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HEAT TRANSFER IN TURBULENT FLOW THROUGH A CIRCULAR TUBE WITH TWISTED TAPE INSERTS

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ABSTRACT
Heat transfer and pressure drop characteristics in a circular tube fitted with twisted tape inserts have been investigated experimentally. Experiments were conducted with tape inserts of three different twist ratios (twist ratio, y=23, y=11.5 and y=8). The tubular test section was electrically heated, and air was allowed to flow as the working fluid. Air velocity, air inlet and outlet temperatures, local wall temperature, local tape surface temperature and pressure drop were measured and calculated for tubes with twisted tape inserts to analyze the friction factor, Nusselt Number and the heat transfer coefficient. Reynolds Number was calculated based on inside diameter of the tube and varied from 2.0×10^4 to 5.5×10^4. The result indicated that average heat transfer coefficient is about 1.3 to 3 times higher than that of the smooth tube.

Keywords: Twist ratio, Heat transfer coefficient, Nusselt number, Pressure drop, Friction factor

1. INTRODUCTION
Heat transfer enhancement is the process of improving the performance of a heat transfer system. It generally means increasing the heat transfer coefficient. The performance of heat exchanger depends how effectively heat is utilized. The high performance of heat exchangers are very much essential in many practical applications such as aerospace, vehicles, refrigeration and air conditioning, cooling of electric equipment and so on. Reduction of the size of the heat exchanger may be possible due to improvement in the performance of heat exchanger. On the other hand, a high performance heat exchanger of a fixed size can give a increased heat transfer rate and also there is decrease in temperature difference between the process fluids enabling efficient utilization of thermodynamic availability. The performance can be improved by using various augmentation techniques such as finned surfaces, integral roughness and insert devices. A variety of different techniques are employed for the heat transfer process and extensive reviews of these methods and their applications have been given by Bergles [1] and Webb [2].

Many active and passive techniques are currently being employed in heat exchangers, with twisted tape inserts providing a cost-effective and efficient means of augmenting heat transfer. For almost a century, twisted-tape inserts have been used to enhance heat transfer. At the time of their beginning known as "retarders" a name perhaps due to the increased pressure drop.

Friction and heat transfer characteristics of turbulent air flowing through tubes with twisted strip swirl promoters were studied experimentally and analytically by Thorsen and Landis [3]. Data were obtained for pitch to diameter ratios as low as 3.15 and for Reynolds number up to $1x10^5$. Both heating and cooling tests have been run for tube wall to fluid bulk temperature ratios from 0.6 to 0.9 to evaluate compressibility and buoyancy effects. They have observed that under large temperature gradients in swirl flow, the heat transfer has been enhanced in heating and reduced in cooling due to buoyancy effects introduced by the centrifugal forces generated in the curved flow fields.

Gee and Webb [4] presented the experimental study of the single-phase forced convection in a circular tube containing two dimensional helical rib roughness. The experiment was run by using air as working fluid and the three helix angles (30^0, 45^0 and 70^0) all having a rib pitch-to-height ratio of 15. Their investigation indicates that helical rib-roughness yields greater heat transfer per unit friction than transverse rib-roughness.

Sethumadhavan and Raja Rao [5] presented results from experimental investigations of heat transfer and fluid friction in a duct tightly fitted with helical wire coil inserts of varying pitch and wire diameter.

Uttarwar and Raja Rao [6] performed a experimental study on isothermal pressure drop and heat transfer with seven coiled-wires inserts of varying pitch in a tube for servotherm medium grade oil in laminar flow and then compared with a smooth tube. It was found that heat capacity was increased as much as 350% and also reduction in heat exchanger area of about 70% to 80%.

Du Plessis and Kroger [7] presented a heat transfer correlation for thermally developing laminar flow in a tube with twisted tape insert. They observed that there was enhancement of heat transfer with lower twist ratio at the cost of extra pumping power needed to overcome the adverse increase in pressure drop.

Manglik and Bergles [8] presented experimental investigations and developed heat transfer and pressure drop correlations for laminar flows of water and ethylene glycol in a tube with twisted tape inserts of three different twist ratios i.e. 3.0, 4.5, and 6.0.They have observed that tape geometry depending on the flow rates strongly influenced the heat transfer and pressure drop characteristics. They also observed swirl effect dominates with low twist pitch (H).

Mergerlin et al. [9] found that bristle brushes provide enhancement ratio (h/h_p) as high as 8.5, but the pressure drop were increased a factor of 2800.

Bergles and Josi [10] presented a survey of the performance of different types of swirl flow devices for laminar flow.

Friction and Nusselt number data were measured and semi-empirically evaluated by Gupte and Date [11] 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 2.659. 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.

This work presents the experimental investigations for airflows in a tube with twisted tape inserts of three different twist ratios for turbulent flow conditions. The various heat transfer parameters are graphically analyzed and observed the effect of twist pitch of tape inserts.

The present study was therefore undertaken:

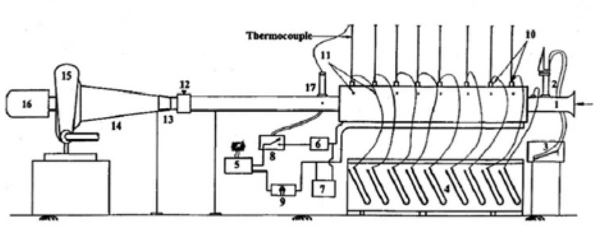
a) To fabricate an experimental facility for studying turbulent flow heat transfer and fluid friction in a tube with twisted tape inserts.
b) To analyze the heat transfer performance.
c) To compare the results of this experiment of previous works (smooth tube)

2. EXPERIMENTAL SET-UP

Test section is provided with inlet section, air supply system and the heating system which is shown schematically in the Fig 1.

2.1 Test Section

The test section was designed and fabricated of Mild Steel (1500 mm long and 70 mm inside diameter) with circular tube .The tube was wrapped with mica sheet and over the mica sheet Nichrome wire (of resistance 0.61 ohm/m) was spirally wounded uniformly with spacing of 4 mm. The Nichrome wire was covered with mica sheet, glass fiber tape and insulation tape to make it electrically insulated by covering with asbestos. The test section was placed in the test rig with the help of the bolted flanges, between which asbestos sheets were installed which act as heat guards in the longitudinal direction. Tape inserts were placed within the tube for each observation whose tape geometry was different.



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

Fig 1. Schematic Diagram of the Experimental Set-up

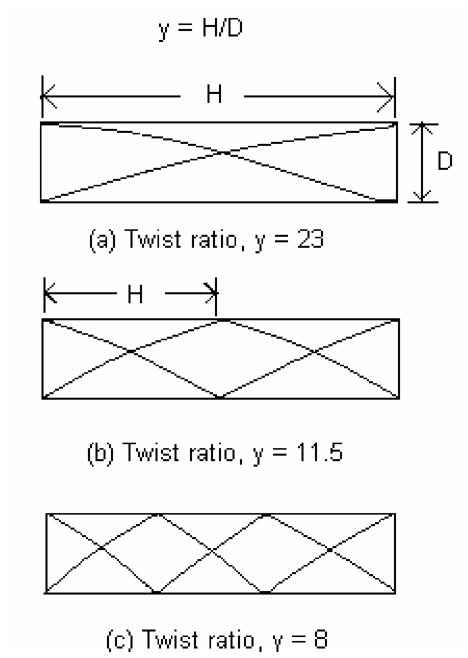


Fig 2. Tape geometry of twisted tape inserts

2.2 Inlet Section

The unheated inlet section cast from aluminum which was shaped to avoid any flow disturbances at upstream of the test section to get fully developed flow in the test section as well.

2.3 Air Supply

A motor operated blower was fitted at the downstream of the test section to supply air that was taken heat from uniformly heated tube for ascertaining the heat transfer performance. A 12° diffuser made of 1.5875 mm mild steel plate was fitted to the suction side of the blower which was used for minimizing head loss at the suction side. A flexible duct was placed to minimize the vibration of the blower. Flow rate of air was controlled by a gate valve, which was fitted at the suction side.

2.4 Heating

The tube was electrically heated at constant heat flux with the help of Nicrome wire and power was supplied to the heater by a 5 KVA variable voltage transformer connected to 220V AC power through a magnetic contactor and temperature controller. A temperature controller was fitted to sense the outlet air temperature to provide signal for switching the heater off or on automatically. It protects the experimental setup from being excessively heated which may happen at the time of experiment when the heating system is in operation continuously for hours to bring the system in steady state condition.

2.5 Procedure

At first the blower was switched on and allowed to run for few minutes to remove all transient characteristics. The flow rate of air was fixed by the flow control valve. Then the electric heater was switched on and the electric power was adjusted with the help of a regulating transformer. Steady state condition for temperature at different locations of the test section was defined Gee and Web (1980) [4] by two measurements. First the variation of tube surface temperatures were observed until the constant value was attained and then the outlet temperature was observed until the temperature didn't deviate over fifteen to twenty minutes. Then the thermocouples readings and manometer readings were taken at the steady state condition. Then air flow rates were changed through the tube by flow control valve under the same electrical heat input and the corresponding readings were taken at steady state condition. The procedure was repeated for different angled twisted tape inserts in the test section.

2.6 Measurement

Air flow rate was measured at the inlet section with the help of traversing Pitot tube. The static pressure tapping were made at the inlet of the test section as well as equally spaced in eight axial locations. U-tube manometers at an inclination of 30° were attached with the pressure tapping and water was used as the manometric fluid. The temperature at the different locations of the section was measured with the K-type thermocouples. The temperature measuring locations are: 1. air outlet temperature at the outlet of the test section, 2. tube surface temperature at eight axial locations of the test section, 3. tape surface temperature at eight axial locations of the twisted tape.

3. EQUATIONS USED FOR CALCULATION

Pressure drop is hydraulic loss due to the roughness of the surface over which the fluid is moving. Pressure drop at any axial location x is given by the following equation $\Delta P = P_i - P_x$

where P_i = Pressure at inlet

P (x) = pressure at any axial location, x

The local friction factor based on inside diameter is given by

$$f = \frac{\left((-\Delta p / X) D_i\right)}{2\rho V^2} \qquad (1)$$

Heat transfer rate is the energy transfer to the air per unit time. Total rate of heat input to the air

$$Q = m^{\cdot} \times c_p \times (T_o - T_i) \qquad (2)$$

For smooth tube, rate of heat input to the air per unit area Heat flux,

$$q = \frac{Q}{A} = \frac{Q}{WL} \qquad (3)$$

The local bulk temperature of the fluid $T_b(x)$ can be defined as the following heat balance equation

$$T_b(x) = T_i + \frac{qA_x}{mc_p} \qquad (4)$$

The local heat transfer coefficient at any axial location can be defined as

$$h_x = \frac{q}{(T_{sx} - T_{bx})} \qquad (5)$$

The average heat transfer coefficient can be defined as

$$\bar{h} = \frac{q}{(T_{sx} - T_{bx})_{avg}} \qquad (6)$$

For twisted tape insert tube, heat input to the air per unit area

$$q = \frac{Q}{A} = \frac{Q}{W_{wt} L} \qquad (7)$$

The local bulk temperature of the fluid $T_b(x)$ can be defined as the following heat balance equation

$$T_b(x) = T_i + \frac{qA_{xt}}{mc_p} \qquad (8)$$

The local Nusselt number based on inside diameter (for both smooth and twisted tape insert tube) can be defined as

$$Nu_x = \frac{h_x \times D_i}{k} \qquad (9)$$

Blower power, Pm can be expressed as

$$P_m = \frac{\Delta P \times m^{\cdot}}{\rho} \qquad (10)$$

4. RESULTS AND DISCUSSION

Various heat transfer parameters are mapped from experimental data. Both Reynolds number and the twist ratio have the great influence in heat transfer performance, which is critically observed, from the graphical analysis.

In Fig 3., local heat transfer coefficient is greater in entrance region and gradually decreases, again increases at the downstream along longitudinal direction of test section. The local heat transfer coefficient increases with decreasing the twist ratio for comparable Reynolds number.

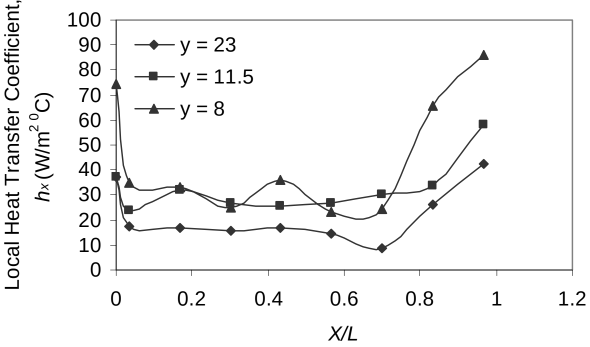


Fig 3. Comparison of Local Heat Transfer Coefficient along the length of the tube for comparable Reynolds Number

The variation of average heat transfer Coefficient against Reynolds number is shown in Fig 4. The average heat transfer coefficient increases as the Reynolds number increases. The average heat transfer coefficient is increased with lower twist ratio due to increase of rate of heat transfer. Due to twisting effect the swirl flow superimposed over the axial flow and also fluctuation of velocities helps to enhance heat transfer.

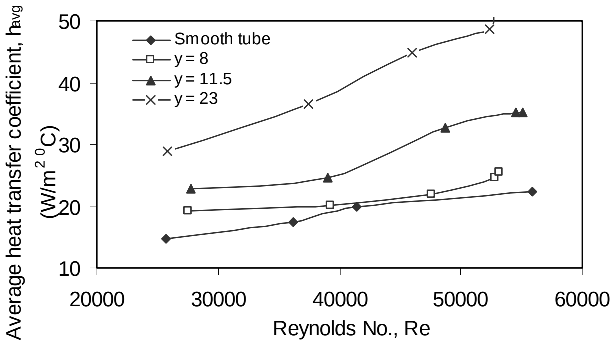


Fig 4. Comparison of Average Heat Transfer Coefficient for different twist ratios

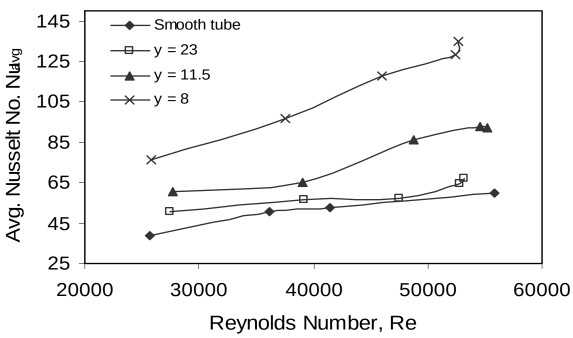


Fig 5. Comparison of Average Nusselt Number for different twist ratios

In Fig 5., Average Nusselt Number is much higher with lower twist ratio. Average Nusselt number increases as Reynolds number increases.

In Fig 6., the pressure gradient is high in the entrance region, then pressure recover and finally approaches the fully developed values away from the entrance region. The pressure drop increases along the length of the tube but greatly increased with lower twist ratio.

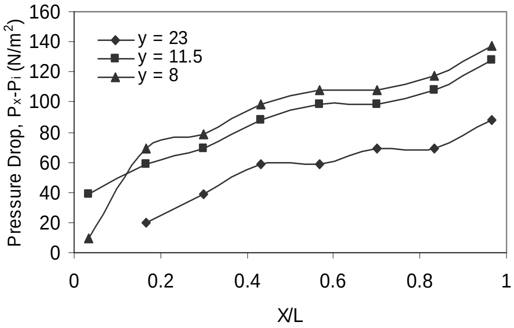


Fig 6. Comparison of Pressure drop distribution along the length of the tube for comparable Reynolds Number

In Fig 7., Friction Factor decreases along the length of the test section for twist ratio of 8. Friction Factor is high near the entrance region and then falls gradually to fully developed value.

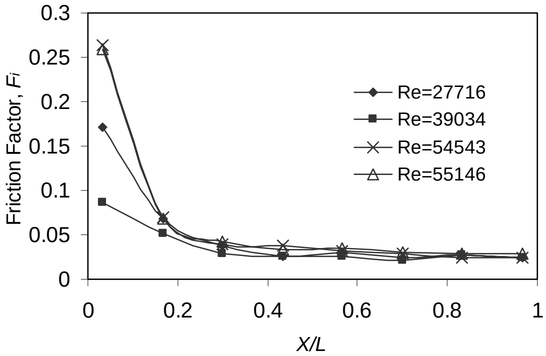


Fig 7. Variation of Friction Factor along the length of the tube for different Reynolds Number (y = 8)

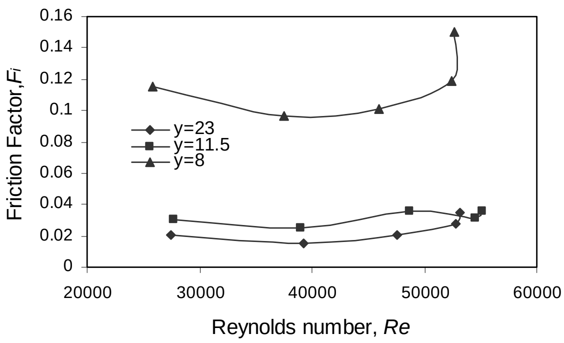


Fig 8. Variation of friction factor against Reynolds Number for different twist ratios

Friction Factor were calculated from the pressure drop measured from different location with respect to inlet section based on the inside diameter of the tube. Friction Factor increases with decreasing twist ratio.

In Fig 9., more blower power is required at higher Reynolds Number. Twisted tape inserts of different twist ratio also affects the required blower power. It is observed that blower power is required to be high in case of lower twist ratio. Extra blower power is needed to overcome adverse increasing pressure drop with decreasing the twist ratio.

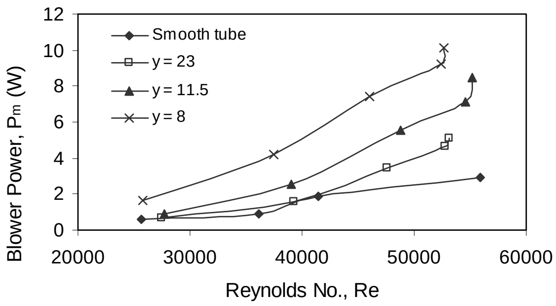


Fig 9. Comparison of Blower Power for different twist ratios

5. CONCLUSIONS

An experimental study has been conducted to investigate the heat transfer augmentation in a tube by means of twisted tape inserts. The study has revealed that the twisted tape inserted tubes enhance heat transfer rate at the cost of increased pumping power. The results of the present study conclude that

a) The local friction factor is high near the inlet section and drops gradually to the fully developed flow.
b) The blower power is required to be high with decreasing the twist ratio. Blower power for tubes with twisted tape inserts varies from 1.2 to 3.5 folds than that of the smooth tube. More blower power is required to be high with lower twist ratio.
c) Average heat transfer coefficient increases with the increase of Reynolds number. Average heat transfer coefficient for tubes with twisted tape inserts is about 1.3 to 3 times higher than that of the smooth tube.
d) Average Nusselt Number increases with the increase in Reynolds Number. Average Nusselt Number is high for a lower twist ratio at a comparable Reynolds Number.

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7. NOMENCLATURE

Symbol	Meaning	Unit
A_x	Cross-sectional area of the tube	(m^2)
A_{xt}	Cross-sectional area of the tape	(m^2)
b	Atmospheric Pressure head	(mm of Hg)
d	Velocity Head at inlet section	(inch of water
D_i	Inside Diameter of the tube	(mm)
H	Length between 180° Twist	(mm)
y	Twist ratio (H/D_i)	-
D_h	Hydraulic Diameter	(m)
f	Friction Factor	-
Nu	Nusselt Number	-
Re	Reynolds Number	-
$\dot{m}$	Mass flow rate (kg/s)	-
W	Wetted Perimeter	(mm)
L	Length of the tube	m

X	Axial Distance	(mm)
t	Thickness of the tape	(mm)
w	Width of the tape	(mm)
μ	Fluid Dynamic viscosity	($N.s/m^2$)
c_p	Specific Heat of air	(kJ/kg˚C)
k	Thermal Conductivity of air	(W/m.˚C)
ρ	Density	(kg/m^3)
ΔP	Pressure Drop	(N/m^2)
t	Room Temperature	(˚C)
T	Temperature (˚C)	(˚C)
Q	Rate of Heat Transfer	(W)
q	Rate of Heat Flux	(W/m^2)