Chemical Engineering Journal 78 (2000) 95–105

Chemical Engineering Journal

www.elsevier.com/locate/cej

Augmentation of convective heat transfer inside tubes with three-dimensional internal extended surfaces and twisted-tape inserts

Q. Liao*, M.D. Xin

Institute of Engineering Thermophysics, Chongqing University, Chongqing, China

Received 9 December 1998; received in revised form 15 July 1999; accepted 5 October 1999

Abstract

Experiments were carried out to study the heat transfer and friction characteristics for water, ethylene glycol, and ISO VG46 turbine oil flowing inside four tubes with three-dimensional internal extended surfaces and copper continuous or segmented twisted-tape inserts. During the experiments, Prandtl numbers ranged from 5.5 to 590 and Reynolds numbers from 80 to 50,000. The experimental results show that this compound enhanced heat transfer technique, a tube with three-dimensional internal extended surfaces and twisted-tape inserts, is of particular advantage to enhance the convective heat transfer for the laminar tubeside flow of highly viscous fluid. For the laminar flow of VG46 turbine oil, the average Stanton number could be enhanced up to 5.8-fold inside tubes with three-dimensional internal extended surfaces and continuous twisted-tape inserts compared with an empty smooth tube, and the friction factor was also increased by almost 6.5-fold. Inside the tubes with three-dimensional internal extended surfaces, replacement of the continuous twisted-tape inserts with the segmented twisted-tape inserts induced a greater decrease in the friction factor but a comparatively smaller decrease in the Stanton number.

Keywords: Convective heat transfer; Frictional characteristics; Laminar flow

1. Introduction

Heat exchangers with the convective heat transfer of fluid inside the tubes are frequently encountered in chemical engineering applications. In order to develop the highly efficient and compact heat exchangers, many experimental investigations have been conducted on enhanced tubes for forced single-phase convective heat transfer inside tubes, such as the tubes with two-dimensional internal roughness [1–5], tubes with three-dimensional internal roughness [6–8], and tubes with twisted-tape inserts [9–12]. References [13] and [14] review some details of enhanced tubes. Saha et al. [15] conducted experimental investigations on heat transfer and friction characteristics in smooth tubes with segmented twisted-tape inserts for laminar flow of water at constant heat flux. They found that properly spaced tape segments provide higher enhancement than does a continuous tape. Webb [16] previewed the development of the enhanced heat transfer with continuous and segmented twisted-tape inserts inside smooth tubes.

The tube with three-dimensional internal extended surfaces (3-DIEST) presented in this article is a new high-efficiency heat-exchange tube for forced internal convection. The schematic of the three-dimensional internal extended surface is illustrated in Fig. 1. The 3-DIEST tubes were fabricated by a patented technique of Liao [17]. This fabrication technique has many advantages. For instant, a thinner tube can be used because less material is removed from the inner surface of the tube. The internal fins can be fabricated to be nearly 8 mm high and many tube materials such as copper, brass, aluminium, steel, and even stainless steel can be used with this technique. In addition, the three-dimensional internal fins can be fabricated in an in-line alignment or a staggered alignment. Liao and Xin [18] conducted experimental investigations on the laminar, transitional, and turbulent heat transfer and flow characteristics in seven 3-DIEST tubes. During the experiments, ethylene glycol was used as working fluid while the Prandtl numbers and Reynolds numbers ranged from 60 to 90 and 250 to 7000, respectively. The experimental results indicated that the average Stanton number in the most superior enhancement can be increased by ca. 2.8-fold in laminar flow and 4.5-fold in transitional and turbulent flow compared to smooth tubes, and correspondingly, the friction factor is increased by almost 1.7-fold in laminar flow and 4-fold in transitional and turbulent flow. The fluid disturbance near

* Corresponding author. Tel.: +86-23-65102474; fax: +86-23-65106656.

1385-8947/00/$ – see front matter
PII: S1385-8947(00)00134-0

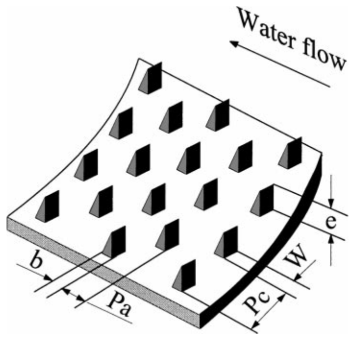


Fig. 1. Schematic of the three-dimensional internal extended surface.

tube wall induced by the three-dimensional roughness has a significant influence on the enhanced heat transfer performance. In single-phase tubeside turbulent or transitional flow, there exist three zones between the tube wall and the central axis: a viscous sublayer, a buffer zone, and a turbulent core zone. The majority of the total thermal resistance is concentrated in the viscous sublayer and the buffer zone for liquid tubeside flow. The drastic disturbance of fluid near tube wall induced by the 3-DIEST fins can lead to a greater decrease of the thermal resistance in the former two zones so that a larger heat transfer enhancement can be obtained. In contrast, for a tubeside laminar flow, the thermal resistance is distributed almost uniformly over the whole tube. Furthermore, the thickness of the laminar boundary is far larger than the height of the roughness, which permits the fluid to pass smoothly around the roughness and induce very weak disturbance near the tube wall. Consequently, the heat transfer enhancement is obviously lower in a tubeside laminar flow than in a tubeside turbulent or transitional flow. Unfortunately, in chemical engineering fields, a great deal of heat exchange is concerned with convective heat transfer of high viscous fluid in the tubeside laminar flow. Due to low heat transfer coefficient of the tubeside laminar flow, the heat exchangers usually have larger sizes. Consequently, much attention was paid to research for new enhanced heat transfer techniques of tubeside laminar flow in previous work. The objective of this paper is to develop a new technique for further enhancing heat transfer of single-phase forced flow, especially laminar flow, inside the 3-DIEST tube. For this purpose, we propose a new compound enhanced heat transfer technique: combining the twisted-tape inserts into the 3-DIEST tube.

It is well known that twisted-tape inserts can induce a secondary flow which promotes fluid exchange between the wall region and the center region so as to enhance heat transfer in the tube center region. Besides, it can also induce a rotary flow which changes the fluid flow direction and increases the fluid flow velocity near the tube wall. Actually, fluid flow near the tube wall is a composite flow of fluid axial flow and rotary flow inside the tube with twisted-tape inserts. Since the twisted-tape inserts can increase not only the heat transfer in the tube center zone but the fluid flow velocity near tube wall also and the 3-DIEST fins can increase the heat transfer in the zone near tube wall, it was envisaged that this compound enhanced technique could bring out further heat transfer enhancement for the 3-DIEST tube. Besides, since there exists a limitation for the pressure drop of fluid flow in the heat exchanger, the 3-DIEST tube with segmented twisted-tape inserts was also investigated experimentally as well as the tube with the continuous twisted-tape inserts in this paper. The experiments were carried out for turbulent flow, transitional flow, and laminar flow with the working fluids: water, ethylene glycol, and VG46 turbine oil.

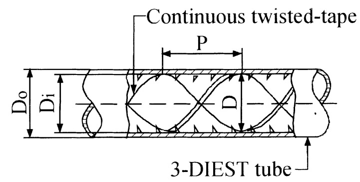
In previous investigations, the twisted-tape inserts were generally applied to enhance convective heat transfer inside a smooth tube. A review of the literature reveals that only a few experimental investigations [19–21] have been carried out for the heat transfer and flow characteristics inside an enhanced tube with twisted-tape inserts. Marner and Bergles [19] testified experimentally that with respect to the empty smooth tube, the heat transfer enhancement of 3-fold in heating and 4-fold in cooling can be obtained in the axial internally finned tube with continuous twisted-tape inserts for tubeside laminar flow of SAE 10 and 20 oil. Usui et al. [20] measured the heat transfer performance of ethylene glycol aqueous solution (the ethylene glycol mass content in the range of 40–60%) with turbulent flow inside the tubes with internally grooved rough surface by means of continuous twisted-tape inserts. The experimental results showed that the heat transfer coefficients inside the compound enhanced tubes were 3.0–3.5 times as much as those inside the empty smooth tube at the same pump power. Zhang et al. [21] conducted experiments on the heat transfer and flow characteristics of air turbulent flow inside three tubes with axial interrupted rib and continuous twisted-tape inserts. The heat transfer enhancement of 1.8–3.2 times can be gained accompanied with the pressure drop penalty of ca. 9–14 times by means of these compound enhanced tubes. No previous work has been found to concern itself with the contents investigated in the present paper. Therefore, the present investigations have great significance for the development of new types of compact heat exchanger in chemical engineering as well as in other industrial fields.

2. Experimental apparatus and procedure

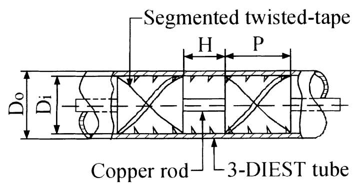
Four 3-DIEST tubes and one smooth tube (I.D. 13.5 mm) were tested. The geometric parameters of these tubes are listed in Table 1. The inside diameter (D_i) listed is measured

Table 1
Geometric parameters of fin inside the 3-DIEST tubes

Tube No.	e/D_i	P_a/e	W/P_a	P_c/W	Fins alignment
Tube 1	0.0771	4.080	0.118	5.421	Staggered
Tube 2	0.0980	3.980	0.114	5.501	Staggered
Tube 3	0.0650	4.050	0.111	5.370	Staggered
Tube 4	0.0763	4.110	0.112	5.418	In-line



(a) The 3-DIEST tube with continuous twisted-tape insert



(b) The 3-DIEST tube with segmented twisted-tape insert

Fig. 2. Schematic of the 3-DIEST tube with continuous or segmented twisted-tape inserts.

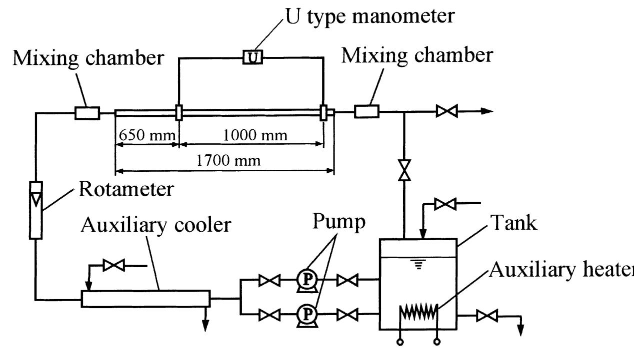


Fig. 3. Schematic of experimental apparatus.

to the bottom of the fins, and the fin-bottom axial thickness (b) is ≈0.8 mm for all the enhanced tubes. A smooth tube was employed to evaluate the experimental accuracy and established a reference for the heat transfer enhancement. Fig. 2 illustrates the schematic of the 3-DIEST tube with continuous or segmented twisted-tape inserts. Three copper continuous twisted-tape inserts with a tape twist ratio of 5, 10, and 15 as well as two copper segmented twisted-tape inserts with a tape twist ratio of 10 and 15 were used in the experiments. The tape twist ratio, P/D, represents the ratio between the length, P, of a tape element with 180° angle of rotation and the tape width, D. The segmented twisted-tape inserts consisted of numerous tape elements with 180° angle of rotation. Every two tape elements were connected with a copper rod of 2 mm diameter by welding. The length, H, of each copper rod equals the length, P, of tape element. All of the twisted-tape inserts used in the experiments are 1000-mm long, 12-mm wide, and 0.5-mm thick. The two ends of the twisted-tape inserts were fixed to the test tube wall in order to prevent the twisted-tape inserts from vibrating.

Water, ethylene glycol, and VG46 turbine oil were chosen as the test fluids. Prandtl numbers of fluid ranged from 5.5 to 590. The Reynolds numbers varied from 80 to 50,000. A schematic of the experimental apparatus is depicted in Fig. 3. The experimental apparatus consisted of a tank with auxiliary heater, two centrifugal pumps, two mixing chambers, a test chamber, an auxiliary shell-and-tube type heat exchanger, and a measurement system. It should be mentioned here that the two centrifugal pumps were only used in the experiments for water and ethylene glycol flow and were replaced with an oil pump in the experiments for VG46 turbine oil flow. The test chamber contained a 650-mm long smooth entrance section and a 1000-mm long heated test section uniformly wounded with 0.8-mm diameter nickel–chrome heating wires. The heat input to the test section was provided by the heating wires connected with an AC transformer and power supply. Thermal insulation for the heated test section was provided by a 50-mm thick glass-fiber cover. In experiments, temperature was measured with nickel–chrome and nickel–silicon thermocouples of 0.2 mm in diameter. Nine thermocouples were embedded uniformly along tube length with 100 mm in spacing outside the test section to measure the wall temperature distribution. In addition, three thermocouples were inserted into the upstream end of test section to measure the inlet temperature, and three thermocouples were inserted into the outlet mixing chamber to measure the outlet temperature. Measurement of the test-fluid flow rate was carried out with a rotameter that was calibrated before the tests. The pressure drop between the inlet and outlet of the test section was measured by a U-type manometer connected to two piezometric rings where eight static-pressure taps (hole diameter 2.0 mm) were drilled on the inner circumference of each ring. The fluid bulk temperature was controlled by adjusting the power to the auxiliary heater in the water tank as well as the flow rate of cooling water in the auxiliary heat exchanger. The isothermal pressure drop test was conducted with 20°C water and with 30°C ethylene glycol and VG46 turbine oil. In each test run, the data for temperatures, flow rates, and fluid pressure drops were recorded after steady-state was established.

3. Experimental results and discussion

3.1. Experimental uncertainties

The isothermal friction factor (f) was defined as

$$f = \frac{\Delta P}{(\rho u_{\mathrm{m}}^2/2)(L/D_{\mathrm{i}})} \tag{1}$$

where ΔP is the pressure drop between the inlet and outlet of test section and u_{m} the bulk average fluid velocity.

Before the experiments, the isothermal friction factors were determined experimentally for the turbulent flow of water inside a smooth tube. Test results are shown in Fig. 4. Discrepancies between the experimental data and Blassius' correlation or f=0.3164(Re)$^{-0.25}$ were less than ±9% over a range of Reynolds number from 7000 to 35,000. The friction factor, f, involves the two parameters, the pressure drop,

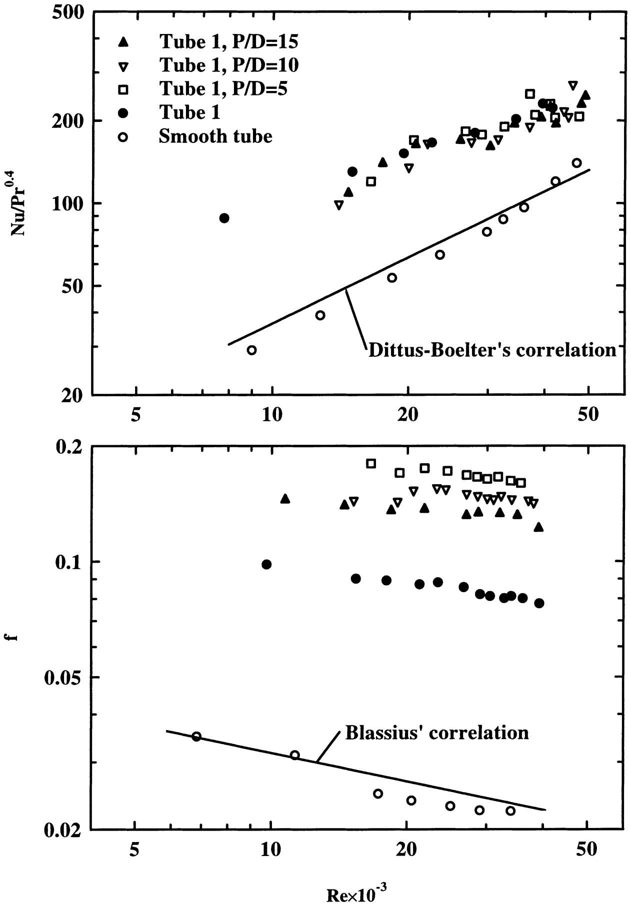


Fig. 4. Heat transfer and friction characteristics inside Tube 1 with continuous twisted-tape inserts for water turbulent flow (Pr≈5.5).

ΔP, and the bulk average fluid velocity, u_m, which need to be measured. The uncertainties of ΔP and u_m were estimated as ±2.5 and ±2.0%, respectively. Using the uncertainty estimation method of Kline and McClintock [22], it was estimated that the uncertainty in the calculated friction factor was ±5.0%.

The heat transfer rate in the test section was calculated using

$$Q = \dot{m}c_p(T_{out} - T_{in}) = U_0A_0(\bar{T}_{wo} - \bar{T}_f) \tag{2}$$

where

$$\frac{1}{(U_0A_0)} = \frac{1}{(h_iA_i)} + \frac{\ln(D_o/D_i)}{(2\pi k_w L)} \tag{3}$$

The internal convective heat transfer coefficient, h_i, was determined by combining Eqs. (2) and (3).

The Nusselt number and Stanton number are defined as

$$\mathrm{Nu} = \frac{h_iD_i}{k_l} \tag{4}$$

$$\mathrm{St} = \frac{\mathrm{Nu}}{(\mathrm{RePr})} = \frac{h_i}{(\rho c_p u_m)} \tag{5}$$

All the fluid thermophysical properties were determined at the average of the inlet and outlet bulk temperatures, $\bar{T}_f$.

The thermal equilibrium test showed that the heating power of the heating wires wrapped over the test section was 6% larger than the absorbed heat of fluid. This was caused by thermal loss from the test section. The smooth-tube heat transfer characteristics are illustrated in Fig. 4 for the turbulent flow of water. The maximum discrepancy of experimental data was less than ±11% as compared with Dittus–Boelter's correlation $\mathrm{Nu}=0.023(\mathrm{Re})^{0.8}(\mathrm{Pr})^{0.4}$.

The uncertainty of heat rate was estimated to ±4.5%. The uncertainties of Nusselt and Stanton numbers were estimated to ±7.0 and ±9.0%, respectively.

3.2. Heat transfer and flow characteristics for water turbulent flow inside the 3-DIEST tubes with continuous twisted-tape inserts

In order to study the heat transfer performance of fluid turbulent flow inside the 3-DIEST tube with continuous twisted-tape inserts, we first conducted experiments on the heat transfer and flow drag for water turbulent flow inside Tube 1 with continuous twisted-tape inserts of 5, 10, and 15 in twist ratio. During the thermal experiments, Prandtl number of water was kept near 5.5. The experimental results are shown in Fig. 4. It can be seen that the heat transfer coefficients for water turbulent flow inside Tube 1 with different continuous twisted-tape inserts were very close to that without twisted-tape inserts, whereas the friction factors were obviously increased by installing the continuous twisted-tape inserts. For liquid tubeside turbulent flow, the thermal resistance in the viscous sub-layer and the buffer zone near tube wall determines the majority of the total thermal resistance. The fluid in the two zones can be drastically disturbed by the three-dimensional roughness inside the empty 3-DIEST tubes. In this case, both the secondary flow and the rotary flow near tube wall induced by the twisted-tape inserts have almost no influence on heat transfer inside the 3-DIEST tube. Besides, the 3-DIEST fins have comparatively small sizes and comparatively uniform outside shapes. Therefore, the variation of fluid flow direction induced by the twisted-tape inserts has almost no influence on the local heat transfer characteristics near the tube wall inside the 3-DIEST tube. It can be concluded that the heat transfer for water turbulent flow inside the 3-DIEST tube cannot be increased significantly by means of the continuous twisted-tape inserts. In contrast, the friction factor was increased greatly because of the increase in the viscous dissipation caused by the secondary flow and the rotary flow as well as the increase in the friction surface due to the existence of the twisted-tape inserts. The friction factor was increased with the decrease of the twist ratio of the twisted-tape inserts, as shown in Fig. 4. The references [20] and [21] indicated that heat transfer enhancement of 3.0–3.5 times and 1.8–3.2 times can be brought about by the continuous twisted-tape inserts for the turbulent flow of the ethylene glycol aqueous solution inside the tubes with internally grooved rough surface and the air turbulent flow inside tubes with axial interrupted rib, respectively. This is in contrast to our experimental results. It can be explained from the different configuration characteristics of the roughness in these enhanced tubes as well as the different thermophysical properties of the working fluid. In the tube with internally grooved rough surface [20], there exists a stagnant zone when fluid flows between the two neighboring grooves. The twisted-tape inserts in this enhanced tube caused the fluid flow direction to change and the flow velocity increased near tube wall so as to eliminate the stagnant zones and increase the turbulence near the tube wall. As a result, the heat transfer for fluid turbulent flow inside the tube can be enhanced significantly by means of the twisted-tape inserts. Similarly, in the tube with axial interrupted rib [21], the interrupted axial fin has a larger axial length relative to its circumferential width, therefore the disturbance of fluid near tube wall induced by the roughness is greatly affected by the flow direction and velocity. In addition, for the air turbulent tubeside flow, the proportion of the thermal resistance in the turbulent core to the total thermal resistance is far larger than that for liquid turbulent flow, implying that the fluid exchange between the wall region and the center region can lead to larger heat transfer enhancement in this enhanced tube. Consequently, the heat transfer of the tube with interrupted axial rib can be significantly enhanced by virtue of the twisted-tape inserts.

3.3. Heat transfer and flow characteristics for the turbulent and transitional flow of ethylene glycol inside the 3-DIEST tubes with continuous twisted-tape inserts

The heat transfer and flow characteristics for ethylene glycol turbulent and transitional flow inside Tube 1 and Tube 2 with continuous twisted-tape inserts of 5, 10, and 15 in twist ratio are shown in Figs. 5 and 6, respectively. The Pr ranged from 65 to 110 in the experiments. From these figures, we can see that the friction factors inside the 3-DIEST tubes with continuous twisted-tape inserts were larger than the corresponding empty 3-DIEST tubes in the whole experimental range and were increased with the decrease of the tape twist ratio at a given Reynolds number. However, the Stanton number difference between the 3-DIEST tubes with continuous twisted-tape inserts and the empty 3-DIEST tubes was smaller in the turbulent flow region, and became larger in transitional and laminar flow regions. As Re decreases, the heat transfer enhancement resulting from the twisted-tape inserts was increased. At lower Re, the fluid disturbance induced by the three-dimensional roughness was comparatively weaker and the thermal resistance in the tube center zone was larger inside the empty 3-DIEST tube. After the continuous twisted-tape inserts were installed, both the increase of fluid flow velocity and the generation of secondary flow can lead to a remarkable heat transfer enhancement. These experimental results prove that the twisted-tape inserts are suitable to enhance heat transfer of fluid laminar flow inside the 3-DIEST tube.

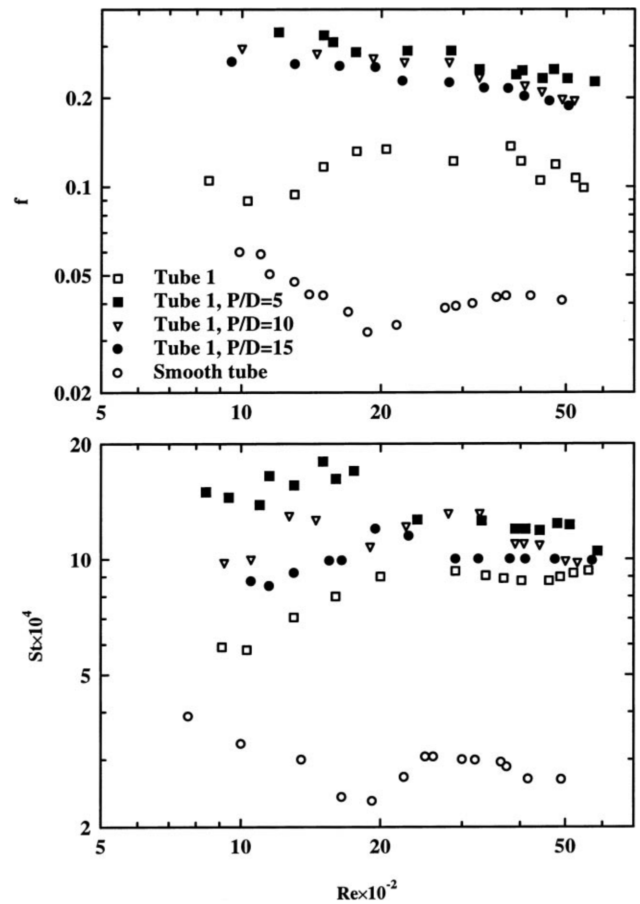


Fig. 5. Heat transfer and friction characteristics inside Tube 1 with continuous twisted-tape inserts for ethylene glycol flow (Pr≈65−110).

The friction factor correlation of Tube 2 with continuous twisted-tape inserts for the turbulent and transitional flow of ethylene glycol is given as

$$f = 11.18\,\mathrm{Re}^{-0.387}\left(\frac{P}{D}\right)^{-0.161} \tag{6}$$

The friction factors predicted from Eq. (6) agree with the experimental data within −5.2–10%. The available range is $710 \leq \mathrm{Re} \leq 4240$.

The heat transfer correlation is as follows

$$\mathrm{St} = 0.00595\,\mathrm{Re}^{-0.130}\left(\frac{P}{D}\right)^{-0.162},$$
$$740 \leq \mathrm{Re} \leq 4300 \quad \text{and} \quad 65 \leq \mathrm{Pr} \leq 110 \tag{7}$$

The experimental data fit the correlation in the range of −10–18%.

3.4. Heat transfer and flow characteristics for the laminar flow of VG46 turbine oil inside the 3-DIEST tubes with continuous twisted-tape inserts

In order to further quantify the heat transfer enhancement for laminar flow inside the 3-DIEST tube by means of the

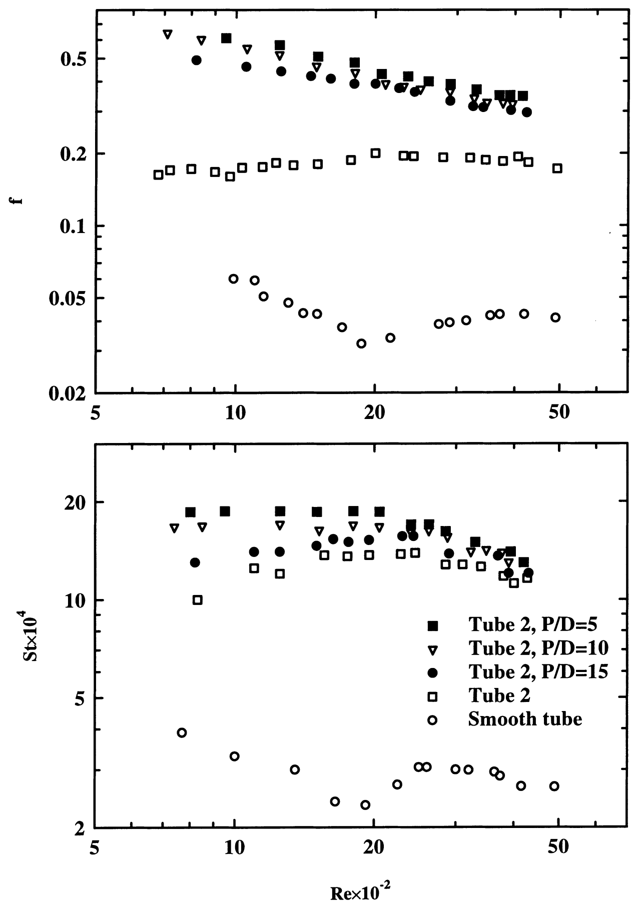


Fig. 6. Heat transfer and friction characteristics inside Tube 2 with continuous twisted-tape inserts for ethylene glycol flow (Pr≈65−110).

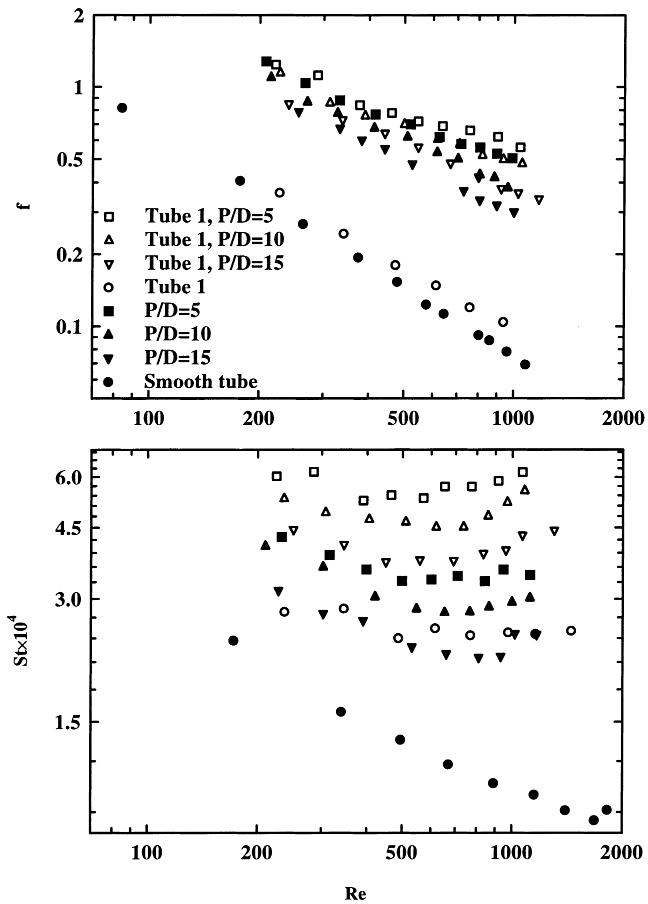


Fig. 7. Heat transfer and friction characteristics inside Tube 1 or smooth tube with continuous twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

continuous twisted-tape inserts, we conducted sequentially experiments on the heat transfer and flow characteristics for VG46 turbine oil laminar flow inside Tubes 1, 2, 3, and 4 with continuous twisted-tape inserts of 5, 10, and 15 in twist ratio. The Pr numbers ranged from 320 to 590. The heat transfer and flow drag inside a smooth tube with the same twisted-tape inserts were experimentally measured to give a reference for the compound enhanced heat transfer technique. The experimental results are depicted in Figs. 7–10. It can be seen that the heat transfer of the 3-DIEST tubes was obviously increased by means of the continuous twisted-tape inserts as well as the friction factor. Both the heat transfer and friction factor were increased with the decrease of the tape twist ratio. As mentioned earlier, in fluid laminar flow inside the empty 3-DIEST tube, the fluid can pass smoothly around the three-dimensional roughness so that very weak disturbance is produced and the thermal resistance is distributed almost uniformly along the tube radial. Consequently, the increase of fluid flow velocity and the secondary flow induced by the twisted-tape inserts can result in significant increase of heat transfer inside the 3-DIEST tubes. From Fig. 10, it can be seen that the Stanton number inside Tube 4 with continuous twisted-tape inserts of 5 in twist ratio was 5.8-, 2-, and 2.3-fold over the empty smooth tube, the empty Tube 4, and the smooth tube with the same twisted-tape inserts, respectively. Correspondingly, the friction factor was increased by 6.5-, 4.5-, and 1.3-fold. Evidently, it can be concluded that the compound enhanced heat transfer technique of the 3-DIEST tube with continuous twisted-tape inserts is very suitable to enhance heat transfer for the laminar flow of high viscous fluid inside tube. However, while the heat transfer was enhanced, the friction factor was greatly increased also. This is required to be improved for the practical application of the compound enhanced technique in industrial fields.

In Figs. 7–10, we can find that the friction factor inside the smooth tube with continuous twisted-tape inserts of 5 in twist ratio was very close to that inside the empty 3-DIEST tube with the same twisted-tape insert. This indicates that the viscous dissipation increase induced by the secondary flow and the rotary flow as well as the larger friction surface resulted from the continuous twisted-tape inserts dominating the total flow drag inside the 3-DIEST tubes with continuous twisted-tape inserts. As compared with the 3-DIEST tubes with continuous twisted-tape inserts, the smooth tube with the same twisted-tape inserts had far smaller Stanton num-

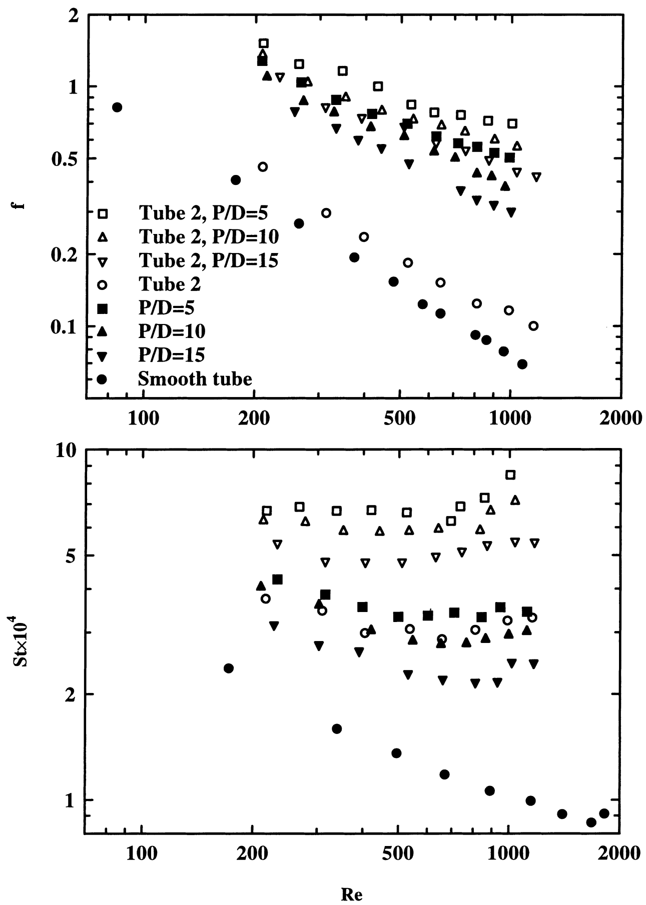


Fig. 8. Heat transfer and friction characteristics inside Tube 2 with continuous twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

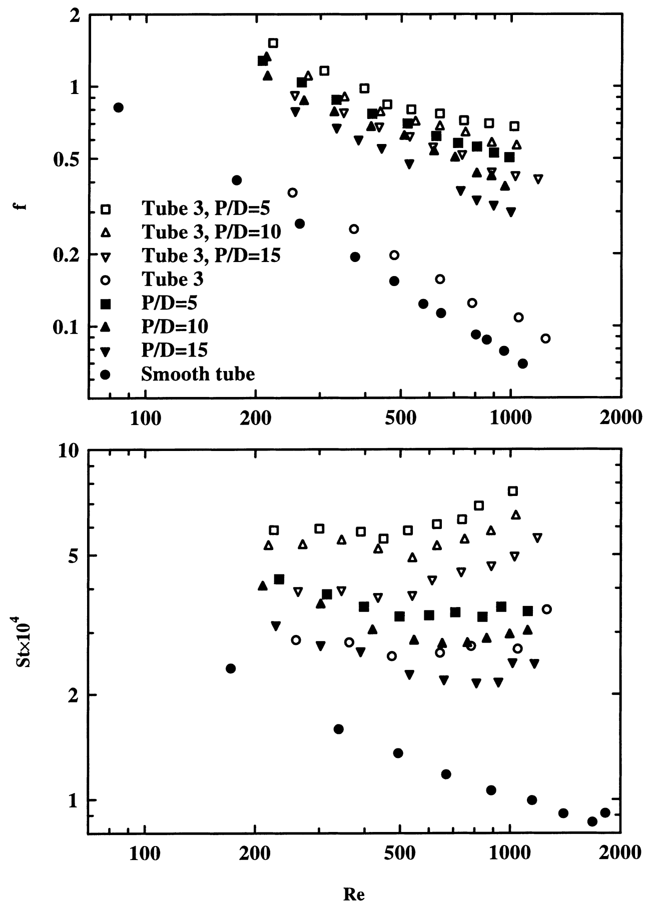


Fig. 9. Heat transfer and friction characteristics inside Tube 3 with continuous twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

bers but similar friction factors. Therefore, the heat transfer enhancement performance of the 3-DIEST tube with continuous twisted-tape inserts is obviously better than that of the smooth tube with the same twisted-tape inserts.

To evaluate the performance improvement, we use the thermal performance ratio, η, defined by William et al. [22] as

$$\eta = \left(\frac{\mathrm{St}/\mathrm{St_s}}{f/f_\mathrm{s}}\right)^{1/3} \tag{8}$$

where St and f are the Stanton number and friction factor for the enhanced heat transfer tubes, $\mathrm{St_s}$ and f_s are the Stanton number and friction factor for smooth tubes at the same operating condition. Fig. 11 shows the variations of the thermal performance ratio with Reynolds number for Tube 4 with continuous twisted-tape inserts of 5 in twist ratio, the empty Tube 4, and the smooth tube with continuous twisted-tape inserts of 5 in twist ratio. We can see that the thermal performance ratio of Tube 4 with continuous twisted-tape inserts of 5 in twist ratio had the highest thermal performance ratio of 2.46−4.70 in these enhanced tubes. The thermal performance ratio of the smooth tube with twisted-tape inserts of 5 in twist ratio was the smallest.

For engineering applications, we give the heat transfer and friction factor correlation of Tube 4 with continuous twisted-tape inserts of 5 in twist ratio which has the highest thermal performance ratio in all the tested tubes. The friction factor correlation is as follows

$$f = 1.519\,\mathrm{Re}^{-0.361}\left(\frac{P}{D}\right)^{-0.085} \tag{9}$$

The discrepancies between the correlation predictions and the experimental data are in the range of −6.5–11%. The correlation is applicable in the range of 189≤Re≤1080.

The heat transfer correlation of Tube 4 with continuous twisted-tape inserts of 5 in twist ratio is given as

$$\mathrm{St} = 0.00186\,\mathrm{Re}^{-0.0646}\left(\frac{P}{D}\right)^{-0.3056} \tag{10}$$

The maximum deviation between the correlation predictions and the experimental data is less than ±12%. The available ranges of the correlation are 190≤Re≤1180 and 320≤Pr≤590.

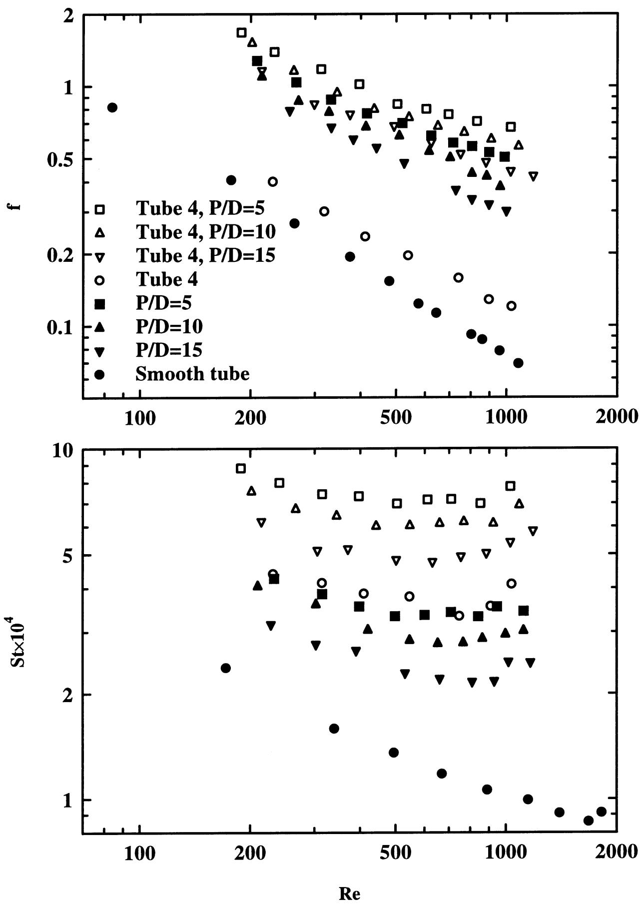


Fig. 10. Heat transfer and friction characteristics inside Tube 4 with continuous twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

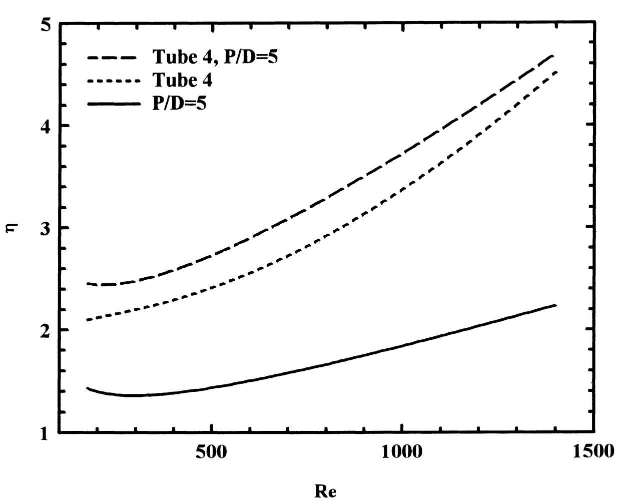


Fig. 11. Thermal performance ratio of the empty Tube 4, Tube 4 with continuous twisted-tape inserts, and the smooth tube with continuous twisted-tape inserts.

3.5. Heat transfer and flow characteristics for the laminar flow of VG46 turbine oil inside the 3-DIEST tubes with segmented twisted-tape inserts

Since the increase of friction surface resulting from the continuous twisted-tape inserts is one of the main reasons why the friction factor inside the 3-DIEST tube is increased significantly by the continuous twisted-tape inserts, the segmented twisted-tape inserts may decrease the friction surface so as to permit a decrease in the friction drag inside the 3-DIEST tube with twisted-tape inserts. For this reason, we carried out experiments on the heat transfer and flow characteristics for the laminar flow of VG46 turbine oil inside a smooth tube, Tube 3, and Tube 4 with the segmented twisted-tape inserts of 10 and 15 in twist ratio. Figs. 12–14 illustrate the experimental results. In Fig. 12, it can be seen that the Stanton number and friction factor inside the smooth tube were decreased by a factor of 12 and 48% with segmented twisted-tape inserts of 10 in twist ratio as much as with continuous twisted-tape inserts of the same twist ratio, respectively. Figs. 13 and 14 indicate that the Stanton number was decreased by a factor of 18 and 15% inside Tube 3

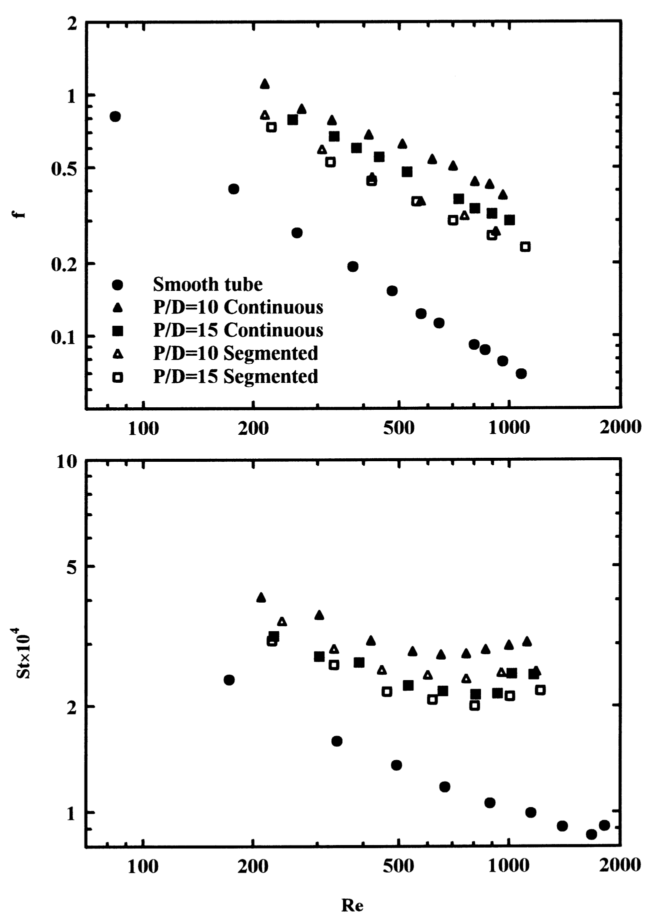


Fig. 12. Heat transfer and friction characteristics inside smooth tube with segmented twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

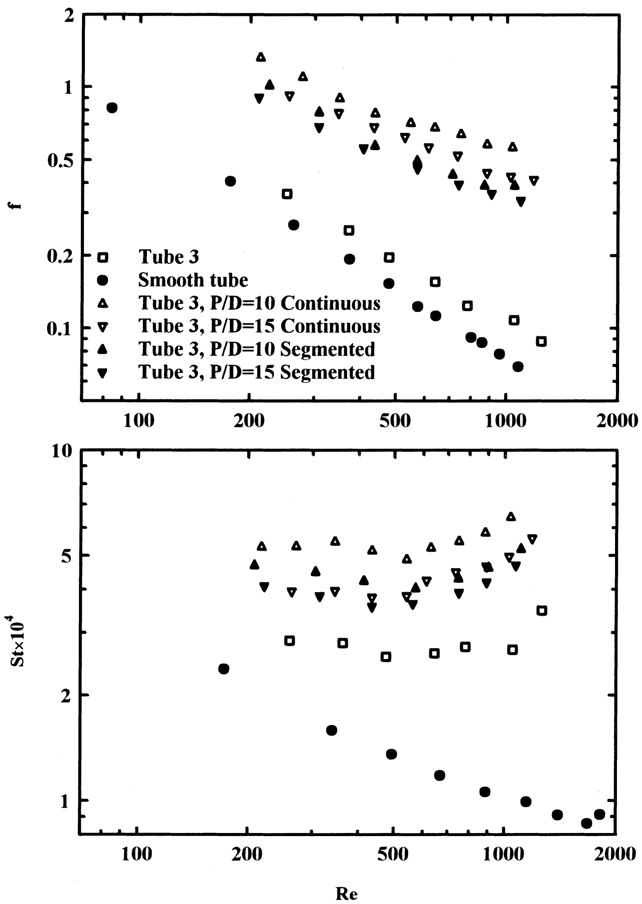


Fig. 13. Heat transfer and friction characteristics inside Tube 3 with segmented twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

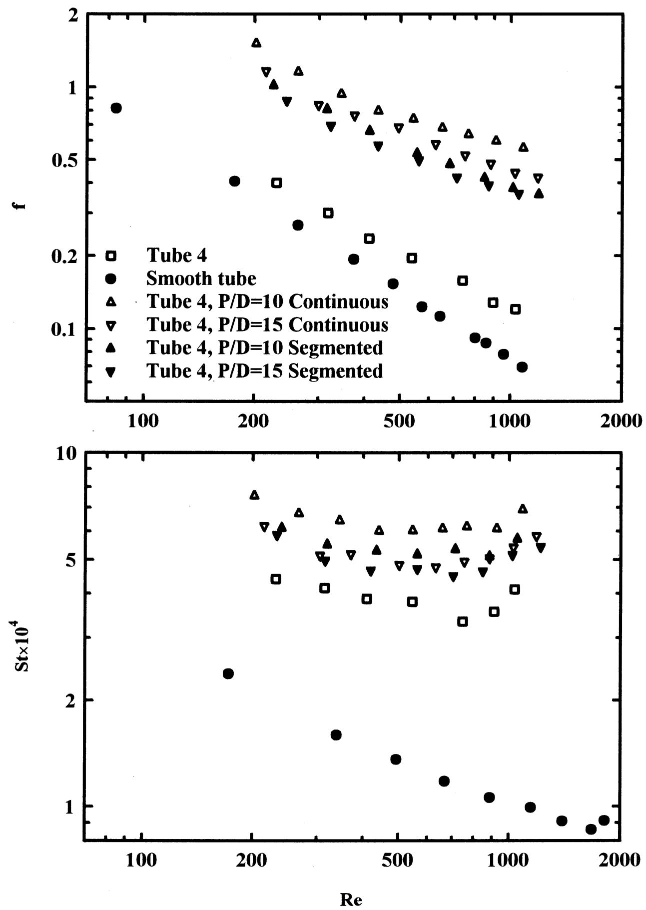


Fig. 14. Heat transfer and friction characteristics inside Tube 4 with segmented twisted-tape inserts for VG46 turbine oil flow (Pr≈320−590).

and Tube 4 with segmented twisted-tape inserts of 10 in twist ratio over these tubes with continuous twisted-tape inserts of the same twist ratio, respectively. Correspondingly, the friction factor was decreased by a factor of 44 and 41%. The decrease of St is obviously lower than that of the friction factor induced by replacement of continuous twisted-tape inserts with the segmented twisted-tape inserts. It should be noted here that the decrease in friction factor inside the smooth tube was very close to that inside the 3-DIEST tubes, which confirms that the reduced friction surface of the segmented twisted-tape inserts is a principal reason for this decrease of friction factor. The three-dimensional roughness has almost no influence on this variation of friction factor. We can conclude that the segmented twisted-tape inserts can decrease efficiently the flow drag but have very little influence on the heat transfer as compared to the 3-DIEST tube with continuous twisted-tape inserts.

4. Conclusions

This article presents the experimental results on heat transfer and friction characteristics for various liquids with turbulent, transitional, and laminar flow inside four 3-DIEST tubes with different continuous or segmented twisted-tape inserts.

In turbulent and transitional flow regions, the heat transfer is increased only a little, whereas the friction factor is increased significantly for liquid flow inside the 3-DIEST tube by means of twisted-tape inserts. The compound enhanced heat transfer technique of the 3-DIEST tube with twisted-tape inserts is suitable for enhancing the heat transfer for highly viscous fluid laminar flow inside a tube.

As compared with the empty smooth tube, the Stanton number of the 3-DIEST tube with continuous twisted-tape inserts can be enhanced up to 5.8-fold and the corresponding friction factor is increased by 6.5-fold with the laminar tubeside flow of VG46 turbine oil. The thermal performance ratio is up to 2.46−4.70. Both the Stanton number and friction factor inside the 3-DIEST tube with twisted-tape inserts increase with the decrease of the tape twist ratio.

For fluid flow inside the 3-DIEST tube with twisted-tape inserts, the significant increase of friction factor is mainly caused by the twisted-tape inserts.

The use of segmented twisted-tape inserts can decrease the friction surface so as to decrease the friction factor inside the 3-DIEST tubes with twisted-tape inserts by comparison to the continuous twisted-tape inserts. The friction factor can be decreased by a factor of 41−44% with the laminar flow of VG46 turbine oil inside the 3-DIEST tubes with segmented twisted-tape inserts of 10 in twist ratio over with continuous twisted-tape inserts in the same twist ratio, while the Stanton number is decreased by a factor of 15−18%. The use of segmented twisted-tape inserts can reduce efficiently the friction factor but induce a comparatively smaller decrease of the Stanton number for the 3-DIEST tube with twisted-tape inserts.

5. Nomenclature

A_i	Surface area at the bottom of fin inside tube
A_o	Surface area outside tube
b	Axial thickness of the fin bottom
c_p	Specific heat of fluid
D	Width of twisted-tape insert
D_i	Inside diameter of tube at the bottom of fin
D_o	Outside diameter of tube
e	Height of fin
f	Friction factor of enhanced tube
f_s	Friction factor of smooth tube
H	Length of each copper rod in the segmented twisted-tape insert
h_i	Average convective heat transfer coefficient
k_l	Thermal conductivity of fluid
k_w	Thermal conductivity of tube wall
L	Tube length of test section
$\dot{m}$	Mass flow rate of fluid
Nu	Nusselt number, $\mathrm{Nu}=h_i D_i/k$
P	Length of a tape element with a rotary angle of 180°
P_a	Axial pitch of fins
P_c	Circumferential pitch of fins
Pr	Prandtl number, $\mathrm{Pr}=\rho c_p \nu/\mathrm{k}$
Q	Heat transfer rate
Re	Reynolds number, $\mathrm{Re}=u_m D_i/\nu$
St	Stanton number of enhanced tube, $\mathrm{St}=h_i/(\rho c_p/u_m)$
St_s	Stanton number of smooth tube
$\bar{T}_f$	Average of fluid bulk temperatures in the test section
T_{in}	Inlet bulk temperature of fluid
T_{out}	Outlet bulk temperature of fluid
$\bar{T}_{wo}$	Average wall surface temperature outside test section
u_m	Bulk average velocity of fluid
U_o	Overall heat transfer coefficient
W	Width of fin

Greek letters

Δp	Pressure drop of fluid
η	Thermal performance ratio, $(\mathrm{St}/\mathrm{St}_s)/(f/f_s)^{1/3}$
ν	Kinematic viscosity of fluid
ρ	Density of fluid

References

[1] R.L. Webb, E.R. Eckert, R.J. Goldstein, Heat transfer and friction in tubes with repeated rib roughness, Int. J. Heat Mass Transfer 14 (1971) 601–617.
[2] T.C. Carnavos, Cooling air in turbulent flow with internally finned tubes, Heat Transfer Eng. 1 (1979) 41–46.
[3] J.G. Withers, Tube-side heat transfer and pressure drop for tubes having helical internal ridging with turbulent/transitional flow of single phase fluid. Part 1. Single-helix ridging, Heat Transfer Eng. 2 (1980a) 48–58.
[4] J.G. Withers, Tube-side heat transfer and pressure drop for tubes having helical internal ridging with turbulent/transitional flow of single phase fluid. Part 2. Multiple-helix ridging, Heat Transfer Eng. 2 (1980) 43–50.
[5] W.J. Marner, A.E. Bertles, Augmentation of highly viscous laminar heat transfer inside tubes in constant wall temperature, Exp. Therm. Fluid Sci. 2 (1989) 252–267.
[6] R.A. Gowen, J.W. Smith, Turbulent heat transfer from smooth and rough surfaces, Int. J. Heat Mass Transfer 11 (1968) 1657–1673.
[7] K. Takahashi, W. Nakayama, H. Kuwahara, Enhancement of forced convective heat transfer in tubes having three-dimensional spiral ribs, Heat Transfer Jpn. Res. 17 (1988) 12–28.
[8] T.J. Rabas, R.L. Webb, P. Thors, N.K. Kim, Influence of roughness shape and spacing on the performance of three-dimensional helically dimpled tubes, J. Enhanced Heat Transfer 1 (1993) 53–64.
[9] J.P. du Plessis, D.G. Kroger, Heat transfer correlation for thermally developing laminar flow in a smooth tube with a twisted-tape insert, Int. J. Heat Mass Transfer 30 (1987) 509–515.
[10] C.J. Genis, W.L. Rautenbach, High heat-flux, forced-convection heat transfer for tubes with twisted-tape inserts, ASME HTD 68 (1987) 1–9.
[11] M. Monheit, Experimental evaluation of the convective characteristics of tubes with twisted tape inserts, ASME HTD 68 (1987) 11–18.
[12] M.M. Rao, V.M.K. Sastri, Experimental investigation for fluid flow and heat transfer in a rotating tube with twisted-tape inserts, Heat Transfer Eng. 16 (1995) 19–28.
[13] A.E. Bergles, Some perspective on enhanced heat transfer — second-generation heat transfer technology, J. Heat Transfer 110 (1988) 1082–1096.
[14] T.S. Ravigururajan, T.J. Rabas, Turbulent flow in integrally enhanced tubes. Part 1. Comprehensive review and database development, Heat Transfer Eng. 17 (1996) 19–29.
[15] S.K. Saha, U.N. Gaitonde, A.W. Date, Heat transfer and pressure drop characteristics of laminar flow in a circular tube fitted with regularly spaced twisted-tape elements, Exp. Therm. Fluid Sci. 2 (1989) 310–322.
[16] R.L. Webb, Principles of Enhanced Heat Transfer, Wiley, New York, 1994.
[17] G.Y. Liao, Three-Dimensional Internally Fined Tube and Machining Technique, Chinese Patent: 88102575.5 (1988).
[18] Q. Liao, M.D. Xin, Experimental investigation on forced convective heat transfer and pressure drop of ethylene glycol in tubes with three-dimensional internally extended surface, Exp. Therm. Fluid Sci. 11 (1995) 343–347.
[19] W.J. Marner, A.E. Bergles, Augmentation of tubeside laminar flow heat transfer by means of twisted-tape inserts, static-mixer inserts, and internally finned tubes, Proc. 6th Int. Heat Transfer Conf., Toronto 5 (1978) 358–365.
[20] H. Usui, Y. Sano, Heat transfer enhancement effects by combined use of internally grooved rough surfaces and twisted tape, Heat Transfer Jpn. Res. 3 (1985) 19–32.

[21] Y.M. Zhang, J.C. Han, C.P. Lee, Heat transfer and friction characteristics of turbulent flow in circular tubes with twisted-tape inserts and axial interrupted ribs, J. Enhanced Heat Transfer 4 (1997) 297–308.

[22] F. William, M.A.M. Pirie, C. Warburton, Heat transfer from surfaces roughened by Ribs, Symp. Vol. Augmentation of Convective Heat and Mass Transfer, in: A.E. Bergles, R.L. Webb, (Eds.), ASME, New York, 1970, pp. 36–43.