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Friction and heat transfer characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements

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Abstract

Heat transfer and pressure drop characteristics in a circular tube fitted with regularly spaced twisted-tape elements have been investigated experimentally. Laminar swirl flow of a viscous fluid having intermediate Prandtl number range was considered. The swirl was generated by regularly spaced twisted-tape elements with single twist in the tape module and connected by thin circular rods. The tape-width and the rod diameter were both varied. The effect of higher-than-zero phase angle between consecutive tape elements have been studied. Heated as well as isothermal friction factor data has been generated. The heat transfer test section was heated electrically imposing axially and circumferentially constant wall heat flux (UHF) boundary condition. Reynolds number, Prandtl number, twist ratio, space ratio, tape-width, rod-diameter and phase angle govern the characteristics. 'Pinching' of the tapes in place rather than connecting the tape elements by rods is a better proposition from the thermohydraulic performance point of view. Reducing tape-widths yield poor results. Higher-than-zero phase angle is of no use; rather it increases only manufacturing complexity. The difference of heated friction factor and isothermal friction factor for the periodic swirl flow is substantially less than that in case of straight flow through plain tube.

Keywords: Laminar flow; Regularly spaced twisted-tapes; Swirl flow; Convective heat transfer; Viscous liquid flow

1. Introduction

Heating or cooling of viscous liquids in the process industries, heating or cooling of oils, heating of circulating fluid in solar collectors, heat transfer in compact heat exchangers are examples of laminar flow heat transfer in tubes. Heat transfer augmentation techniques play a vital role for laminar flow heat transfer since the heat transfer coefficients are generally low for laminar flow in plain tubes. Insertion of twisted-tapes in tubes is one such augmentation technique. Fig. 1(a) shows the layout of a full-length twisted-tape insert inside a circular tube. Tape inserts are inexpensive; they can be easily employed to improve the thermal performance of the existing systems. Also, for a given heat load, smaller heat exchangers can be made, thereby reducing capital investments. Twisted-tapes reduce the dominant thermal resistance of the viscous stream and reduce the required heat transfer surface area. However, the thermal improvements are accompanied by increased pressure drop.

Date and Singham [1], Date [2], and Hong and Bergles [3] investigated heat transfer enhancement in laminar, viscous liquid flows in tubes with uniform heat flux (UHF). In the numerical solutions of Date and Singham [1] and Date [2], the flow conditions were idealized for zero tape thickness but the tape twist and

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Nomenclature

A_c flow cross-sectional area, m^2
C_p constant pressure specific heat, J/(kg K)
D internal diameter of the test tube, m
D_h hydraulic diameter of the test tube $[=(4A_c)/P]$, m
d^* rod diameter, m
d non-dimensional rod diameter (d^*/D), dimensionless
f fully developed friction factor based on internal diameter of the tube, Eq. (1), dimensionless
H pitch for 180° rotation of twisted-tape, m
h heat transfer coefficient, $W/(m^2\ K)$
k fluid thermal conductivity, W/(m K)
L length of the test section, m
m mass flow rate of fluid, kg/s
Nu axially averaged Nusselt number based on internal diameter of the tube $[(hD)/k]$, dimensionless
P wetted perimeter in the particular cross-section of the duct, m
ΔP_z pressure drop over a length z, N/m^2
Pr fluid Prandtl number $[(\mu C_p)/k]$, dimensionless
Q heat input to the test section, W
R resistance of the heater element, Ω
Re Reynolds number based on internal diameter of the tube $[(\rho UD)/\mu]$, dimensionless
S gap length between two consecutive twisted-tape elements, m
s space ratio $(=S/D)$, dimensionless
T_{bi} inlet bulk mean temperature of the fluid, °C
T_{bo} outlet bulk mean temperature of the fluid, °C
T_{bz} local bulk mean temperature of the fluid, °C
T_{wz} local tube wall temperature, °C
w^* twisted-tape width, m
w non-dimensional twisted-tape width (w^*/D), dimensionless
U fluid mean axial velocity, m/s
y twist ratio $(=H/D)$, dimensionless
V voltage output from the autostat, V
Z axial length, m

Greek symbols
δ tape thickness, m
μ fluid dynamic viscosity, kg/m s
ρ density of the fluid, kg/m^3

fin-effects were included. Date [2] found that the axial velocity profile became flatter with decreasing twist ratio and increasing Reynolds number. He also observed the pronounced effect of Pr and appreciable tape–fin effect on Nu, especially at larger Re and f. Re is not constant for laminar swirl flow, which is in contrast with straight flow. Flow development length for swirl flow was found to be much smaller than that for straight flow. Hong and Bergles [3] observed that the circumferential temperature profile for the swirl flow is related to the tape orientation. The circumferential variation of the tube wall temperature was, however small. Unlike Date's prediction [2], Hong and Bergles [3] did not observe as pronounced an effect of Pr on Nu at High Pr. They have reported a correlation for predicting Nu in fully developed swirl flows, based on their experimental data for water and ethylene glycol in electrically heated tubes. Additional data for UHF

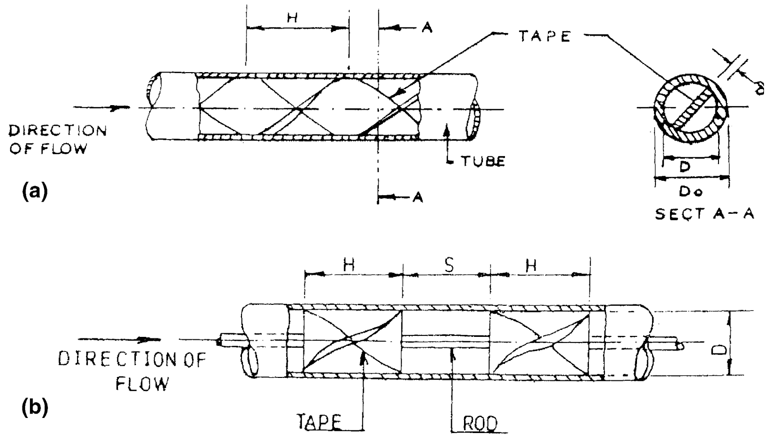


Fig. 1. (a) Layout of a cicular tube containing a full-length twisted-tape. (b) Layout of a circular tube containing regularly spaced twisted-tape elements.

conditions have been reported for laminar flows of air, Watanabe et al. [4] and oil, Bandyopadhyay et al., [5]. Du Plessis [6], van Rooyen and Kroeger [7] observed that, for laminar swirl flow heat transfer in a smooth tube subjected to axially constant tube wall temperature (UWT), the heat transfer rate increases considerably for a moderate increase in pressure drop. Marner and Bergles [8–10] have reported experimental data for laminar flows of ethylene glycol and polybutene 20 with a twisted-tape ($y = 5.39$) in an isothermal tube. Their experimental data over a wide Prandtl number range (1260–8130) are of immense value in the twisted-tape-generated swirl flow heat transfer literature. They found that in very viscous liquid flows, swirl flows do not set in and the heat transfer enhancement is simply due to the duct partitioning. Manglik and Bergles [11–13] made an attempt at developing generalized Nusselt number and friction factor correlations. Manglik and Bergles [14] analyzed numerically the laminar flow heat transfer in a semicircular tube with uniform wall temperature (UWT) defining the lower bound of heat transfer augmentation in circular tubes with twisted-strip inserts. Carlson et al. [15] observed the favorable effect of the twisted-tape inserts for augmenting the heat transfer rate in liquid metal reactor steam generators. Du Plessis and Kroger [16–18] have presented correlations for f and Nu based on their constant property numerical solution for laminar flows. Dasmahapatra and Raja Rao [19,20] have reported experimental data for non-Newtonian liquids. Sukhatme et al. [21,22] reported as high as 44% of total heat transfer due to the tape–fin effect for the laminar flow of a high Pr (about 700) fluid-like servotherm medium oil (the trade name given to the particular variety of oil by the Indian Oil Corporation). They also observed that Nu correlation of Hong and Bergles [3] is applicable for Pr as high as 730.

Very recently Lokanath [23] reported experimental data on laminar flow of water through a horizontal tube under UHF and fitted with half-length twisted-tapes. He found that half-length tapes were more effective than the full-length tapes.

Saha et al. [24] have considered regularly spaced twisted-tape elements connected by thin circular rods as shown in Fig. 1(b). The non-axisymmetric laminar swirl flow generated by the tape element decayed in the free space between the tape elements, only to be again augmented by the tape element that followed. Thus a periodic helical flow was established. They observed that the pressure drop associated with the full-length twisted-tape could be reduced >40% reduction in pumping power) without impairing the heat transfer augmentation rates for $y \leqslant 7.5$ and $s \leqslant 5$ although for these cases f are higher than those for the full-length tapes with same y values. Date and Gaitande [25] developed correlations for predicting characteristics of laminar flow in a tube fitted with regularly spaced twisted-tape elements by solving momentum and energy equations by integral method taking semi-empirical approach. Additional momentum-change losses associated with the developing character of the flow in the annular and twisted-tape sections, even when the flow was periodically fully developed caused this increased pressure drop.

Date and Saha [26] numerically solved the Navier–Stokes and energy equations in their three-dimensional parabolic form and predicted the friction and the heat transfer characteristics for laminar flow in a circular tube fitted with regularly spaced twisted-tape elements that are connected by thin circular rods. The predictions have agreed closely with the experimental data for water. It is shown that a strong length to diameter ratio effect is associated with the configuration considered. It is found that considerably enhanced thermohydraulic performance is achievable by increasing the number of turns on the tape elements, by reducing the connecting rod diameter, and at high fluid Pr.

Monheit [27] evaluated the performance of tapes with surface modifications such as punched holes and plain slit edges vis-a-vis the plain tapes and it needs further study.

Al-Fahed et al. [28] found that the tight-tape fit gives a better performance over the loose-fit tape. Pinjala and Raja Rao [29] proposed a predictive correlation to fit well all the available laminar swirl flow heat transfer results with respect to pseudoplastic type of power-law fluids under constant wall temperature condition. Shivkumar and Raja Rao [30] studied compound augmentation of laminar flow heat transfer to generalized power-law fluids in spirally corrugated tubes by means of twisted-tape inserts. But they did not observe any better results compared to the case of twisted-tape insert in the plain circular tube. Isothermal and non-isothermal friction factors and mean Nusselt number for UWT heating and cooling of Servotherm oil ($Pr = 195$–375) were experimentally determined by Agarwal and Raja Rao [31] for laminar twisted-tape-generated swirl flow in a circular tube. They proposed a correlation representing the effect of heat transfer on friction factors.

The flow field in a rectangular duct behind a row of twisted-tape vortex generators was investigated by Hochdorfer and Gschwind [32] by means of laser-Doppler anemometry (LDA) and a mass transfer photometric measuring method. They observed that the amplitude of the cross-flow velocity distribution describes the downstream damping of vortices.

Aoyama et al. [33,34] studied the buoyancy effect on laminar heat transfer in a horizontal straight tube containing a twisted-tape swirler. They found that, when the buoyancy as well as the tape torsion is significant, the peripherally averaged Nusselt number oscillates along the tube axis in the thermally developing region and the turning pitch of the tape nearly coincides with the period of the oscillation. The trajectories drawn by them on the surface of the axial velocity versus the axial pressure gradient showed the chaos-like fluid motion when the tape-torsion is large.

Manglik and Bergles [35] made a numerical analysis of the fully developed laminar heat transfer in circular-segment ducts with uniform wall temperature and studied the two extremities of the fin effects of a straight-tape insert, i.e., 100% and zero fin efficiencies.

Patil [36] studied the friction and the heat transfer characteristics of laminar swirl flow of pseudoplastic type power-law fluid in a circular tube using varying width full-length twisted-tapes under a uniform wall temperature condition. He found that, from the considerations of enhanced heat transfer and savings in pumping power and in tape material cost, reduced-width tape inserts are attractive for enhancing laminar swirl flow heat transfer. He has observed 17–60% reduction in friction factor and 5–24% reduction in Nusselt number for 15–50% reduction in tape-width.

2. Range of parameters tested

In the present report, laminar flow results from experimental investigations carried out in a circular tube fitted with regularly spaced twisted-tape elements with single twist in the tape elements are presented. For this, a closed-loop experimental facility was made. Servotherm medium oil was used as the working fluid (giving intermediate Pr range). Heat transfer data were obtained in an electrically heated test section giving axially and circumferentially constant wall heat flux (UHF). The phase angle (Φ) between successive tape elements is varied. The tape-width and the rod diameter are also varied. The direction of twist imparted to each successive element is the same.

The fin effect of the tape elements is deliberately suppressed since the fin-effect has not been studied. Both isothermal and heating friction factor data were collected and are presented.

Experiments were conducted over the following ranges of independent parameters:

Reynolds number	$45 < Re < 1150$
Twist ratio (y)	$2.50 \leqslant y \leqslant 5$
Prandtl number (Pr)	$205 < Pr < 518$
Space ratio (s)	$0 \leqslant s \leqslant 5.00$

Here the twist ratio (y) and the space ratio (s) are defined as (H/D) and (S/D), respectively, where H is the pitch for 180° rotation of the twisted-tape, S is the gap between two successive twisted-tape elements in the tape–rod assembly and D is the internal diameter of the test-section tube.

3. Experimental setup

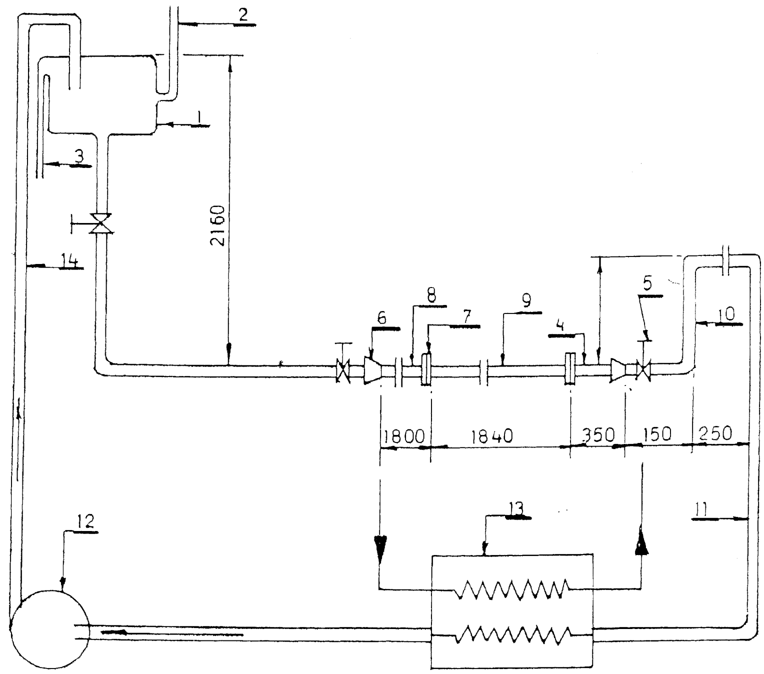
The investigations were carried out in a closed-loop experimental facility as shown in Fig. 2. The loop consisted of:

1. an overhead tank (0.25 m^3 capacity located at an elevation of 2.75 m),
2. a calming section (1.8 m long, 13 mm i.d., 20 mm o.d., PVC tube),
3. the test section (details follow),
4. an insulated mixing section i.e., exit section (13 mm i.d., 20 mm o.d., 350 mm long PVC tube),
5. a riser section with 150 mm high kink,
6. a heat exchanger (1 HP).

The twisted-tapes were made of 0.5 mm thick stainless steel strips, the width of the strip being 1 mm less than the inside diameter of the test section tube for the full-width tape. This situation closely resembles real applications where removable tapes are inserted in existing heat exchanger tubes. Twisted-tapes were made separately for pressure drop tests and heat transfer tests because of different diameters of the test section. For pressure drop tests, the reduced-width tapes had a width (w^*) of 9 and 6 mm. Thus three non-dimensional widths ($w = w^*/D$) were 0.923 ($=12/13$), 0.692 ($=9/13$), 0.461 ($=6/13$). For heat transfer measurements, the reduced-width tapes had a width (w^*) of 8 and 5 mm. Thus three non-dimensional widths ($w = w^*/D$) were 0.909 ($=10/11$), 0.727 ($= 8/11$) and 0.454 ($= 5/11$). The strips were twisted on a lathe by manual rotation of the chuck. Twisted strips were heated periodically by flame to relax the stresses to prevent them from buckling and untwisting when they were removed from the lathe. Twisted-tapes for heat transfer tests were covered with insulated tape to suppress the fin effect since the tape–fin effect was not studied. Pieces of stainless steel rods of 3, 2.5, 2 and 1.5 mm diameter (d^*) were used to make tape–rod assembly. Thus, for pressure drop tests, the non-dimensional rod diameter ($d = d^*/D$) were 0.231 ($= 3/13$), 0.192 ($= 2.5/13$), 0.154 ($= 2/13$), 0.115 ($=1.5/13$) and those for heat transfer tests were 0.273 ($=3/11$), 0.227 ($=2.5/11$), 0.182 ($= 2/11$) and 0.136 ($=1.5/11$). It should be noted here that the rod diameter for both the pressure drop test and the heat transfer test was physically equal. However, the non-dimensional rod diameter for the pressure drop test was different from that for the heat transfer test because of the small difference of the pressure drop test-section tube diameter from the heat transfer test-section tube diameter.

The ends of the rods were slotted to accept the twisted-tape elements. Twisted-tape elements were spot welded to the ends of the rods. The phase angle, ϕ were 0°, 90° and 180°.

A PVC tube of 1.85-m long, 13-mm i.d. and 20 mm o.d. was used as test section for the pressure drop tests. The pressure taps were made of 30-mm long, 3-mm i.d.,



DIMENSIONS ARE IN MM

1. OVERGEAD TANK
2. LEVEL INDICATOR
3. OVER FLOW TO SUMP
4. EXIT SECTION
5. NEEDLE VALVE
6. REDUCER
7. FLANGES
8. CALMING SECTION
9. TEST SECTION
10. RISER
11. FLOW OUTLET TO H.E.
12. GEAR PUMP
13. HEAT EXCHANGER (H.E.)
14. SUPPLY TO TANK

NOT TO SCALE

Fig. 2. The experimental setup.

4.5-mm o.d. acrylic tube. Threads were cut on one end of the taps and drilled holes on the test section were threaded internally. The threaded portion of the taps was covered with Teflon tape before inserting them into the tapped hole to prevent leakage. Additionally, using Araldite ensured the rigidity of the taps. Flexible PVC tubing to a manometer connected the pressure taps at either end of the test section. Pressure drop was measured by means of vertical U-tube manometer with dibutyl phthalate (specific gravity 1.047) and mercury as the manometric fluid for lower *Re* and higher *Re*, respectively. This ensured reasonably good accuracy by producing large-scale reading with dibutyl phthalate for lower *Re*.

The heat transfer test section is shown in Fig. 3. It consisted of 1.84-m long, 11-mm i.d., 12-mm o.d. stainless steel (304 SS) tube. The tube was uniformly heated by passing alternating current from a stabilized three-phase source through 30 SWG fibre glass insulated nichrome wire having 16 Ω/m resistance wound around the tube. The electrical winding was made in such a way that current passed in both directions to cancel the electromagnetic field effects. Electrical resistance was measured in several sections to ensure uniform heat flux. The heater was made in three sections, each section having four parallel heaters. Autotransformers were used to regulate the heat flux supplied to the test section. To measure the outside wall temperature of the tube, 30 SWG copper-constantan thermocouples were used. The thermocouples were silver soldered and were first taken through the circumferential grooves and then radially through the heater wire turns and the insulation put on the test section. Eleven thermocouples were installed on the outside wall of the tube to measure the wall temperature. The first five were placed at 15, 115, 225, 350 and 430 mm from the point where the heating started and the remaining six were placed at 525, 565, 627, 688, 738 and 788 mm from the downstream end. Using polytetrafluoroethylene

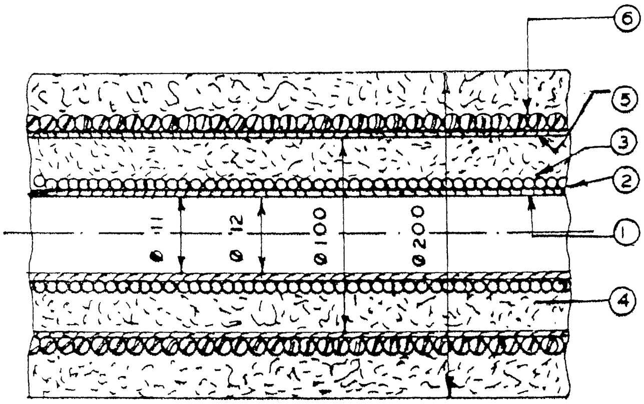


Fig. 3. Heat transfer test section.

spacers positioned between flanges minimized axial conduction losses. Winding asbestos rope and putting a glasswool blanket of 50 mm thickness around the asbestos rope minimized the radial heat losses. The asbestos rope was wound on a GI (Galvanized Iron) cylinder (split in two halves) to protect the electrical connections. The space between the heater wire and the GI cylinder was also filled with glasswool. A precision digital multimeter measured the thermocouple output. The power input was calculated from the measured voltage and the resistance. The current was also measured to provide a crosscheck. Fluid bulk temperature measurements were made at the inlet and the exit of the test section. Wire meshes were used in the exit (mixing) section at the downstream end of the test-section tube. Wire meshes provided thorough mixing of the fluid to give the accurate outlet bulk mean temperature of the fluid. Saha et al. [24] and Hong and Bergles [3] have observed that the circumferential temperature variation for the swirl is very small. Accordingly, no circumferential variation of temperature was measured in the present investigation.

4. Operating procedure and data reduction

4.1. Pressure drop tests

The pressure drop tests were carried out in the PVC tube. The friction factor was defined as

$$f = 1/2[(\Delta P_z)/(\rho U^2)](D/z), \quad (1)$$

where

$$U = \dot{m}/(\rho A_c) \quad (2)$$

and ΔP_z is the pressure drop over a length z.

Before the pressure drop measurements were taken, the test section was freed of air bubbles by venting them through the riser section at the end of the test section. The oil flow was taken from the overhead tank where a constant level was maintained. Typically 15 min were required for settling of flow after each change of mass flow rate. In addition to controlling the mass flow rate, the fluid was heated as necessary to obtain a reasonable range of laminar Reynolds number. The manometer was used to measure pressure drop. The mass flow rate was measured by collecting the fluid for a certain period and a stopwatch. Uncertainty in the mass flow rate measurement was within ±1%. Mass flow rates as measured by weighing were within ±0.5% of that measured by a rotameter. However, rotameter was not used since the available rotameter could not cover the entire range of the mass flow rate. Ideally, two rotameters, covering the entire range of mass flow rates tested, could have been used in the test setup. This would have been the commonly used approach. However, second rotameter was not available and it was decided to measure the mass flow rates by collecting the fluid for a certain period and a stopwatch. This did not affect the accuracy of the experiment.

4.2. Heat transfer tests

The heat transfer tests were performed in the stainless steel tube. Servotherm medium oil provided substantial variation in Pr. The length averaged Nu was defined as

$$Nu = \frac{hD}{k} = \frac{Q}{\pi DL}\left[\int_O^L \frac{\mathrm{d}z}{T_{\mathrm{wz}} - T_{\mathrm{bz}}}\right]\left(\frac{D}{k}\right), \quad (3)$$

where

$$Q = V^2/R = mC_{\mathrm{p}}(T_{\mathrm{bo}} - T_{\mathrm{bi}}). \quad (4)$$

Q was measured by two methods as shown in Eq. (4). The arithmetic mean of two values was taken. Data for variation by more than 5% in two methods were discarded. The wall temperature T_{wz} at any z in Eq. (3) was measured directly, whereas T_{bz} was calculated by interpolation assuming linear variation of fluid bulk temperature along the length of the tube. A thermal steady-state was generally reached in 60–90 min. The minimum and the maximum changes in the fluid viscosity from the inlet to the outlet section were 8% and 26% of the value at the inlet, respectively. Hence all fluid properties were calculated at $(T_{\mathrm{bi}} + T_{\mathrm{bo}})/2$ and functional relationships of properties with temperature available in Bandyopadhyay et al. [5] were used for this purpose. No significant improvement is expected, Hong and Bergles [3], by using the film temperature. The tube wall temperature drop was calculated by considering the steady-state, one-dimensional radial heat conduction equation and this drop was applied uniformly along the length of the tube. The wall temperature and the bulk temperature were combined with the heat flux to give the axially local heat transfer coefficient and the Nusselt number. The axially averaged Nusselt number was obtained by Simpson's rule of numerical integration. The flow was hydrodynamically fully developed and with uniform temperature profile at the entrance of the test section. The inside tube diameter rather than the hydraulic diameter was used in defining Nu and Re for performance analysis and easy comparison. Marner et al. [37] proposed this envelope diameter approach for reporting enhanced tube data. Since the tape inhibits free convection, the Grashof number or Rayleigh number, important parameters in plain tube laminar flow, are unimportant in the present swirl flow $(Gr/Re^2 < 1)$. Further details of the apparatus, procedure and data reduction are available in Halder [38].

An uncertainty analysis conducted along the lines suggested by Kline and McClintock [39] showed that the

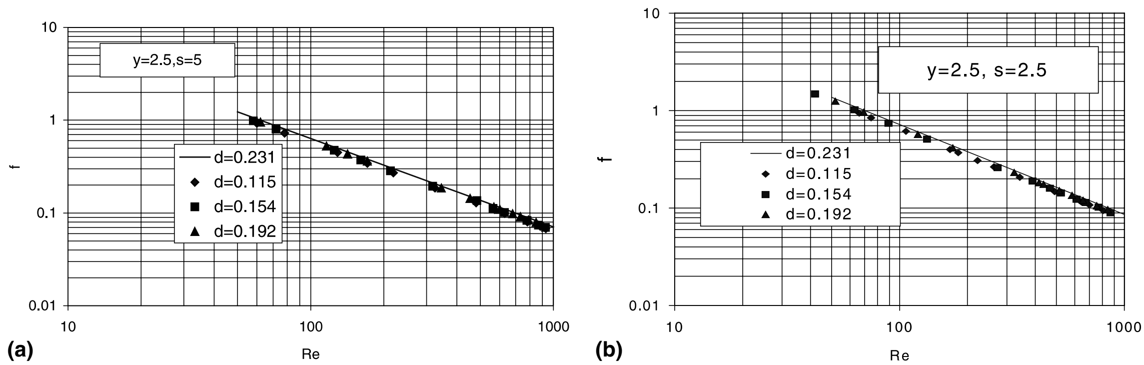


Fig. 4. Variation of friction factor with Reynolds number for regularly spaced twisted-tape elements with different rod diameter. (a) $y = 2.5$, $s = 5$; (b) $y = 2.5$, $s = 2.5$.

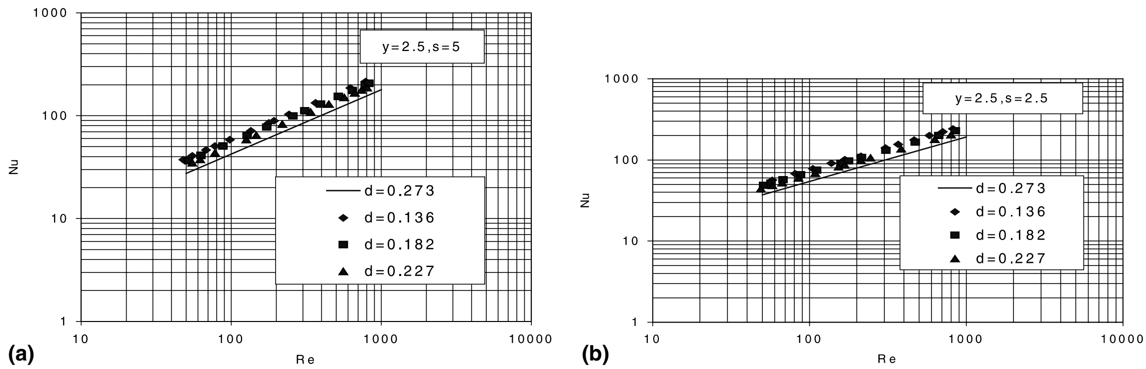


Fig. 5. Variation of Nusselt number with Reynolds number for regularly spaced twisted-tape elements with different rod diameter. (a) $y = 2.5$, $s = 5$; (b) $y = 2.5$, $s = 2.5$.

uncertainties involved in the friction factors were within ±8% whereas those involved in the estimation of Nusselt number and Reynolds number were within ±7% and ±2%, respectively.

5. Results and discussion

The results obtained in this study are presented and discussed in this section. The solid lines in Figs. 4–9 represent the correlations of Saha et al. [24]. Figs. 4, 6 and 8 show the isothermal friction factor data.

5.1. Data for different rod diameters

Figs. 4 and 5 show the variation of friction factor with Reynolds number and variation of Nusselt number with Reynolds number, respectively. For friction factor data $w = 0.923$. For heat transfer data $w = 0.909$. Full-width tape had the non-dimensional width little less than unity for easy entrance and removal of the tapes. This resembles the practical situation. Again, the difference in the value of w for the pressure drop tests from that for the heat transfer tests is due to the difference in test-section tube diameters. For small rod diameter cases, the friction factor is 5–10% less and heat transfer increases

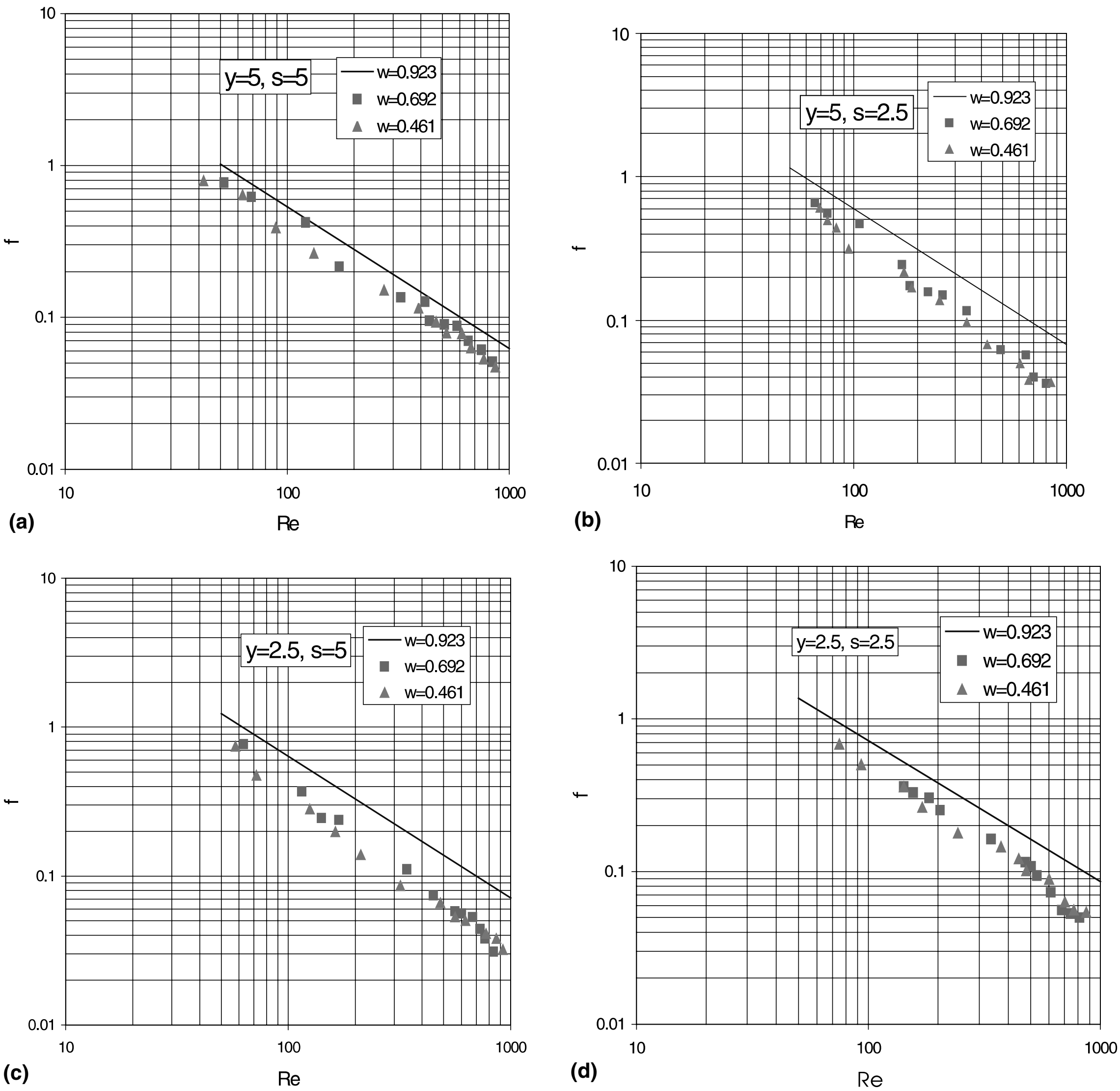


Fig. 6. Variation of friction factor with Reynolds number for regularly spaced twisted-tape elements with varying tape-width. (a) $y = 5$, $s = 5$; (b) $y = 5$, $s = 2.5$; (c) $y = 2.5$, $s = 5$; (d) $y = 2.5$, $s = 2.5$.

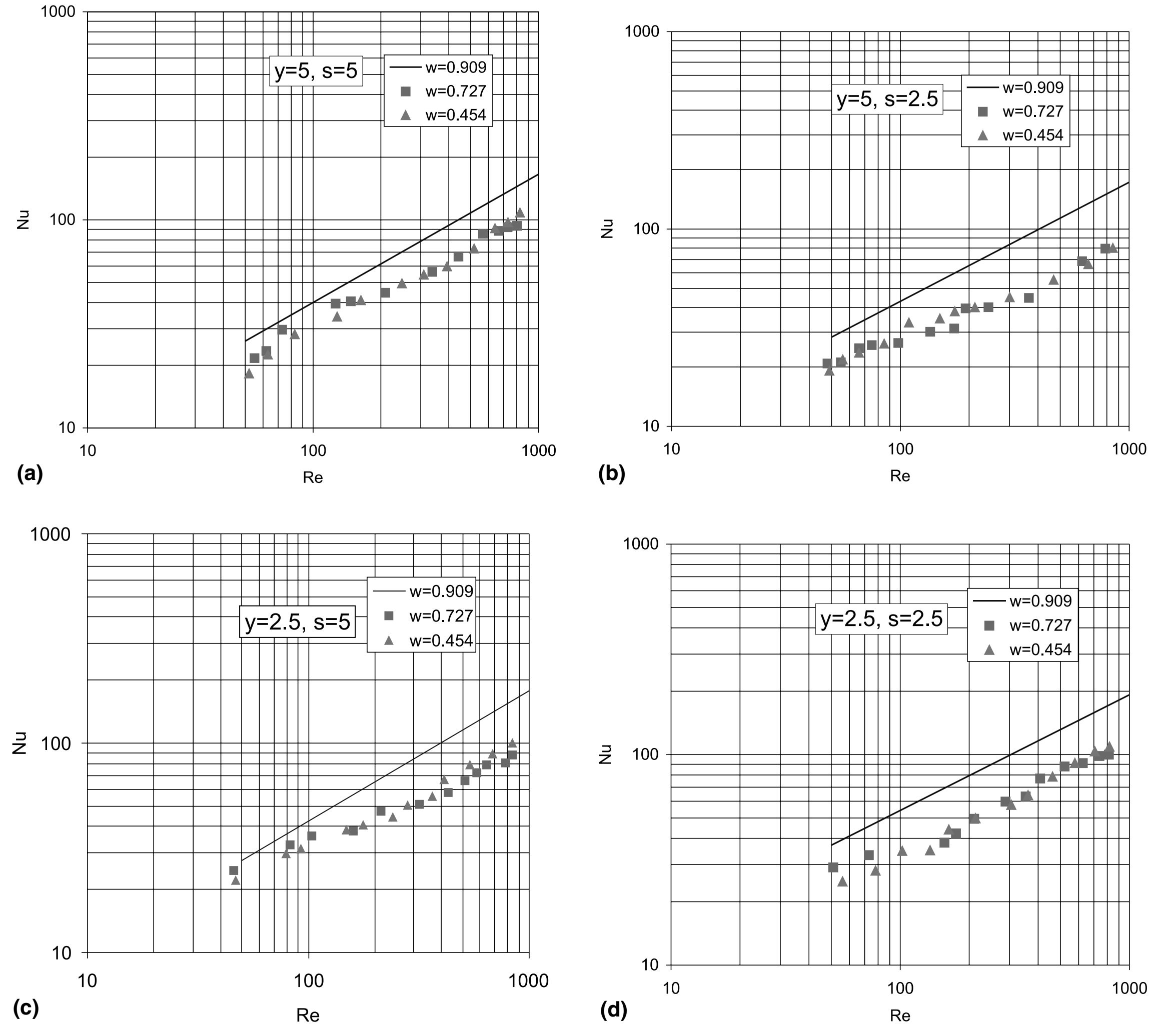


Fig. 7. Variation of Nusselt number with Reynolds number for regularly spaced twisted-tape elements with varying tape-width. (a) $y = 5$, $s = 5$; (b) $y = 5$, $s = 2.5$; (c) $y = 2.5$, $s = 5$; (d) $y = 2.5$, $s = 2.5$.

by 20–40%. The reason is that the local values of Nusselt numbers in the inlet region of the tape section do not show sharp decline, Date and Saha [26]. In fact the local Nusselt numbers remain almost constant in both the tape and the annular sections. Also, for the small diameter cases, negative axial velocities were not predicted even at the first axial step in the tape section, and thus the swirl was not suppressed. A small rod diameter thus has a favorable overall effect. Hence doing away with the connecting rod and locating the tape elements in place probably by 'pinching' the tube is desirable.

5.2. Data for different tape-width

Fig. 6 shows the variation of friction factor with Reynolds number for varied y and s and for each case with $w = 0.692$ and $w = 0.461$. For all data $d = 0.231$. Tapes with twist ratio, $y > 5$ do not generate sufficient swirl and were not considered. The gap length giving $s > 5$ is not sufficient to maintain the swirl intensity to the sufficient extent downstream of the tape. The swirl decays fast and subsequently the straight flow ensues in the gap before the fluid finds the next tape element on its way for the swirl to build up again. This yields lower Nu characteristic of the laminar straight flow. This Nu is much lower than that for the swirl flow. Therefore, s was restricted to less than or equal to five only. The experimental data are compared to the correlation of regularly spaced twisted-tape elements with $w = 0.923$ available in Saha et al. [24]. It is observed from the figures that f decreases for $w = 0.692$ compared to that for $w = 0.923$ for all y and s. For $y = 5$ and $s = 5$, the reduction is

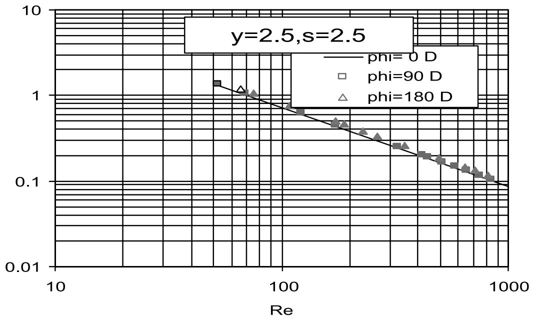


Fig. 8. Variation of friction factor with Reynolds number for regularly spaced twisted-tape elements with varying phase angle between successive tape elements.

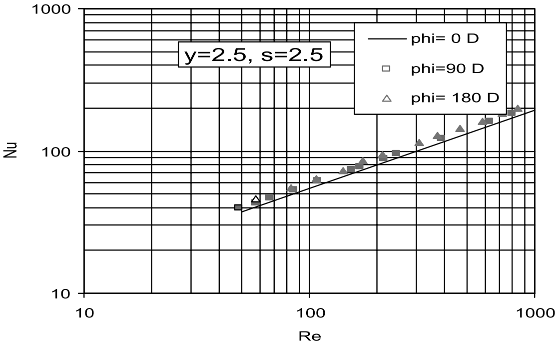


Fig. 9. Variation of Nusselt number with Reynolds number for regularly spaced twisted-tape elements with varying phase angle between successive tape elements.

18–20%; for $y = 5$ and $s = 2.5$, the reduction is 28–32%, for $y = 2.5$ and $s = 5$, the reduction is 11–45% and for $y = 2.5$ and $s = 2.5$, the reduction is 25–35%. Saha et al. [24] have observed that some particular combinations of y and s (here $y = 2.5$ and $s = 5.0$) do show a striking effect of Reynolds number. In fact, each combination of y and s represents particular fin-geometry, which is very different from the others. The reduction in friction factor increases as the Reynolds number increases. Friction factor, however, does not reduce much (<6%) for $w = 0.461$ from $w = 0.692$ for any combination of y and s and at any Re. The reason for decrease in the friction factor from $w = 0.923$ to $w = 0.692$ is the less amount of tape surface encountered for frictional pressure drop. However, for $w = 0.461$, the pressure loss associated with the momentum change that occurs when two non-axisymmetric velocity profiles issuing from the twisted-tape module mix in the decaying swirl section in the space module is much greater than the pressure loss that would have been encountered for $w = 0.692$. This can be appreciated from the fact that there is more prominent effect of mixing phenomena for $w = 0.461$ compared to the case of $w = 0.692$ even though for the former there is less tape-surface encountered for frictional pressure drop. The experimental results can, therefore, be expected and the friction factor does not decrease much for $w = 0.461$ compared to that for $w = 0.692$.

Fig. 7 shows the variation of Nu with Re for $y = 5$ and $s = 5$, $y = 5$ and $s = 2.5$, $y = 2.5$ and $s = 5$, $y = 2.5$ and $s = 2.5$. For all data $d = 0.273$. The experimental data are compared with the correlation of regularly spaced twisted-tape elements with $w = 0.909$ available in Saha et al. [24]. It is seen from the figures that Nusselt number decreases for $w = 0.727$ compared to that for $w = 0.909$ for all twist ratio and space ratio. For $y = 5$ and $s = 5$, the reduction is 25–32%; for $y = 5$ and $s = 2.5$, the reduction is 25–45%; for $y = 2.5$ and $s = 5$, the reduction is 22–45%; and for $y = 2.5$ and $s = 2.5$, the reduction is 13–43%. Reduction in the Nusselt number generally increases as the Reynolds number increases. Date and Saha [26] observed a considerable reduction in local Nusselt number at the inlet to the tape module and a sudden rise in local Nusselt number at the inlet to the space module. Because the fluid bulk mean temperature increases linearly axially, the peculiar variations in local Nusselt number are associated with the local values of tube wall temperature. This is explained as follows:

The flow in the twisted-tape section is non-axisymmetric. At the inlet to the space module the fluid jumps on the connecting rod due to swirl. This effect, coupled with the tendency of the fluid to attain circumferential symmetry in the space between two consecutive tape elements, gives high velocity near the tube wall. This reduces wall temperature suddenly giving high local Nusselt number. This gives higher axially averaged Nusselt number than that in case of straight annular flow. It should be noted here that the flow in the gap between two consecutive tape elements resembles an annular flow. At the inlet to the tape module, the flow enters with a weak swirl and accelerates rapidly near the axis and the fluid jumps from the rod suppressing the swirl. This reduces flow velocities near the tube wall causing higher tube wall temperature and lower local Nusselt number. Thus axially averaged Nusselt number in the tape module is substantially less than that in case of full-length tape. This effect is more transparently visible in case of $w = 0.727$ than in case of $w = 0.909$ because of more mixing phenomena. Thus in the case of $w = 0.727$ we get lower axially averaged Nusselt number compared to $w = 0.909$. But for $w = 0.454$, swirl does not decay substantially and the swirl subsists further downstream of the tape module and the velocity and the temperature profiles do not become shallower to give the still lower axially averaged Nusselt number.

Performance analysis carried out as suggested by Bergles et al. [40] have shown that decreasing the tape-width for the regularly spaced twisted-tape elements yields poorer performance than the full-width tape elements.

5.3. Data for different phase-angle

Figs. 8 and 9 show the effect of the phase-angle between two successive tape elements on friction factor and Nusselt number, respectively, for $y = 2.5$ and $s = 2.5$. For friction factor data, $w = 0.923$ and $d = 0.231$. For Nusselt number data $w = 0.909$ and $d = 0.273$. The friction factor increases by 5% from $\phi = 0°$ to $\phi = 90°$ and by 12% from $\phi = 0°$ to $\phi = 180°$. The corrresponding Nusselt number increases are 8% and 15%, respectively. This observation can be attributed to more vigorous mixing phenomena of the fluid in case of higher-than-zero phase-angle. Here mixing of the fluid particles is possible, even though the fluid is viscous, since the tight twists of the tape (in the present case $y = 2.5$) causes sufficient swirl. Performance analysis carried out as suggested by Bergles et al. [40] have shown that higher-than-zero phase angle is of no use. It only increases the complexity of manufacturing of the tape–rod assembly.

5.4. Data for isothermal and heating friction factor

Fig. 10 shows the variation of isothermal and heated friction factor with Reynolds number for $y = 2.5$, $s = 5$ and $y = 2.5$, $s = 2.5$, respectively. In both cases $w = 0.923$ and $d = 0.231$. The heated friction factor is 7% and 6% less, respectively, than those for isothermal cases. The difference for the swirl flow data was substantially less than the corresponding difference for the plain tube. This is reasonable if one considers the difference in the cross-sections of the flow channels. For the swirl flow, the effective channel size is decreased, but the total peripheral shear stress is substantially increased. If it is assumed that the viscosity correction represents the effective decrease in this total shear stress due to the decrease in fluid viscosity at the heated surfaces, this correction would be equal for the swirl and plain tube cases only if the thermal boundary layer thickness was the same at the tape surface and at the tube wall. Since the heat transfer from the tape is only a small percentage of the total, it is probable that with heat addition the decrease in shear stress at the tape surface is less than that at the tube wall. The magnitude of the isothermal correction factor required for swirl flow would then be also less. To account for this difference, the viscosity ratio exponent of 0.35 that is used to correct the plain tube heated friction factor to isothermal conditions is multiplied by (D_h/D) in order to correct the heated swirl data. This method accurately corrected all heated swirl flow friction factor data of this investigation to isothermal conditions. The same method has also been adopted by Lopina and Bergles [41].

6. Conclusions

Experimental data on the intermediate range of Prandtl number $(205 < Pr < 518)$ twisted-tape-generated laminar swirl flow friction factor and Nusselt number have been presented for the case of a circular tube subjected to uniform wall heat flux. The swirl was generated by regularly spaced twisted-tape elements of different tape-widths, different rod diameters, different phase-angle between successive tape elements. Isothermal as well as heated friction factor data were taken. It has been observed that 'pinching' of tapes in place rather than connecting the tape elements with rods is a better proposition from the thermohydraulic performance point of view. Reducing the tape-widths is worse than the full-width tapes in the tape–rod assembly. Higher than zero phase-angle between the successive tape elements only increases the tape–rod manufacturing complexity rather than yielding any better thermal-hydraulic performance. Difference between heated friction factor and isothermal friction factor for regularly spaced twisted-tape elements is substantially less than that in case of plain tube just as in the case of full-length twisted-tapes.

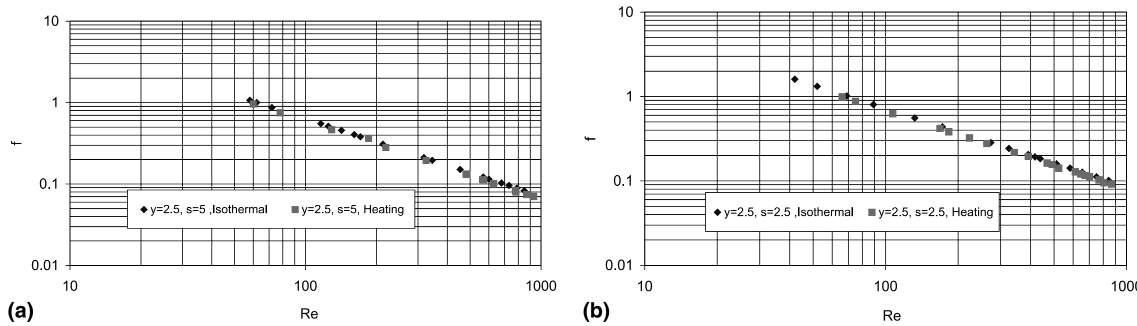


Fig. 10. Variation of friction factor with Reynolds number for regularly spaced twisted-tape elements (comparison of isothermal data with heating data). (a) $y = 2.5$, $s = 5$; (b) $y = 2.5$, $s = 2.5$.

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