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Prediction of Friction Factor and Heat Transfer Coefficient for A Non-Circular Duct With Twisted Tape

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ABSTRACT

a Heat transfer and pressure drop characteristics in a circular tube and non-circular duct (square and hexagonal) with twisted tapes have been investigated experimentally using air as working fluid for the Reynolds number in the range of 800 to 1,05,000. The swirl was generated by tape inserts of various twist ratios. The various twist ratio considered for the study are (Y= 3.5, 4.5, 5.5, 6.5). The width of the tape used was34 mm. The heat transfer test section was heated electrically using Nichrome wire, imposing axially and circumferentially constant wall heat flux (UHF) boundary condition. The length of the test section used was one meter. The circular tube and non-circular duct have same hydraulic diameter of 35 mm. The heat transfer and pressure drop characteristics were compared for various ducts. Both pressure drop and heat transfer increase as the twist ratio (Y) decrease. The highest Nusselt number is obtained for twist ratio of 3.5 for each duct. For the same test conditions Nusselt number and friction factor increases as the number of sides increase of non-circular duct. The maximum Nusselt number and pressure drop is observed in case of circular tube.

***Keywords*— a Non-circular ducts, Twisted Tapes, Hydraulic diameter, Twist ratio.**

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I. INTRODUCTION

Ducts with non-circular cross section are widely used in the heat exchangers and other devices. Heat transfer occurs at the duct wall. Laminar flow develops an insulating blanket around the duct wall and restricts heat transfer. Conversely, turbulent flow, due to the agitation factor, develops no insulating blanket and heat is transferred very rapidly. In the design of heat exchangers turbulent flow is preferred over laminar flow as it leads to better mixing of fluids which increases Nusselt number and in turn heat transfer. In many instances, the designer is faced with an existing equipment, where the space occupied by the cooling passage is minimum and the heat and mass flow rates are limited by the size of an existing or retrofit pump or fan. In these situations, where a coolant passage must be designed so that the volume of the passage is restricted to some value and the heat rate and mass flow rate of the coolant are dictated by the available equipment. In such cases the non-circular duct might be the option. Because of size and volume constraints in applications to aerospace, nuclear, biomedical engineering and electronics, it may be required to use non-circular flow-passage geometries, particularly in compact heat exchangers. The process of increasing the thermal performance of heat exchangers, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as *Heat transfer Augmentation*. These techniques are also referred as

Heat *transfer Enhancement* or *Intensification*. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger. Use of Heat transfer enhancement techniques lead to increase in heat transfer coefficient but at the cost of increase in pressure drop. So, while designing a heat exchanger using any of these techniques, analysis of heat transfer rate & pressure drop has to be done. Twisted tape inserts which is passive augmentation technique for enhancement of heat transfer can be inserted in non-circular duct. The purpose of the twisted tape in the turbulent region is to reduce that resistance near the wall to promote better heat transfer. A duct with a twisted tape produces swirl and cause intermixing of the fluid which leadsto better performance than a plain duct. Heat transfer rate is improved effectively with the increase in the frictional losses. The correlations for Nusselt Number *Nu* and friction factor *f* also depend on whether flow is laminar or turbulent. Hence it is desirable to a correlation that is applicable for wide range of Reynolds number.

Various types of heat transfer augmentation techniques have been studied extensively which include active techniques, passive techniques and compound techniques [1, 2, 4-7]. Kumar and Murugesan [3] studied passive heat transfer augmentation techniques using twisted tapes.

Heat transfer enhancement in circular tubes using twisted tapes has been carried out by many researchers for past decades [9-19]. Agarwal [9] determined isothermal and non-isothermal friction factors and mean Nusselt number for uniform wall temperature (UWT) of servotherm oil (*Pr* 195-375), which was experimentally determined for flow through circular tube with twisted tape insert with (Re 700-4000). The Nusselt number was found out to be 1.21- 3.4 times the Nusselt number of a plain tube. Based on this correlation was developed to find out isothermal friction factor and Nusselt number.Saha and Dutta [10] investigated Laminar swirl flow of a large Prandtl number (205<*Pr*<518) viscous fluid was considered. The swirl was generated by short-length twisted-tape inserts; regularly spaced twisted-tape elements with multiple twists in the tape module and connected by thin circular rods. Shah et al. [11] experimentally investigated heat transfer and pressure drop characteristics in a circular tube fitted with regularly spaced twisted tape element. Laminar flow of viscous fluid with mid Prandtl number was considered. SmithEiamsa-ard [13] used a helical tape insert in the tube with a view to generating swirl flow that helps to increase the heat transfer rate of the tube. The flow rate of the tube is considered in a range of Reynolds number between 2300 and 8800.Jaisankar el al. [15] experimentally investigated heat transfer, friction factor and thermal performance of twisted tape solar water heater with various twist ratios and the results were compared with plain tube collector for the same operating conditions with Reynolds number varied from 3000 to 23,000.Nagarajan and Sivashanmugam [17] invested heat transfer and friction factor characteristics of circular tube fitted with 300 right-left helical screw inserts with 100 mm spacer of different twist ratio for laminar and turbulent flow.

Compound heat transfer augmentation technique also has been area of interest for many researchers [19-21]. Sarma et al. [24] formed generalized correlations to predict friction factors and convective heat transfer coefficients with twisted tapes in a tube for a wide range of Reynolds numbers and Prandtl numbers. Satisfactory agreement was obtained between the present correlations and the data of others validate the proposed correlations. The theoretical predictions were compared with earlier correlations revealing good agreement between them.

In the above literature review, most studies are mainly focused on the heat transfer enhancement using twisted tape. An investigation on non-circular ducts is rarely reported. Therefore, the main aim of the present work is to form a correlation for Nusseltnumber and friction factor non-circular

duct with twisted tape. Experiment is performed on circular tube and non-circular ducts using air as the test fluid for Reynolds number of wide range.

II. EXPERIMENTAL SETUP

The experiment wasconducted in an open-loop experimental facility as shown in Fig. 1. The loop consisted of a blower, reducer, control valve, orifice meter to measure the flow rate, calming section (1100 mm= 31.5 D_h) and the heat transfer test section with twisted-tape insert is (1000mm). Since the air flow is swirling at the exit of test section, an exit section (525 mm= 15 D_h) is provided at the end of test section. The layout of all the sections is shown in Fig. 2. All the ducts are made of stainless steel. All the non-circular ducts have same hydraulic diameter (D_h) of 35mm. The thickness (t) of the ducts is 1.5mm. Hence the inner diameter (D_i) and outer diameter (D_o) of all the ducts is 35mm and 38mm respectively. Twisted tapes were made from stainless steel strips of thickness (δ) 0.6 mm and (D) width 34 mm and twist ratios (Y=H/D) 3.5, 4.5, 5.5, 6.5. They were fabricated by twisting a straight tape, about its longitudinal axis, while being held under tension on lathe machine. Once the twisting was done they were held on lathe machine, and were heated using brazing torchforstress relieving.The twistratio is defined as the ratio ofpitch length of twisted tape for (180º) to the width of tape.The test section was heated using Nichrome wire to provide uniform heat flux boundary condition.The electrical output was controlled by variac transformer to obtain constant heat flux along entire length of test section.

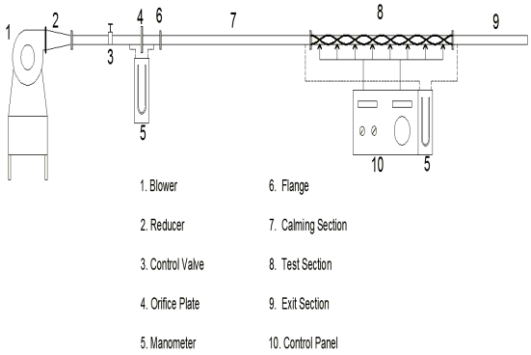


Fig.1 Test setup.

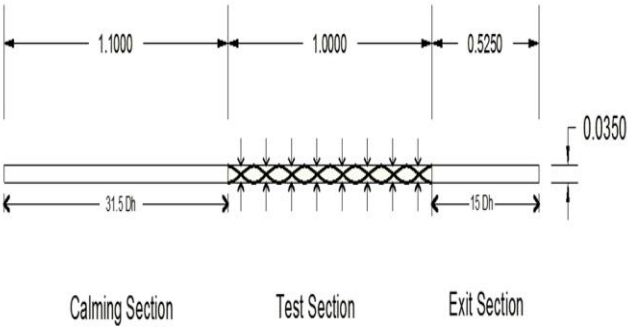


Fig.2Arrangement and dimensions of different sections.

The number of turns of Nichrome wire was calculated according to the maximum heat input required. Gauge 22 Nichrome wire was used which has a resistance of 3.2 Ω/m. One more layer of glass tape is wound over the Nichrome wire. To minimize the heat loss to surroundings due to convection, a layer of asbestos wool is wrapped over the last layer of the glass tape. The inner and outer temperatures of the bulk air were measured using K-Type thermocouple. The K- Type thermocouple (chromel – alumel) is the most common general purpose thermocouple with a sensitivity of approximately 41µV/ºC. The wall temperatures were measured by thermocouples which were brass brazed on the test section at equal distance. All the temperatures were recorded on multi temperature indicator. A blower of 3 m^3/min flow rate was used to draw air from surroundings and was directed through the orifice meter and passed to the heat transfer test section with twisted tape. The flow rate was controlled using a control valve. The air flow rate was measured by orifice meter which was calibrated using anemometer. Manometric fluid used in U-tube manometer was water.

III.DATA REDUCTION

In this work air is used as working fluid. The height of water column measured by U-Tube manometer is converted to equivalent height of air column, then Re is calculated by using equations (1) to (4).

Height of air column is calculated by:

$$H_{air} = H_w\left(\frac{\rho_w}{\rho_{air}} - 1\right) \quad (1)$$

Velocity of air is calculated as:

$$V_{air} = C_d\sqrt{2\, g\, H_{air}} \quad (2)$$

Mass flow rate is given by:

$$\dot{m} = \rho_{air} A_{cs} V_{air} \quad (3)$$

Reynolds number is given by:

$$Re = \frac{\rho_{air} V_{air}\, D_h}{\mu} \quad (4)$$

Heat carried by air is given by:

$$Q_{hc} = \dot{m}\, C_p\, (T_{bo} - T_{bi}) \quad (5)$$

The steady state of heat transfer is assumed equal to heat loss in test section which is given by:

$$Q_{hc} = Q_{conv} \quad (6)$$

The convection heat transfer from test section is given by:

$$Q_{conv} = hA_s\, (T_{wm} - T_{bm}) \quad (7)$$

Where the average wall temperature (T_{wm}) and bulk mean temperature (T_{bm}) are calculated by equation (8) and (9) respectively.

$$T_{wm} = \left(\frac{T_{s1}+T_{s2}+T_{s3}+\cdots+T_{sn}}{n}\right) \quad (8)$$

$$T_{bm} = \left(\frac{T_{bo}+T_{bi}}{2}\right) \quad (9)$$

The average heat transfer coefficient, *h* and average Nusselt number *Nu* is given by:

$$h = \frac{\dot{m} C_p (T_{bo} - T_{bi})}{A_s (T_{wm} - T_{bm})} \quad (10)$$

$$Nu = \frac{h\, D_h}{k} \quad (11)$$

Friction factor*f* can be given by:

$$f = \frac{H_{air}\, 2\, g\, D_h}{L\, V^2} \quad (12)$$

Uncertainty was determined by the method of Kline andMcClintock [25]. The uncertainty in Reynolds number, friction factor and Nusselt numberwere ±2.18%, ±6.7%, and ±2.8% respectively. The fluid temperaturerise along the heated duct is not very high and fluid thermalproperties being well documented, therefore, the uncertainties influid properties variation have been neglected without much lossin accuracy

IV.RESULT & DISCUSSION

A. *Verification of plain duct*

The experimental results of Nusselt number and friction factor for plain ducts are validated. The Nusselt number of plain circular duct is compared with DittiusBoelter equation. The Nusselt number for non-circular ducts is obtained by multiplying Nusselt number of circular duct with a constant that depends on the duct cross section given by Mo Yang el al. [23].

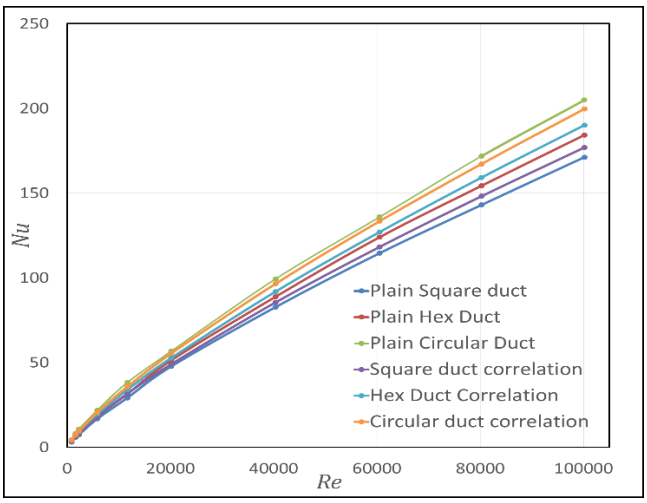
$$Nu = 0.023\, Re^{0.8} Pr^{0.4} \quad (13)$$

$$Nu_{duct} = C_{\varnothing} * Nu \quad (14)$$

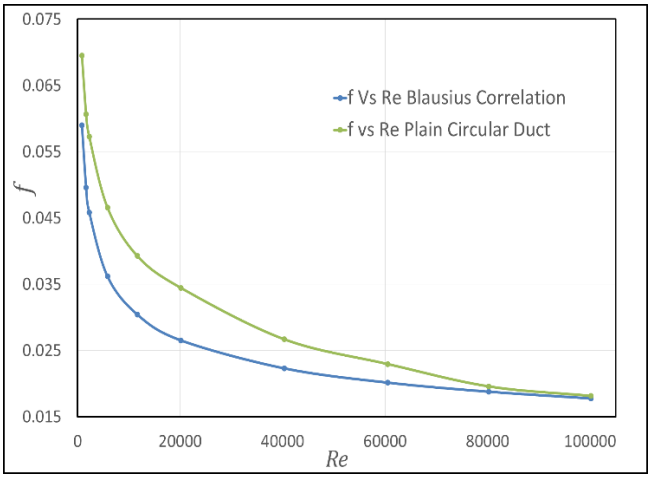
The friction factor of circular duct is compared with the Blausius correlation.

$$f = 0.316\, Re^{-0.25} \quad (15)$$

Fig. 3(a) and 3(b)showscomparison of *Nu* and *f* obtained from present study with the correlations (13, 14, and 15). The results of the present work are within ±10% of the results obtained from DittiusBoelter and Blausius equation.



(a)

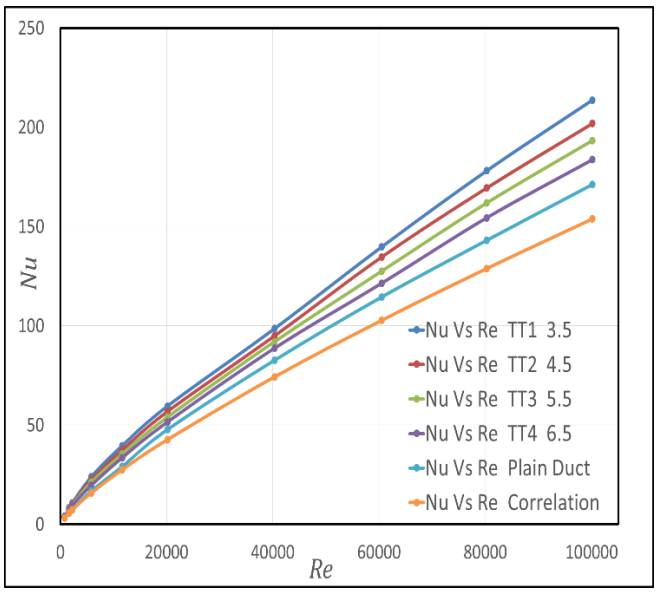


(b)

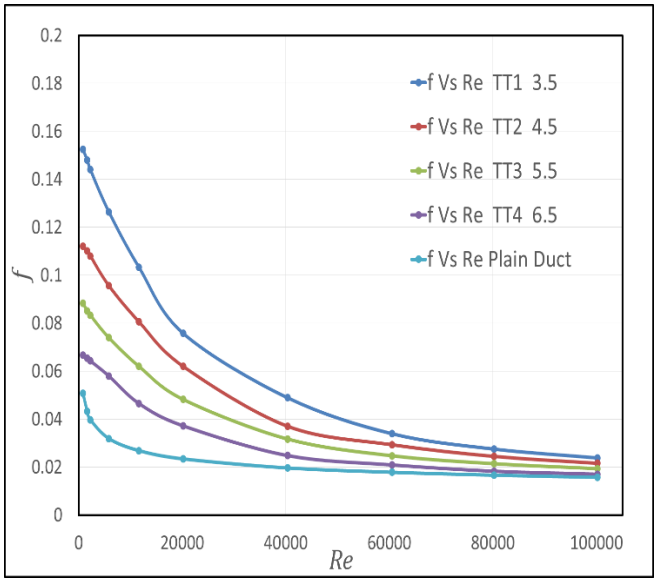
Fig. 3 Verification of (a) Nusselt Number (b) friction factor for plain ducts.

B. Square duct with twisted tape

Fig. 4(a) and 4(b) shows variation of Nusselt number and friction factor with Reynolds for square duct with twisted tape. One can observe that Nusselt number for square duct with twisted tape is higher than plain square duct at any Reynolds number. It is observed that Nusselt number increases as the twist ratio decreases, this is due to better mixing of fluid for lower twist ratio. As the twist ratio increase the swirling motion decreases which leads to poor mixing and reduced heat transfer rates. It is also observed that Nusselt Number increases with Reynolds Number. Highest Nusselt number observed in this case is 213.73 for twist ratio of3.5. The Nusselt number is greater by 24.8%, 17.92% 12.93% and 7.29% than plain square duct for twist ratios of 3.5, 4.5, 5.5 and 6.5 respectively.



(a)



(b)

Fig. 4 Variation of (a) Nusselt Number (b) friction factor with Reynolds number for square duct with TT.

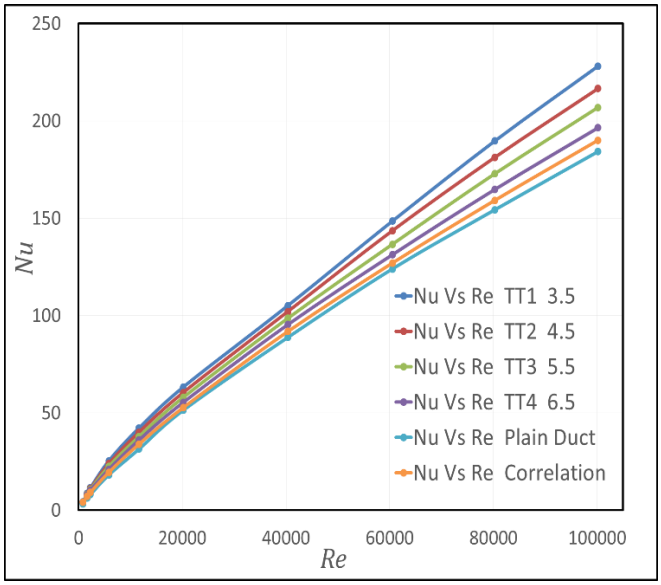
It is also observed that the plain square duct results in lowest pressure drop. The pressure drop increases as the twist ratio decreases. Friction factor is maximum at lowest Reynolds number and it decrease gradually as the Reynolds number increases. Maximum friction factor is observed at twist ratio 3.5 and Reynolds number of 823.

The present results of the Nusselt number (*Nu*) and friction factor (*f*) for the square duct with twisted tape inserts are correlated as follows:

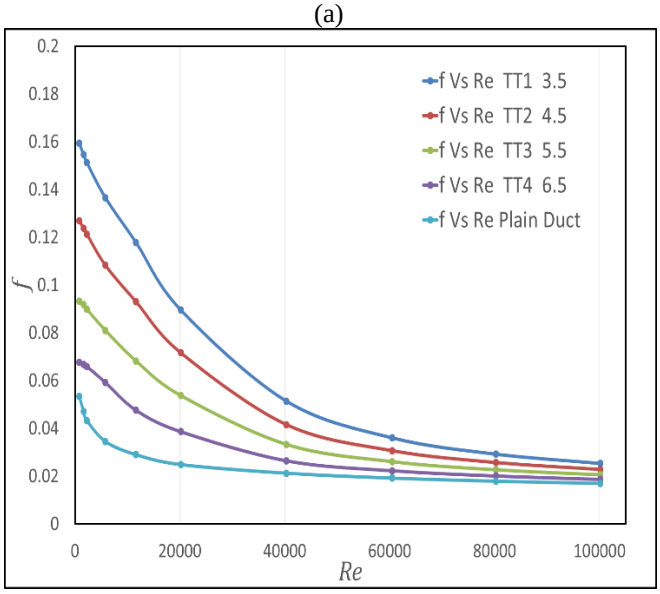
$$Nu = 0.0417\,(Re)^{0.7963}(Pr)\,(Y)^{-0.214}$$
$$f = 6.544\,(Re)^{-0.352}(Y)^{-1.055}$$

C. Hexagonal duct with twisted tape

Fig.5 (a) and 5(b)shows variation of Nusselt number and friction factor with Reynolds for Hexagonal duct with twisted tape. Same trend is observed in case of hexagonal duct i.e. the Nusselt number increases with Reynolds number and decrease in twist ratio. The friction factor decreases as both Reynolds number and twist ratio increases. The maximum Nusselt number observed in this case is 228.14 which is higher than the square duct for the corresponding conditions. This is due to the fact that as the number of sides of non-circular duct increase the blocking effect at corner wall is less which increase the fluid velocity in the corner thereby increasing the heat transfer. However the friction factor of hexagonal duct is more compared to square duct.



(a)



(b)

Fig. 5 Variation of (a) Nusselt Number (b) friction factor with Reynolds number for Hexagonal duct with TT.

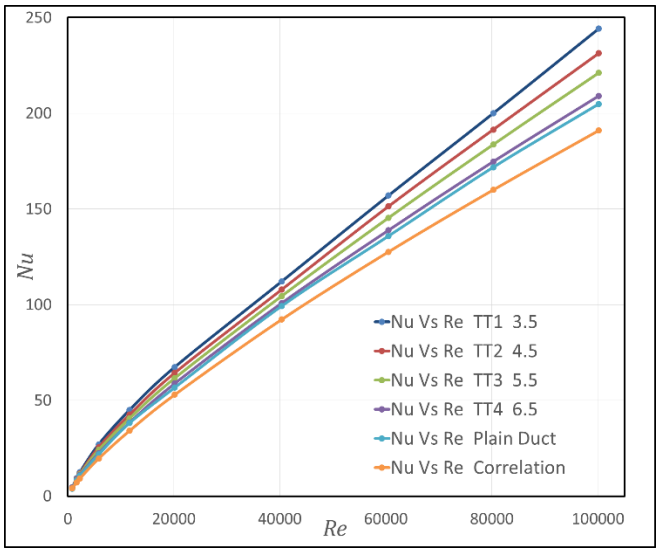
The results of the Nusselt number (*Nu*) and friction factor (*f*) for the hexagonal duct with twisted tape inserts are correlated as follows:

$Nu = 0.042\ (Re)^{0.7974}(Pr)\ (Y)^{-0.204}$

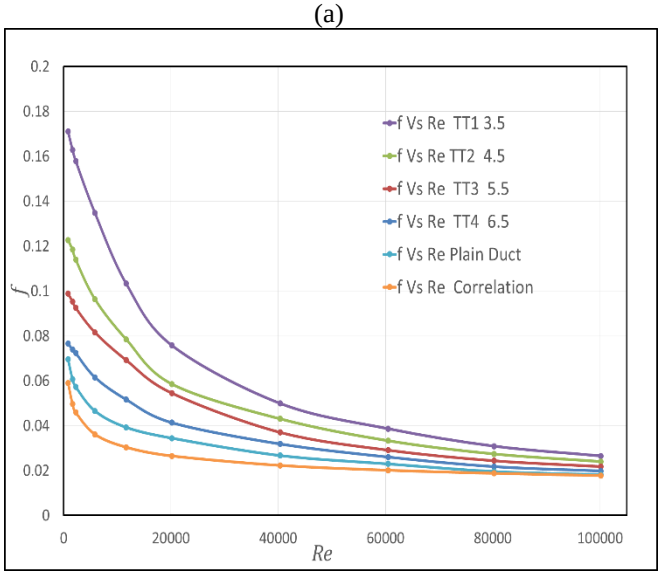
$f = 7.269\ (Re)^{-0.353}(Y)^{-1.115}$

D. Circular duct with twisted tape

Fig. 6(a) and 6(b) shows variation of Nusselt number and friction factor with Reynolds for Circular duct with twisted tape. Similar trend is observed in case of circular duct as well. The Nusselt number increases with Reynolds number and decrease in twist ratio. The friction factor decreases as both Reynolds number and twist ratio increases. Since circle is considered to have infinite number of sides the blocking effect is least among all ducts hence it has highest mean fluid velocity resulting in highest Nusselt number. The maximum Nusselt number observed in this case is 244.2 which is highest among all ducts, but it also results in highest pressure drop. The Nusselt number is greater by 19.4%, 12.92% 7.9% and 2.05% than plain circular duct for twist ratios of 3.5, 4.5, 5.5 and 6.5 respectively.A general trend of Nusselt number and friction factor can be observed. As the number of sides increase the Nusselt number and friction factor increase.



(a)



(b)

Fig. 6 Variation of (a) Nusselt Number (b) friction factor with Reynolds number for Circular duct with TT.

The results of the Nusselt number (*Nu*) and friction factor (*f*) for the circular duct with twisted tape inserts are correlated as follows:

$Nu = 0.0451\ (Re)^{0.7981}(Pr)\ (Y)^{-0.206}$

$f = 6.444\ (Re)^{-0.345}(Y)^{-0.936}$

V.CONCLUSION

The heat transfer and pressure drop characteristics were compared for various ducts. Both pressure drop and heat transfer increase as the twist ratio (Y) decrease. The highest Nusselt number is obtained for twist ratio of 3.5 for each duct. For the same test conditions Nusselt number and friction factor increases as the number of sides increase of non-circular duct. The maximum Nusselt number and pressure drop is observed in case of circular tube.

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NOMENCLATURE

A : Area (m^2)
C_d : Coefficient of discharge
C_p : Specific heat (J/kg. k)
$C_Ø$: Constant for non-circular duct
D : Width of twisted tape (m)
D_h : Hydraulic diameter (m)
D_i : Inner diameter (m)
D_o : Outer diameter (m)
f : Friction factor
g : Gravitation constant (m/s^2)
h : Heat transfer coefficient (W/m^2. K)
H : Manometric column (m)
H : Helical pitch for (180°) rotation of TT (m)
k : Thermal conductivity of air (W/m.k)
L : Length of duct (m)
$\dot{m}$: Mass flow rate (kg/s)
Nu : Nusselt number
Pr : Prandtl number
Q : Heat transfer (W)
Re : Reynolds number
t : Thickness of duct (m)
T : Temperature (k)
V : Velocity (m/s)
Y : Twist ratio (H/D)

Greek Letters

ρ : Density (kg/m^3)
δ : Thickness of TT (m)
μ : Dynamic viscosity (N s/m^2)
v : Kinematic viscosity (m^2/s)

Subscripts

air : Air
bi : Bulk inlet
bm : Bulk mean
bo : Bulk outlet
conv : Convective
cs : Cross section
hc : Heat conducted
w : Water
wm : Wall mean
s : Surface
TT : Twisted tape

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