosity Rp=2.5% to 20.8%. So it is more useful than the plain tube.

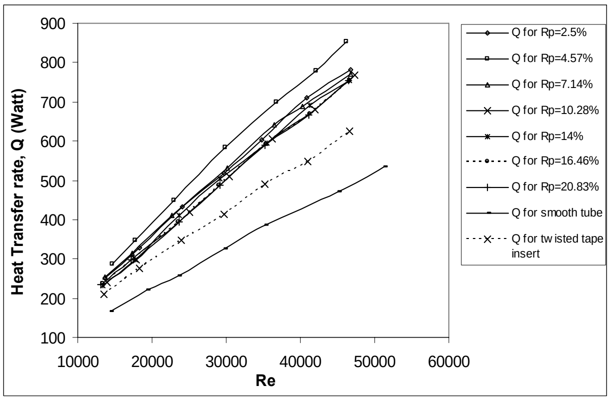


Fig 12: Variation of the heat transfer rate with porosity of the twisted tape inserts at different Reynolds numbers.

By using insert and increasing the flow rate, heat transfer rate is increased. But at that condition, pumping power required is also increased. Relation of the pumping power with the increase of the heat transfer rate is shown in the Fig. 13. The figure shows that at the same pumping power, heat transfer rate is higher for perforated twisted tape than that of the plain tube. So, the use of perforated twisted tape inserts for heat transfer enhancement is advantageous over the plain tube. Among the inserts, porosity of 4.6% shows the best performance within the tested range.

3.5 Effect of Reynolds Number and Porosity of The Twisted Tape on Pressure Drop

Fig 14 shows the variation of total pressure drop with Reynolds number for plain tube and tube with perforated twisted tape inserts. It is clear from the figures that at lower Reynolds number, change of pressure drop is comparatively high, and at higher Reynolds number, it is low. This may be explained by the fact that at lower values of Reynolds number, corresponding to lower flow rates, air can pass all the pores and touches the tape and create high frictional forces. At a given Reynolds number, pressure drop in the tape insert is higher than that of the plain tube at any axial location of the test section. For tube with the inserts, due to turbulence and secondary flow more pressure drop occurs than that of the plain tube.

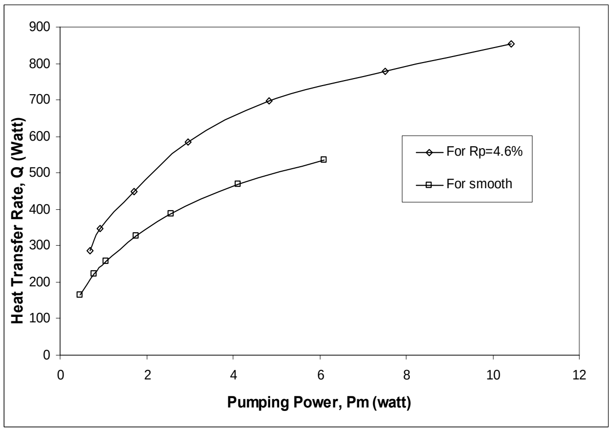


Fig 13: Variation of the heat transfer rate with the pumping power.

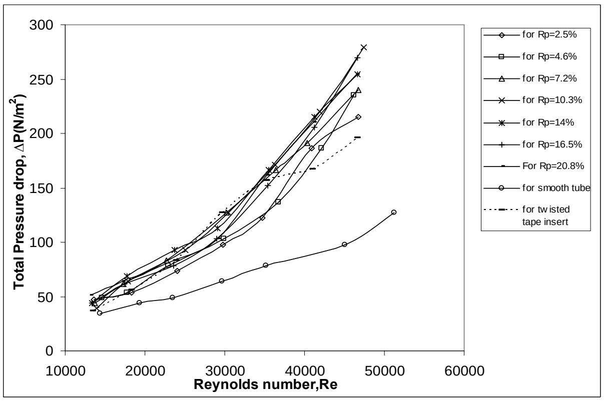


Fig 14: Variation of the total pressure drop with porosity of twisted tape insert at different Reynolds numbers.

3.6 Pumping Power

Fig. 15 presents the variation of pumping power with Reynolds number for both the plain tube and the tube with perforated twisted tape inserts. For all the tubes pumping power increases as the Reynolds number increases. From the figure, it can be noted that the required pumping power for the same Reynolds number for the tube with perforated twisted tape insert is slightly higher than that of the plain tube. The presence of small vortices behind the tape is responsible for higher pressure drop and pumping power. The required pumping power for tubes with perforated twisted tape insert increases upto 2.4 folds than that of the plain tube.

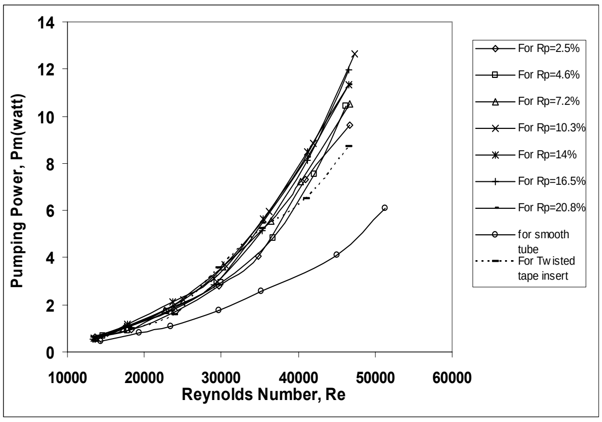


Fig 15: Variation of the pumping power required for different porosity of twisted tape inserts at different Reynolds numbers.

3.7 Effectiveness

Effectiveness is the measure of performance of heat exchanger. Considering the wall temperature to be constant and same through out the entire heated section, effectiveness of the heat exchanger **(ε)** can be defined as:

$$\text{Effectiveness}, \varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}}$$

$$= \frac{m \times C_p (T_o - T_i)}{m \times C_p (T_{wav} - T_i)}$$

$$\Rightarrow \varepsilon = \frac{(T_o - T_i)}{(T_{wav} - T_i)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots Eq(2)$$

Where,
T_o= Bulk outlet temperature of the air (^{o} C)
T_i = Bulk inlet temperature of the air (^{o} C)
T_{wav}= Average wall temperature of the test section of the tube (^{o} C)

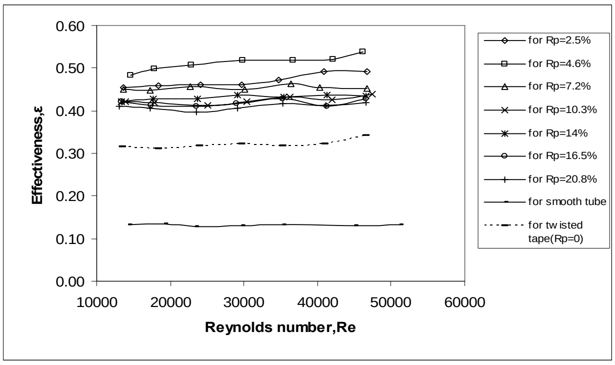


Fig. 16: Variation of the heat transfer effectiveness with porosity of the twisted tape inserts at different Reynolds numbers.

For performance analysis, effectiveness is the measure of heat exchanger performance where wall temperature should be considered as constant. Assuming constant wall temperature through the test section, the heat transfer effectiveness for both plain tube and tubes with perforated twisted tape inserts are calculated using Eq(2). Fig. 16 shows the variation of heat transfer effectiveness for plain tube and the tube with the inserts as a function of Reynolds number. From the figure it is observed that the heat transfer effectiveness increases slightly with the increase of Reynolds number. But it is higher for the tube with the tape inserts than that of the plain tube. The heat transfer effectiveness for tube with perforated twisted tape insert varies from 3.70 to 4.0 folds than that of the plain tube.

4. CONCLUSIONS

An experimental study has been conducted to investigate the turbulent flow heat transfer in a tube with perforated twisted-tape-insert. The study has revealed that the perforated twisted tape insert causes an increase of heat transfer rate at the cost of increased pumping power. The conclusions of the present study are given below:

1. The friction factor is high at the inlet of the test section and drops sharply towards the downstream upto around x/L=0.2 and then becomes almost constant.
2. The pumping power required for the tube with the perforated twisted tape insert varies from 1.2 to 2.25 times compare to that of the plain tube.
3. The average heat transfer coefficient for tube with perforated twisted tape insert varies from 4.4 to 5.5 folds compared to that of the plain tube.
4. Nusselt number is high in the entrance and exit regions. In the region approximately between x/L = 0.2 and 0.8, the Nusselt number is more or less constant.
5. Among the inserts tested, the tube with porosity of 4.6% gives the highest heat transfer rate, Q for the same Reynolds number and is around 1.8 times the value of the plain tube.
6. The heat transfer effectiveness for the tube with the perforated twisted tape insert is higher than that of the plain tube and the maximum value, being 4.0 folds, occurs for Rp = 4.6%.
7. Heat transfer rate, Q for the tube with the perforated twisted tape insert is higher than that of the twisted tape insert of the same twist ratio but without perforation, the data being taken from the report of BRTC-2005, BUET. But the required pumping power for the tube with the perforated twisted tape insert generally is not significantly different from that of the non-perforated twisted tape insert.

5. REFERENCES

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