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Heat transfer and pressure drop in a spirally grooved tube with twisted tape insert

P. Bharadwaj, A.D. Khondge, A.W. Date *

Mechanical Engineering Department, Indian Institute of Technology Bombay, Mumbai 400 076, India

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ABSTRACT

In this paper, experimentally determined pressure drop and heat transfer characteristics of flow of water in a 75-start spirally grooved tube with twisted tape insert are presented. Laminar to fully turbulent ranges of Reynolds numbers have been considered. The grooves are clockwise with respect to the direction of flow. Compared to smooth tube, the heat transfer enhancement due to spiral grooves is further augmented by inserting twisted tapes having twist ratios $Y \simeq 10.15$, 7.95 and 3.4. It is found that the direction of twist (clockwise and anticlockwise) influences the thermo-hydraulic characteristics. Constant pumping power comparisons with smooth tube characteristics show that in spirally grooved tube *with and without* twisted tape, heat transfer increases considerably in laminar and moderately in turbulent range of Reynolds numbers. However, for the bare spiral tube and for spiral tube with anticlockwise twisted tape ($Y = 10.15$), reduction in heat transfer is noticed over a transition range of Reynolds numbers.

1. Introduction

Passive heat transfer enhancement techniques (for example, wall roughness, swirl flow inducement) are preferred over active (for example, surface vibration, electro-static fields) ones to obtain more compact heat exchangers and to reduce energy costs. In this paper, we consider compound heat transfer enhancement due to insertion of a twisted tape in an internally grooved tube.

Compared to a smooth tube, internal spiral grooves (see Fig. 1) impart swirl to the essentially axial flow in the vicinity of the tube wall as well as provide benefits of surface roughness (groove height is typically smaller than the laminar sub-layer thickness in a turbulent flow) [1]. The groove geometry is characterised by groove height e, groove base width t_b and tip width t_t, helix angle α and the number of starts N_s. The axial pitch p_a of the grooves is given by $(\pi D_i/N_s) \times \tan\alpha$ where D_i is the nominal internal diameter of the tube. Webb et al. [1] reported experimental data for $0.024 \leqslant e/D_i \leqslant 0.041$, $18 \leqslant N_s \leqslant 45$, $25 \leqslant \alpha \leqslant 45$ and $t_t/D_i = 0.015$. They developed following correlations for single phase flows ($5.15 \leqslant Pr \leqslant 6.29$)

$$f = 0.108 Re^{-0.283} N_s^{0.221} \alpha^{0.78} \left(\frac{e}{D_i}\right)^{0.785}$$
$$Nu = 0.00933 Re^{0.819} Pr^{0.33} N_s^{0.285} \alpha^{0.505} \left(\frac{e}{D_i}\right)^{0.323} \qquad (1)$$

The correlations are applicable to turbulent range of Reynolds numbers only. These correlations show that compared to a smooth tube, the enhancement in Nu will exceed that in f in respect of number of starts N_s.

Unlike Webb et al. [1], Usui et al. [2] have considered two 75-start spirally grooved copper tubes ($e/D_i = 0.022$ and 0.011, $\alpha = 30°$) as well as a smooth tube. The grooved tubes were inserted (see Fig. 2) with clockwise and anticlockwise twisted tape with twist ratio $Y = H/D_i = 4.2$ and 7.8. The experiments were again performed in the turbulent range of Reynolds numbers only. They used aqueous solution of Glycerine and the Prandtl number was varied from 5.4 to 46.6. They reported that both Nu and f for anticlockwise tapes were higher than those for clockwise tapes for both values of Y at all Prandtl numbers. Usui et al. [3] analysed the data further and defined a performance enhancement parameter

$$i_E = \frac{(Nu/Nu_{sm})}{(f/f_{sm})^{0.291}} \qquad (2)$$

The analysis showed that $1 \leqslant i_E \leqslant 1.2$ when only a twisted tape was inserted in a smooth tube. But, in a grooved tube with clockwise tape (twist direction aiding the swirl induced by the groove) i_E increased from about 1.2 at $Re_h = 1000$ to nearly 1.8 at $Re_h > 10^4$. In this range of Re, for an anticlockwise twisted tape however, i_E was found to be nearly 3.5. Thus, the overall performance of a grooved tube with a twisted tape insert was shown to be significantly higher than that of a smooth tube. Unfortunately, Usui et al. have not presented raw f and Nu-data for each Pr, Y and e/D in their paper nor are any correlations presented.

There are many other related contributions in which twisted tapes are inserted in *corrugated* [4,5] or *spirally fluted* tubes [6] in

* Corresponding author.
E-mail address: awdate@me.iitb.ac.in (A.W. Date).

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Nomenclature

Cp	specific heat
D	tube diameter
e	groove height
f	fanning friction factor
h	enthalpy or heat transfer coefficient
k	thermal conductivity
L	length of test section
$\dot{m}$	mass flow rate
Nu	Nusselt number
N_s	number of starts
Pr	Prandtl number
Q	volume flow rate or heat input
q	heat flux
R	heat transfer gain ratio or electrical resistance
Re	Reynolds number
T	temperature
t_b, t_t	Groove base and tip dimensions
U	velocity
V	voltage
Y	twist ratio
Greek symbols	
α	spiral angle
Δ	incremental value
μ	dynamic viscosity
ρ	density
Suffixes	
b	refers to bulk value
e	refers to exit condition
en	refers to enhanced surface
eq	refers to equivalent resistance or diameter
i	refers to inner diameter or inlet value
o	refers to outer diameter
sm	refers to smooth tube
w	refers to wall

which both single phase as well as flow-boiling and flow-condensation are considered. These are not reviewed here.

The present contribution is similar to that of Usui et al. [2] but differs in three respects:

1. Characteristics of both laminar and turbulent flows are measured.
2. Measurements are restricted to water only ($Pr \simeq 5.4$).
3. Twist ratios considered are $Y = 10.15$, 7.95 and 3.4.

Section 2 describes the experimental set-up and the procedure. The experimental data are presented in Section 3. Overall performance evaluations are presented in Section 4. Section 5 reports conclusions.

2. Experimental apparatus and procedure

2.1. The set-up

Experiments were conducted in an open loop rig (see Fig. 3) with water as the working fluid. Water is continuously supplied from a ground reservoir to an overhead tank (0.25 m^3 at 2.5 m elevation) by a centrifugal pump (0.5 hp) from where it is drawn to the main line via a regulating valve. Water level in the overhead tank was maintained constant through an overflow line. The overflow of the elevated tank is discharged into the reservoir.[1] The main line is 1 in. dia and then reduced to 1/2 in. using a reducer. Before connecting to the test section, a rubber bellows is used to minimise vibrations and flow pulsations. Further, a 1.2 m long calming section is also provided.

The test section used a 75-start internally grooved copper tube supplied by BLUESTAR Ltd., Bombay ($D_i = 14.808$ mm, $\alpha = 23°$, $e = 0.3048$ mm, groove base $= t_b = 0.265$ mm, groove tip $= t_t = 0.0475$ mm).[2] It is 1040 mm long and 15.875 mm OD. The tube was held between inlet and exit sections by means of MS flanges. Two pressure taps (35 mm long, 4.5 mm OD and 3.0 mm ID copper tubings) 1000 mm apart were soldered to the tube for pressure drop measurement. The exit section is 400 mm long PVC pipe with two riser section for venting air bubbles (if any) and to ensure that the test section was always full with water. The flow rate through the test section is regulated by means of two valves in the exit section. Rubber gaskets are used to eliminate leakages at all joinaries. The horizontalness of the test section is checked by a spirit level.

The volume flow rate Q (lits/s) through the test section is measured by collecting water in a bucket (20 lits in turbulent flow and 5 lits in laminar flow) over a fixed time period measured by a stop watch. The pressure drop through the test section is measured by 30° inclined U-tube manometers having benzile alcohol (density 1042.17 kg/m^3) in the laminar range of Reynolds numbers and carbon tetra chloride (density 1581.38 kg/m^3) in the turbulent range.

The heat transfer experiments were conducted under constant wall heat flux conditions. The pressure tap openings were closed using copper rods and M-seal. The 1 m long test section was first cleaned with light emery paper. 0.25 mm Chromel–Alumel thermocouples beads were soldered into the tube wall (in a 3 mm wide slot) at 16 locations along the length of the test section. The thermocouple wires were covered with insulation. Then, the bare tube was covered with a 0.2 mm thick fibre glass tape to electrically isolate the tube from the heater coil.

The heater coils were now wound with fibre glass insulated 30 SWG nichrome wire having 16.67 Ω/m resistance. The wire is covered with fibre glass insulation 1.8 mm dia. The insulated wire was wound without any spacing between successive turns to nearly mimic constant wall heat flux conditions. The heater was designed to give at least 3 °C temperature rise over 1 m length at the highest Reynolds number (35,000 in the present set-up). This required heat input of nearly 3.75 kW. In order to accommodate such a large heat input and to limit the current to 3 A, the heater was divided into three sections. Each section has 2 parallel heating coils (resistance 80 Ω each) connected to one phase of a 3-phase 230 V AC supply. Each heater coil spanned approx. 16 cm length of the test section. The heater wires were connected to copper wires through porcelain connectors of 10 A rating. The copper wires were covered with fibre-glass sleeves and were taken out of the insulation layers provided on the tube.

To prevent heat loss, layered insulation was put on the test section. Firstly, the heater coil was wound on the tube covered with insulation tape. Then, a 35 mm dia asbestos rope was wound on the heater wire. This ensured that the heater coils were firmly in

[1] In case of laminar flow, the test section was directly fed from the centrifugal pump to avoid fluid pulsations.

[2] Note that the spiral angle of the present tube is different from that of Usui et al. [2] who experimented with a tube $D_i \simeq 14.1$ mm and $e = 0.309$ and 0.16 mm. But values of t_b and t_i are not known in their experiment.

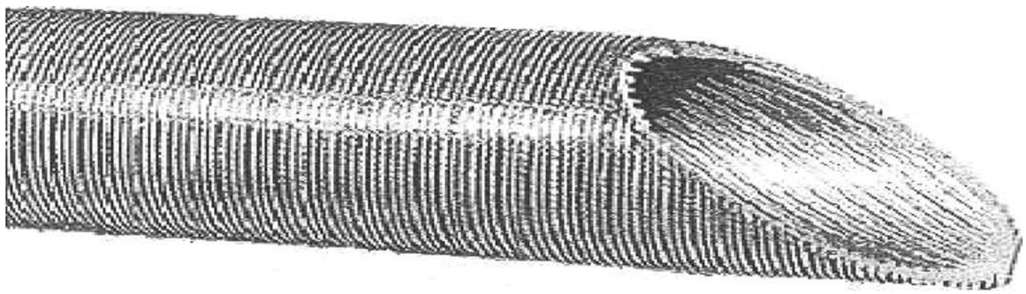

Fig. 1. 75-start grooved tube.

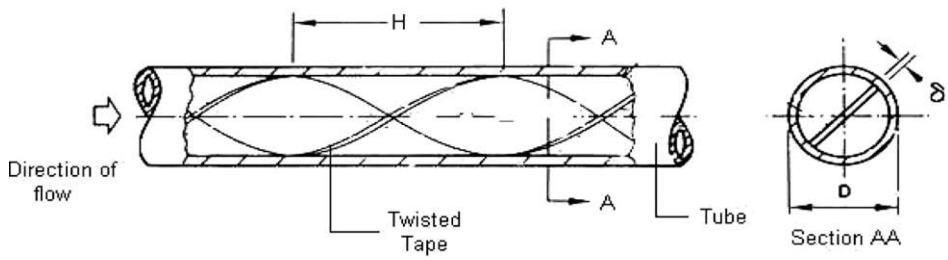

Fig. 2. Twisted tape insert in a tube.

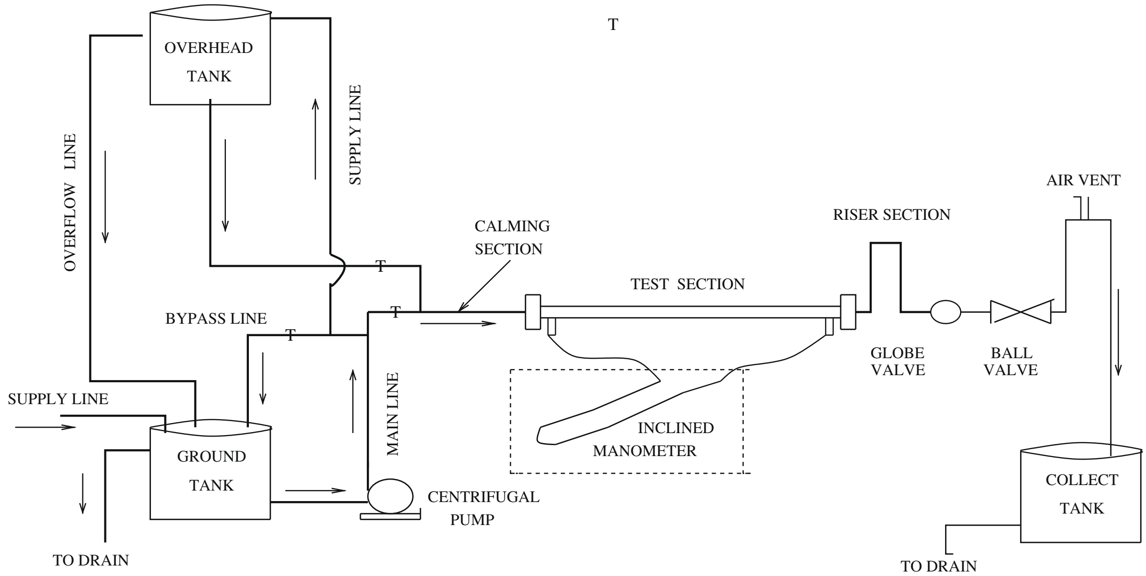

Fig. 3. Open loop layout.

place. Finally, 130 mm thick glass–wool was put and covered with jute cloth. The axial conduction loss from the connecting flanges was prevented by using PTFE spacers.

2.2. Experimental procedure

Pressure drop measurement: In order to evaluate *f–Re* relationship for the grooved tube with and without twisted tape, the pressure drop Δp and average velocity U were measured. The friction factor is defined as

$$f = \frac{1}{2}\frac{\Delta p}{L}\frac{D_i}{\rho U^2} \tag{3}$$

where the average velocity was evaluated from $U = Q/A_c$ and nominal cross-sectional area $A_c = \pi D_i^2/4.0$. In order to ensure complete steady state, 15–20 min were allowed to elapse when the flow rate was changed. The Reynolds number is defined based on nominal diameter as

$$Re = \frac{\rho U D_i}{\mu} \tag{4}$$

Heat transfer measurement: Before starting the heat transfer measurements, the thermocouples were calibrated using ice (0 mV) and water boiling points (4.27 mV). Zero errors for each thermocouple were established from the linear fit. Secondly, to facilitate evaluation of the heat loss through the insulation, the test section was first isolated from the water line and closed at both ends. Heat was supplied. It took nearly 5 h to establish steady state. The average outside tube wall temperature was noted. The heat input was taken as the heat loss Q_{loss} associated with the measured wall temperature T_w. This experiment was repeated for different wall temperatures in the range $15 < (T_w - T_{amb}) < 50$ °C. A near-linear fit between Q_{loss} and $T_w - T_{amb}$ was obtained. This loss was accounted while evaluating actual heat input to the flowing water during heat transfer experiments.

The actual heat input to water was taken to be average of following two evaluations. Firstly, from electrical power, heat input was evaluated as

$$Q_{in,1} = \left[\frac{V_1^2}{R_{eq,1}} + \frac{V_2^2}{R_{eq,2}} + \frac{V_3^2}{R_{eq,3}}\right] - Q_{loss} \tag{5}$$

where V is the heater-section voltage, R_{eq} is the equivalent resistance of the heater coil and Q_{loss} is evaluated from calibrations mentioned above. Secondly, heat input was also evaluated from sensible heat gained by water as

$$Q_{in,2} = \dot{m}_w Cp_m (T_{b,e} - T_{b,i}) \tag{6}$$

where $\dot{m}_w$ is the water mass flow rate, Cp_m is the mean specific heat of water evaluated at average (T_m) water bulk temperatures at inlet and exit from the test section. The exit bulk temperature was evaluated under well-mixed conditions by inserting a short twisted tape in the exit section. The actual heat input was evaluated as $Q_{in} = 0.5(Q_{in,1} + Q_{in,2})$.

The length averaged Nusselt number was then evaluated as

$$Nu = \left(\frac{Q_{in}}{\pi D_i L}\right)\left(\frac{D_i}{k_w}\right)\frac{1}{L}\int_0^L \frac{dz}{(T_{w,i} - T_b)_z} \tag{7}$$

where k_w is the water thermal conductivity evaluated at T_m. The water bulk temperature at axial locations $z(T_{b,z})$ was evaluated by linear interpolation between measured $T_{b,i}$ and $T_{b,e}$. The inside tube wall temperature $T_{w,z}$ at axial location z was evaluated from one-dimensional heat transfer as

$$T_{w,i} = T_{w,o} - \left(\frac{Q_{loss}}{\pi D_i L}\right)\left(\frac{D_i}{2k_{cu}}\right)\ln\left(\frac{D_o}{D_i}\right) \tag{8}$$

In the heat transfer experiments, steady state was reached in about 60 min in laminar flow and about 40 min in turbulent flow after each change in water flow rate. At steady state, input voltages V_i, $i = 1,2,3$, $T_{w,o}$ at 16 locations, $T_{b,i}$ and $T_{b,e}$ were noted besides the water flow rate.

The uncertainty analysis was carried out as prescribed by Kline and Mclintock [8]. For f, uncertainty was estimated to be 5% and for Nu, it was 6%.

Finally, the twisted tapes were made from stainless steel strips whose width equalled $D_i - e$. The strips were held in a lathe and twisted in stages. At each stage, after a small twist, the tapes were heated by a blow torch and mildly stretched. The strips were kept in the twisted position for about 6 h before the next stage was undertaken. Clockwise and anticlockwise twisted tapes were made separately.

3. Results

3.1. Grooved tube without twisted tape

Fig. 4 shows the measured f and *Nu-data* for the present 75-start spirally grooved tube with $D_i/e = 14.808/0.3048 = 48.583$. The f-data are correlated as

$$f = 8.2084 Re^{-0.8425} \quad 300 < Re < 3000 \tag{9}$$

$$f = 0.1535 Re^{-0.283} \quad 3000 < Re < 7000 \tag{10}$$

$$f = 2\left[1.45\ln\left(\frac{D_i}{e}\right) + 8.24\right]^{-2.0} \quad Re > 7000 \tag{11}$$

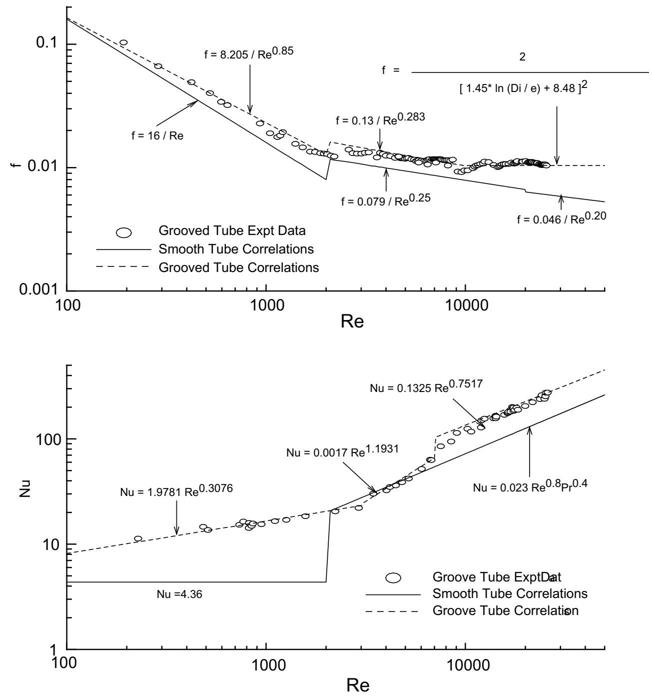

Fig. 4. Characteristics of grooved tube without twisted tape $Pr \simeq 5.42$.

It appears that transition-like characteristics begin at $Re \simeq 3000$ and continue up to $Re = 7000$. Beyond this value of Re, f remains nearly constant which is characteristic of a flow in a rough tube. Hence, following Kays and Crawford [9], for $Re > 7000$, f is correlated as in Eq. (11). Unlike correlation (1), this correlation does not exhibit dependence on Re.

The Nu-data (for $Pr \simeq 5.4$) are correlated as[3]

$$Nu = \frac{1.2(f/2)RePr}{1 + \sqrt{(f/2)}\left[11.90(e^+)^{0.2}Pr^{0.44} - 22.54\right]} \tag{12}$$

where $e^+ = (e/D_i)Re\sqrt{f/2}$. The value of f is taken from Eq. (11).

$$Nu = 1.9781 Re^{0.3076} 300 < Re < 3000 \tag{13}$$

$$Nu = 0.0017 Re^{1.1931} \quad 3000 < Re < 7000 \tag{14}$$

$$Nu = 0.1325 Re^{0.7517} \quad Re > 7000 \tag{15}$$

In the laminar and turbulent ranges of Re, Nu is almost doubled compared to its value in a smooth tube. But, for $3000 < Re < 7000$, the Nu-data almost coincide with the smooth-tube correlation (see bottom Fig. 4) indicating no enhancement in heat transfer in this range. This suggests extremely complex heat and momentum interactions in the vicinity of the tube wall at different Reynolds numbers.

3.2. Grooved tube with twisted tape

In Fig. 5, experimental f-data for clockwise ($Y = 10.18$, 7.98, 3.46) and anticlockwise ($Y = 10.15$, 7.92, 3.33) twisted tapes are compared with the correlations for the grooved tube without twisted tape. The grooves are clockwise. It is seen that the friction factor increases with decrease in twist ratio (tighter the twist, lower is the twist ratio Y) and increase in Reynolds numbers. But, there is no abrupt change in f in the transitional range

[3] For $Re > 7000$, using analogy between heat and momentum transfer, following correlation can be derived to extend applicability for other Prandtl numbers.

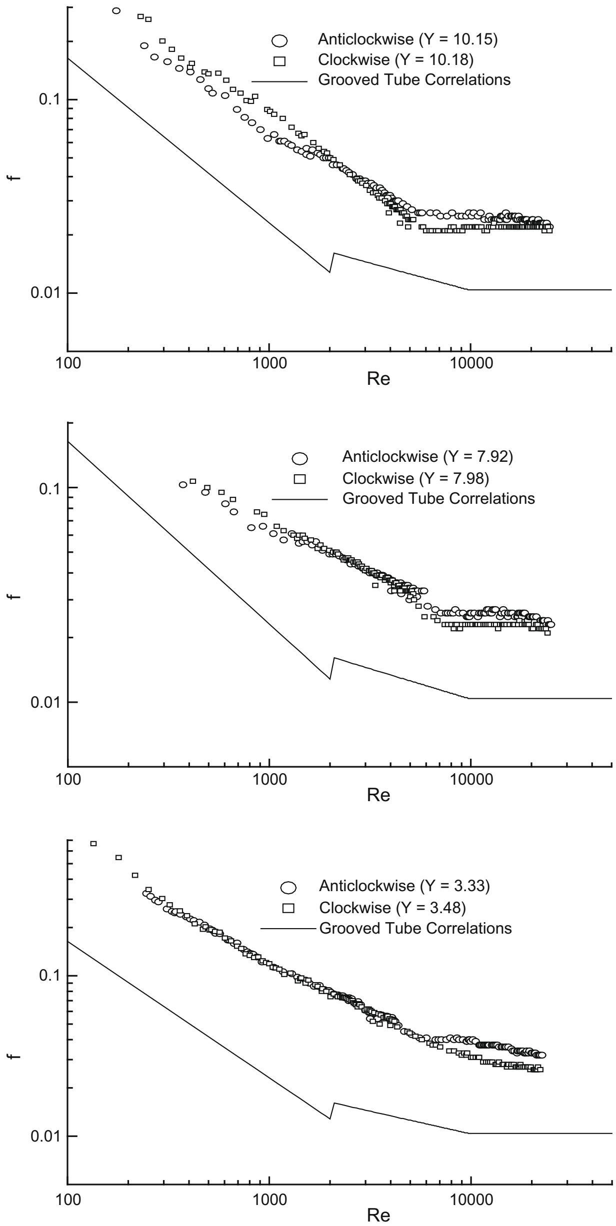

Fig. 5. Friction factor for grooved tube with twisted tape.

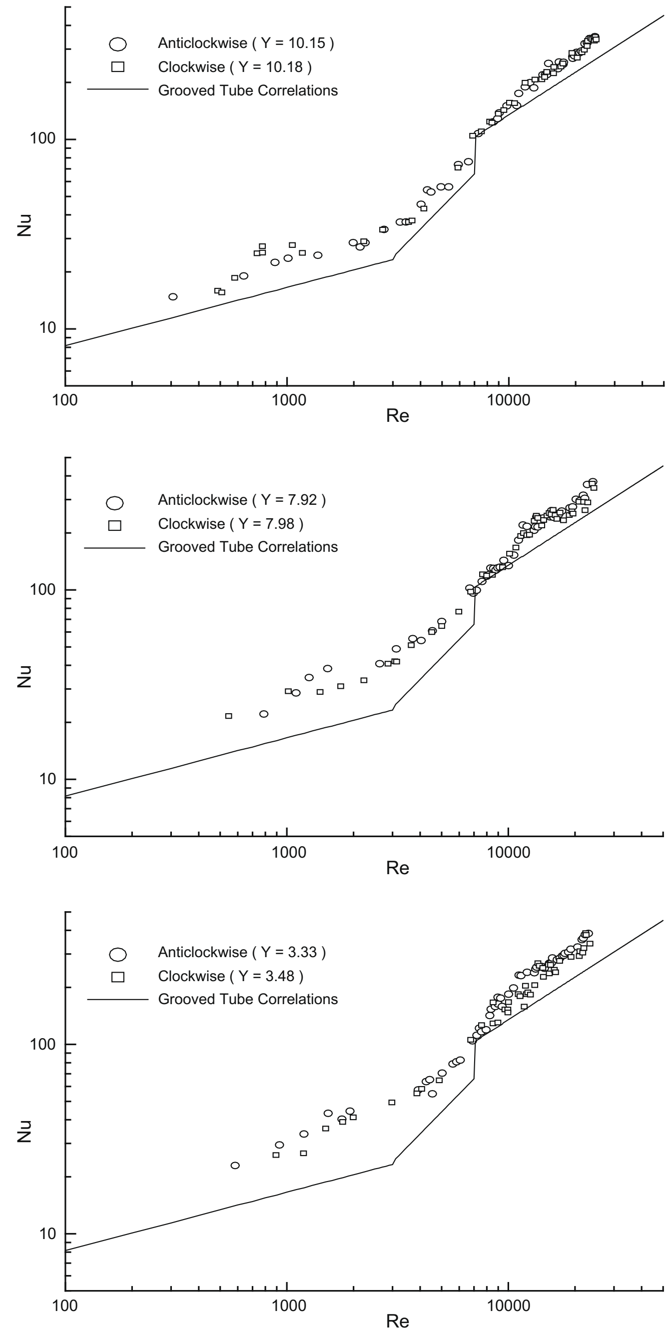

Fig. 6. Nusselt number for grooved tube with twisted tape $Pr \simeq 5.4$.

of Reynolds numbers. This is very characteristic of twisted tape flow [10].

Most notably, however, in the laminar range for $Y = 10.16$, f values for clockwise twist (aiding swirls) are somewhat higher than those for the anticlockwise twist (opposing swirls). At $Y = 7.95$ and 3.4 (tighter twists), the f-data in the laminar range for clockwise and anticlockwise twists nearly coincide. In the turbulent range of Reynolds numbers, however, the f values for anticlockwise twist exceed those for clockwise twist at all twist ratios. The trend in the turbulent range thus accords with the measurements of Usui et al. [2]. Opposing swirls induce greater mixing resulting in higher friction factors or pressure drops. Note also that for $Re > 7000$, the f-data for lower twists $Y = 10.14$ and $Y = 7.95$ show no variation with Re suggesting a rough tube-like behaviour. However at $Y = 3.5$, influence of Re is seen suggesting that the swirl effect overrides the roughness effect.

A similar comparison of the measured Nusselt numbers is shown in Fig. 6. In the laminar range, the Nu values for clockwise twist exceed those for anticlockwise twist for $Y = 10.16$. At $Y = 7.95$ and 3.4, they are nearly coincident. These trends are similar to those observed for friction factor data. Further, a transition-like abrupt change in Nu is observed at $Re = 7000$.

In the turbulent range, however, like the f-, the Nu-data for anticlockwise tapes somewhat exceed those for clockwise tapes irrespective of the value of Y as would be expected. However, the increase in Nu with reduction in Y is not as great as suggested by Usui et al. via the i_E parameter.[4]

[4] The paper by Usui et al. has a plot of f and Nu-data for $Y = 4.2$. Their definitions of f, Nu and Re are based on equivalent diameter whose value cannot be ascertained from the information given in the paper and, therefore, direct comparison of the present data with their data is not possible. None-the-less, a cursory estimate suggests that their f_{clock}, $f_{anticlock}$ and Nu_{clock} data do corroborate present measurements but their $Nu_{anticlock}$ data are higher than the present data by nearly 60%. Also, their $Nu_{anticlock}$ data do not support possible analogy between heat and momentum transfer to the same extent as indicated in their clockwise data. In the present data, the analogy is preserved in both directions of the twist.

Table 1
f-correlations – spirally grooved tube with twisted tape.

Y	Re-range	Clockwise twist	Anticlockwise twist
10.16	<7000	$f = 15.88\ Re^{-0.7597}$	$f = 8.029\ Re^{-0.6467}$
	>7000	$f = 0.021\ Re^{0.0}$	$f = 0.0614\ Re^{-0.1015}$
7.95	<7000	$f = 2.7394\ Re^{-0.537}$	$f = 1.636\ Re^{-0.4671}$
	>7000	$f = 0.0527\ Re^{-0.0912}$	$f = 0.0617\ Re^{-0.097}$
3.4	<7000	$f = 24.84\ Re^{-0.7375}$	$f = 10.274\ Re^{-0.6267}$
	> 7000	$f = 0.418\ Re^{-0.2847}$	$f = 0.2175\ Re^{-0.1927}$

Table 2
Nu-correlations – spirally grooved tube with twisted tape.

Y	Re-range	Clockwise twist	Anticlockwise twist
10.16	<7000	$Nu = 0.1949\ Re^{0.71145}$	$Nu = 0.6916\ Re^{0.535}$
	>7000	$Nu = 0.02564\ Re^{0.9375}$	$Nu = 0.01887\ Re^{0.9724}$
7.95	<7000	$Nu = 0.49054\ Re^{0.6005}$	$Nu = 0.2458\ Re^{0.6749}$
	>7000	$Nu = 0.02237\ Re^{0.951}$	$Nu = 0.00625\ Re^{1.09}$
3.4	<7000	$Nu = 0.2365\ Re^{0.692}$	$Nu = 0.45844\ Re^{0.6143}$
	>7000	$Nu = 0.0496\ Re^{0.8782}$	$Nu = 0.0085\ Re^{1.0677}$

The above observations again suggest that the groove geometry plays a critical role in heat transfer enhancement due to very complex interactions of the flow and temperature fields in the vicinity of the grooves. Clearly more extensive data with systematic variations of groove geometry and Y coupled with semi-empirical or CFD analysis are needed to elaborate further. The fluid dynamics are further complicated by the fact that the tape-edges make only a point-contact with groove tip and therefore fluid transfer past the tape-edge may provide an accelerated flow regime near the groove surface.

3.3. Correlations

The tendencies exhibited by the f- and Nu-data are highly non-linear. Hence, correlations are developed for each value of Y over two ranges of Reynolds numbers and for the two directions of twist. Tables 1 and 2 give correlations for f- and Nu-data, respectively. The correlations predict 90% of the f-data within ±10% and Nu-data within ±15%.

4. Performance evaluation

In order to assess the heat transfer augmentation performance of the 75-start spirally grooved tube (with and without twisted tape) with smooth tube, a *constant pumping power* comparison is made. For this purpose, the heat transfer gain (R) is defined as [7]

$$R = \frac{Nu_{en}}{Nu_{sm}} = F(Re_{sm}) \tag{16}$$

where suffix *en* refers to the enhanced tube and *sm* refers to the smooth tube, Then, for the same pumping power, it can be shown that

$$Re_{en} = Re_{sm}\left[\frac{A_{sm}f_{sm}}{A_{en}f_{en}}\right]^{1/3} \tag{17}$$

where A refers to the surface area. For the present 75-start spirally grooved tube $A_{en} = 1.9142A_{sm}$. From Table 1, $f_{en} = a/Re_{en}^{b}$, so that Eq. (17) reduces to

$$Re_{en} = \left[0.8Re_{sm}\left\{\frac{f_{sm}}{a}\right\}^{1/3}\right]^{1/(1-b/3.0)} \tag{18}$$

where values of f_{sm} are evaluated from $f_{sm} = 16/Re_{sm}$ for $Re_{sm} < 2000$, $f_{sm} = 0.079Re_{sm}^{-0.25}$ for $2000 < Re_{sm} < 20{,}000$ and $f_{sm} = 0.046Re_{sm}^{-0.2}$ for $Re_{sm} > 20{,}000$. For a given Re_{sm}, Re_{en} is evaluated from Eq. (18) and then Nu_{en} is evaluated from Table 2. To evaluate R, smooth tube Nu values are calculated from $Nu_{sm} = 4.36$ for $Re_{sm} < 2000$ and $Nu_{sm} = 0.023Re_{sm}^{0.8}Pr^{0.4}$ for $Re_{sm} > 2000$.

The computed values of R for a spiral tube *without* twisted tape are plotted in Fig. 7. It is seen that heat transfer gain approaches nearly 400% in the laminar range. But for $2500 < Re_{sm} < 9000$, there is reduced heat transfer. For $Re_{sm} > 10{,}000$, however, there is again a gain in heat transfer which reduces from 150% to 133% with increase in Re_{sm}.

Fig. 8 shows variation of R for spirally grooved tube *with* twisted tape. The top figure shows results in the laminar range of Reynolds numbers. It is again seen that for all twist ratios, there is heat transfer enhancement which varies from 160% to 600%. At $Y = 10.14$ and 3.4, anticlockwise twisted tapes perform better than clockwise tape. But, at $Y = 7.95$, the clockwise twisted tape per-

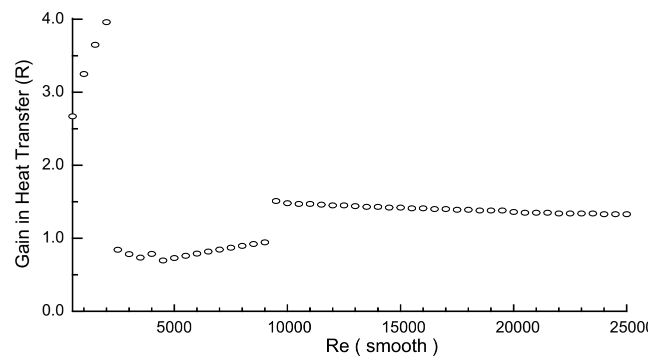

Fig. 7. Heat transfer gain for spirally grooved tube.

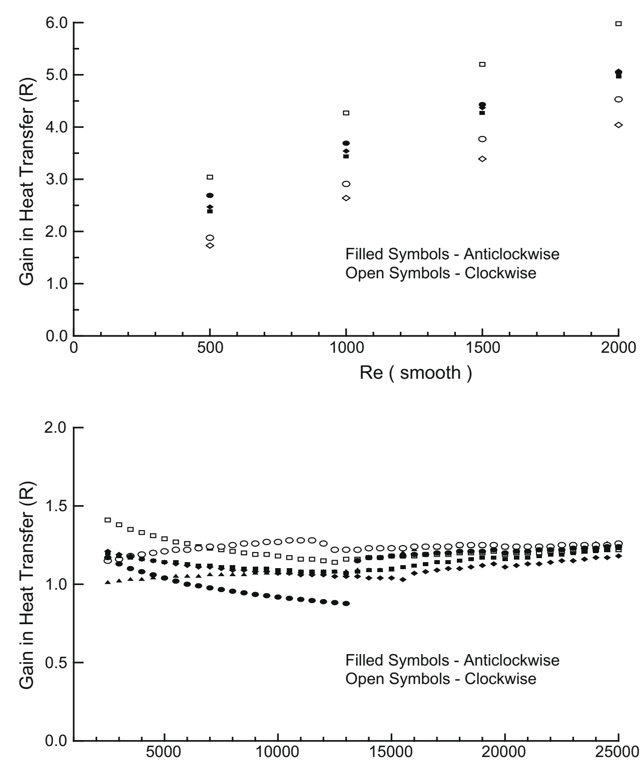

Fig. 8. Heat transfer gain for grooved tube with twisted tape: $Y = 10.15$ (circles). $Y = 7.95$ (squares), $Y = 3.4$ (triangles).

forms better than anticlockwise twisted tape. This intriguing result stems from highly non-linear variations of the *f*- and *Nu-data* in respect of twist ratio *Y*.

The bottom figure shows variation of *R* in the turbulent range. Like the spiral tube without a twisted tape, at $Y = 10.14$ (lowest twist) again deterioration in heat transfer is observed for $6000 < Re_{sm} < 13{,}000$ for the anticlockwise twisted tape. In the higher range ($Re_{sm} > 13{,}000$), however, again $R > 1$. For the clockwise twisted tapes high values of *R* are obtained at $Y = 7.95$ and 10.14. In general, the clockwise twisted tape performs better than the anticlockwise tape at all Reynolds numbers and *Y*. Overall, at lower end of turbulent Reynolds numbers ($Re_{sm} \simeq 3000$), maximum gain is 140% at $Y = 7.95$. At higher end ($Re_{sm} \simeq 25{,}000$) maximum gain of 133% at $Y = 10.15$ and 7.95 is indicated by the evaluations.

5. Conclusion

1. Pressure drop and constant wall heat flux heat transfer measurements in a 75-start spirally grooved tube with and without twisted tapes and with water as working fluid ($Pr \simeq 5.4$) have yielded highly non-linear behaviour of *f* and *Nu* with Reynolds number and twist ratio *Y*.
2. The laminar flow data have been measured for the first time.
3. Constant pumping power comparison with smooth tube shows that the spirally grooved tube *without* twisted tape yields maximum heat transfer enhancement of 400% in the laminar range and 140% in the turbulent range. However, for $2500 < Re_{sm} < 9000$, reduction in heat transfer is noticed. Such a finding has not been reported previously.
4. Similar comparison for spirally grooved tube *with* twisted tape shows maximum enhancement of 600% in the laminar range and 140% in the turbulent range. However, deterioration in heat transfer is observed at $Y = 10.14$ (anticlockwise) for $6000 < Re_{sm} < 13{,}000$.
5. Overall, among the three twist ratios ($Y = 10.15$, 7.95 and 3.4) tested, heat transfer performance of clockwise twisted tape with $Y = 7.95$ is found to be the highest at $Pr \simeq 5.4$ in laminar, transitional and turbulent ranges of Reynolds numbers.
6. The measurements have indicated that flow and heat transfer in a spirally grooved tube are influenced by extremely complex interactions between momentum and heat transfer in the vicinity of the grooved wall resulting in highly non-monotonic and non-linear behaviour at $Pr \simeq 5.4$. Further experimentation and analysis are required to obtain optimum groove geometry and twist ratio *Y* for a given fluid.

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