Pergamon

Int. J. Heat Mass Transfer. Vol. 39, No. 17, pp. 3547–3557, 1996

0017–9310/96 \$15.00+0.00

PII: S0017–9310(96)00039–7

Heat transfer augmentation for the flow of a viscous liquid in circular tubes using twisted tape inserts

S. K. AGARWAL† and M. RAJA RAO

Department of Chemical Engineering, Indian Institute of Technology, Bombay 400 076, India

(*Received in final form* 6 *February* 1996)

Abstract—Isothermal and nonisothermal friction factors and mean Nusselt numbers for uniform wall temperature (UWT) heating and cooling of Servotherm oil (Pr = 195–375) were experimentally determined for their flow in a circular tube (Re_a = 70–4000) with twisted tape inserts (y = 2.41–4.84). Isothermal friction factors were found to be 3.13–9.71 times the plain tube values. The Nusselt numbers were found to be 2.28–5.35 and 1.21–3.70 times the plain tube forced convection values based on constant flow rate and constant pumping power, respectively, for the minimum twist ratio tape. Correlations were developed for the prediction of isothermal friction factors and the Nusselt numbers. A correlation representing the effect of heat transfer on friction factors is also proposed. Copyright © 1996 Elsevier Science Ltd.

INTRODUCTION

The process of improving the performance of a heat transfer system is referred to as *heat transfer augmentation, enhancement* or *intensification*. Swirl flow devices form an important group of passive augmentation techniques. *Twisted tape* is one of the most important members of this group. In many of the tubular heat exchangers used in the chemical process industry, the low heat transfer coefficient of the tube-side viscous liquid in laminar flow and the high coefficient in the annulus or shell-side, for turbulent flow or for condensing steam, correspond to the practical uniform wall temperature (UWT) boundary condition. The overall heat transfer coefficient, in all such cases, is controlled by the low heat transfer coefficient for laminar flow on the tube-side. Thus any effort to enhance the inside heat transfer coefficient results in improving the overall performance of a heat exchanger. Many of the passive and active techniques that are available for augmentation of laminar flow heat transfer have been discussed in detail by Bergles [1], Bergles and Joshi [2] and Webb [3]. Twisted tapes can be most beneficially used to augment laminar flow heat transfer as the improvement in heat transfer has been found to be much more than the corresponding increase in pumping power. Further, they are one of the simplest of the various augmentation devices and can be fabricated in any moderately equipped workshop.

Analytical correlations for isothermal friction factors for laminar flow in tubes with twisted tape inserts, were proposed by Date and Singham [4] and Date [5] and were modified by Shah and London [6] to account for the tape thickness. Du Plessis and Kroger [7] developed an analytical procedure for the calculation of isothermal friction factors in terms of 'effective flow parameters'. Although both the approaches were found [8] to predict almost the same values for higher twist ratios and/or at lower Reynolds numbers, the correlations of Du Plessis and Kroger [7] predict higher friction factors as the twist ratio is decreased (for $y < 5$) and/or the Reynolds number is increased, i.e. for higher values of (Re_a/y).

Hong and Bergles [9] studied the heating of water and ethylene glycol (Pr = 3–192, Re_a = 13–2460, y = 2.45 and 5.08) under uniform heat flux (UHF) boundary condition. They proposed the following correlation for the prediction of fully developed Nusselt numbers:

$$Nu_a = 5.172[1+5.484\times10^{-3}\{Pr(Re_s/y)^{1.78}\}^{0.7}]^{0.5}. \tag{1}$$

Equation (1) was also found to be valid for a wider range of system parameters, as reported by Sukhatme *et al.* [10] and Saha *et al.* [11].

Marner and Bergles [12] were the first investigators to recognize the importance of uniform wall temperature (UWT) boundary conditions to a major group of heat exchangers used in the chemical process industry. They studied UWT heating and cooling of ethylene glycol (Pr = 24–85, Re_a = 380–3470) using a single twisted tape of y = 5.4. Marner and Bergles [13] extended this work to study UWT heating and cooling of a highly viscous liquid polymer Polybutene-20 (Pr = 1260–8130, Re_a = 15–575, Gz = 868–6570) using the same tape of y = 5.4. Based on these limited

† Present address: Department of Chemical Engineering, Regional Engineering College, Rourkela 769 008, India.

NOMENCLATURE

- A_i heat transfer area, $(\pi D L_h)$ [m^2]
- A_s net flow area of the tube with twisted tape, $(\pi D^2/4 - t \cdot D)$ [m^2]
- A_t cross-sectional area of empty tube $(\pi D^2/4)$ [m^2]
- C_p specific heat of the test liquid [$kJ\ kg^{-1}\ K^{-1}$]
- D inside diameter of the tube [m]
- D_1 inner diameter of the annulus [m]
- D_2 outer diameter of the annulus [m]
- D_e equivalent diameter of the annulus [m]
- D_m log-mean diameter of the tube [m]
- D_0 outside diameter of the tube [m]
- f fanning friction factor, $(\Delta P \cdot D/2L_p \cdot \rho V^2)$, dimensionless
- g acceleration due to gravity [$m\ s^{-2}$]
- Gr Grashof number, $(D^3 \rho^2 g \beta \Delta T/\mu^2)$, dimensionless
- Gz Graetz number, $(m_t C_p/k L_h)$, dimensionless
- h film heat transfer coefficient [$W\ m^{-2} \cdot K^{-1}$]
- h_i tube side film heat transfer coefficient [$W\ m^{-2} \cdot K^{-1}$]
- h_0 annulus side film heat transfer coefficient [$W\ m^{-2} \cdot K^{-1}$]
- Δh differential height of vertical manometer, $(\Delta R \sin \theta)$ [m]
- H axial distance for 180° rotation of the tape [m]
- k thermal conductivity of the test liquid [$W\ m^{-1}\ K^{-1}$]
- k_w thermal conductivity of the tube wall [$W\ m^{-1}\ K^{-1}$]
- L_h length of the heat transfer section [m]
- L_p length of test section between pressure taps [m]
- LMTD overall log-mean temperature difference [K]
- m exponent of viscosity ratio in equation (22)
- m_t mass flow rate of the test liquid [$kg\ s^{-1}$]
- N number of heat transfer tubes
- Nu Nusselt number, (hD/k), dimensionless
- P pumping power [W]
- ΔP pressure drop across the test section, $\Delta h\ (\rho_m - \rho)\ g$ [$N\ m^{-2}$]
- Pr Prandtl number, $(C_p \mu/k)$, dimensionless
- Q heat transfer rate [W]
- Re Reynolds number, $(DV\rho/\mu)$, dimensionless
- Re_0 equivalent plain tube Reynolds number [21], dimensionless
- R_1 thermal performance ratio (Nu_a/Nu_0), on constant mass flow rate basis [equation (37)], dimensionless
- R_3 thermal performance ratio (Nu_a/Nu_0), on constant pumping power basis [equation (38)], dimensionless
- ΔR differential height of inclined manometer [m]
- t thickness of the twisted tape [m]
- T_i temperature at inlet of the test section [°C]
- T_w average wall temperature of the tube [°C]
- T_{wi} inside surface temperature of the tube wall [°C]
- T_1–T_7 fluid temperatures as shown in Fig. 1 [°C]
- T_{12}–T_{19} tube wall temperatures for the heater [°C]
- $(\Delta T_i)_m$ log-mean temperature difference as defined in equations (11) and (19) [K]
- ΔT temperature driving force [K]
- U_i overall heat transfer coefficient based on A_i [$W\ m^{-2} \cdot K^{-1}$]
- V average velocity of the fluid [$m\ s^{-1}$]
- x_w tube-wall thickness [m]
- y twist ratio of the twisted tape, (H/D), dimensionless.

Greek symbols

- β coefficient of volumetric expansion, [K^{-1}]
- μ dynamic viscosity of the test liquid [$kg\ m^{-1}\ s^{-1}$]
- ρ density of the test liquid [$kg\ m^{-3}$]
- ρ_m density of the manometric liquid [$kg\ m^{-3}$]
- θ angle of inclination of the manometer with horizontal [deg].

Subscripts

- a for the augmented case, based on D and A_t
- b based on bulk temperature
- iso isothermal
- non-iso non-isothermal
- o for the plain tube case, based on D and A_t
- s for the augmented case based on D and A_s
- w based on wall temperature.

experimental results [12, 13] with only one tape and their own analytical results [14] for UWT, constant property, developing flow in a semi-circular duct ($y = \infty, t = 0$), Manglik and Bergles [15] proposed the following correlation to predict the Nusselt numbers:

$$Nu_a = 4.631(\mu_b/\mu_w)^{0.14}[0.4935\{Pr(Re_s/y)^{3.475}\}^{0.53} + \{1+0.0954(Gz_a)^{0.8685}\}^{2.6316}]^{0.2}. \quad (2)$$

Equation (2) was found to predict the experimental results of Marner and Bergles [12, 13] within $\pm 25\%$. Monheit [16] studied UWT cooling of a synthetic lubricating oil ($Pr = 150$–450, $Re_a = 500$–3500 and $Gz = 30$–1300) using twisted tape inserts ($y = 2.0$–3.7). Reddy [17] investigated the UWT heating of Servotherm oil ($Pr = 440$–480, $Re_a = 30$–385) using tapes of $y = 3.02$–8.60. Twisted tapes have also been studied in combination with an internally finned tube [18] and a spirally corrugated tube [19] as a compound augmentation device, for UWT heat transfer studies.

Recently Manglik and Bergles [20] analysed the existing correlations for the prediction of friction factors and the Nusselt numbers and concluded that:

(1) they show a considerable disagreement among themselves;
(2) most of the correlations do not have general applicability and are restricted to the flow and heat transfer conditions of the experiments on which they were based; and
(3) in the case of numerical solutions, the underlying assumptions tend to model an over-simplified situation

Further, their [20] experimental results for water and ethylene glycol ($Pr = 3.5$–100 and $y = 3.0$–6.0) did not agree completely with any of the existing correlations. In view of these findings, they developed correlations for the prediction of friction factors and Nusselt numbers in terms of a dimensionless swirl parameter, S_w, obtained by the balance of forces.

In the present study [8], experiments were conducted to determine isothermal and nonisothermal friction factors and mean Nusselt numbers for uniform wall temperature (UWT) heating and cooling of Servotherm oil (medium grade), flowing on the tube-side of a double-pipe heat exchanger with twisted tape inserts. The performance of tubes with twisted tape inserts was compared to that of the plain tube based on constant flow rate (ratio R_1) and fixed pumping power (ratio R_3) as proposed by Bergles *et al.* [21]. Correlations were also developed for the prediction of friction factors and the Nusselt numbers.

The range of the system parameters studied is as follows:

Prandtl number $Pr = 195$–375;
Reynolds number $Re_s = 70$–4000;
Graetz number range $Gz = 180$–7100;
viscosity ratio $(\mu_b/\mu_w) = 3.19$–6.87 (for heating) and $= 0.15$–0.36 (for cooling).

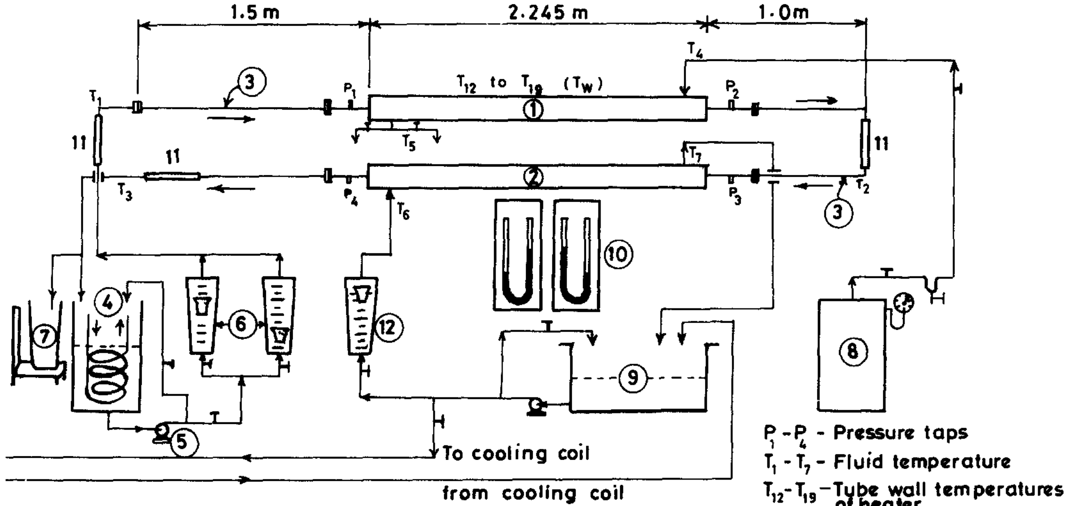
EXPERIMENTAL WORK

Experimental set-up

Schematic diagram of the experimental set-up is shown in Fig. 1. Experiments were conducted in two double-pipe heat exchangers—one for heating and the other for cooling, used in series. The test liquid was flowing on the tube-side (copper tube, $D = 25.0$ mm, $L_h = 2245$ mm and $L_p = 2590$ mm) and the heat transfer medium was passed, in counter-flow, through the annulus. Dry and saturated steam at a pressure of about 0.2 MN m^{-2} (*g*) was used as the heating medium in the heater. Chilled water at a constant flow rate of 0.692 kg s^{-1} (temperature $\cong$ 16°C) was used as the cooling medium in the cooler. The pressure drops across the test-section were measured by using inclined U-tube manometers ($\theta = 5$–$90°$). The manometers were mounted on a specially designed aluminium frame which can be adjusted easily to give any angle of inclination to the horizontal. Mercury was used as the manometric liquid. A mild steel storage tank (capacity 0.50 m^3) equipped with a cooling coil was used as a storage tank for the test liquid. Teflon coated chromel–alumel (30 SWG) thermocouples were used to measure the temperatures at inlet and outlet of the tube-side fluid (T_1–T_3) and the annulus-side fluids (T_4–T_7) and also the temperatures of tube-wall of the heater (T_{12}–T_{19}). Static mixers were used to mix the test liquid thoroughly before the temperatures T_1–T_3 were measured. Two rotameters (range 1–10 and 10–100 l min^{-1}) were provided to indicate the approximate flow rate of the test liquid, the actual being determined for each run separately by actual measurement. The temperature in the test liquid storage tank was maintained constant by circulating chilled water (temperature $\cong$ 16°C) through the cooling coils.

Eight thermocouples were silver brazed to the tube wall of the heater, at equally spaced locations along the length of its tube-wall. The thermocouples were located in small grooves (about 0.75 mm deep) on the top surface of the tube only. This was done in view of the experimental set-up being designed for the uniform wall temperature boundary conditions, especially for the case of twisted tape inserted tubes. The thermocouple wires were taken out of the annulus through brass glands, brazed to the wall of the outer pipe at each of these axial locations. Both of the heat exchangers were similar, except that the cooler (second exchanger used for cooling) was not provided with the thermocouples on its tube-wall. All the thermocouples were connected to a multipoint (22-point) digital temperature indicator (resolution 0.1°C). The heat exchangers and the connected piping were adequately insulated.

Twisted tapes were made from stainless steel strips of thickness 1.00 mm and width 23.5 mm. They were



1. Double pipe heat exchanger (heater)	5. Centrifugal pump for oil	10. Manometers (2 in no.)
2. Double pipe heat exchanger (cooler)	6. Test liquid rotameters	11. Static mixers
3. Calming sections	7. Sample weighing arrangement	12. Rotameter for chilled water
4. Test liquid storage tank with cooling coil inside	8. Boiler	
	9. Chilled water unit	

Fig. 1. Schematic diagram of experimental set-up.

fabricated by twisting a straight tape, about its longitudinal axis, while being held under tension. This was done with the help of a lathe.

Procedure

The experimental set-up was first standardized by determining the flow friction and heat transfer results in a plain empty tube and comparing them with the available correlations. Steady-state values of isothermal and nonisothermal friction factors and mean Nusselt numbers for uniform wall temperature (UWT) heating and cooling of Servotherm oil were then determined with each of the twisted tape inserts.

Data reduction

The Fanning friction factor and the mean Nusselt number were based on the inside diameter of the empty smooth tube, as suggested by Marner *et al.* [22], to facilitate comparison of the performance of twisted tape inserted tubes with the plain tube values. Unless otherwise stated, the test liquid properties were evaluated using equations (3)–(7), at the arithmetic mean bulk temperature in the test section

$$\rho = (878.48 - 0.6525 \cdot T)\ \mathrm{kg\ m^{-3}} \quad (3)$$

$$C_p = (1870 + 3.692 \cdot T)\ \mathrm{J\ kg^{-1} \cdot K^{-1}} \quad (4)$$

$$(\mu/\rho) = \exp(5.4628 - 5.9653 \times 10^{-2} \cdot T + 2.5938 \times 10^{-4} \cdot T^2 - 4.4743 \times 10^{-7} \cdot T^3) \times 10^{-6}\ \mathrm{m^2\ s^{-1}} \quad (5)$$

$$k = (0.1326 - 7.42 \times 10^{-5} \cdot T)\ \mathrm{W\ m^{-1}\ K^{-1}} \quad (6)$$

$$\beta = -1/\rho\,(\mathrm{d}\rho/\mathrm{d}T) = \frac{0.6525}{(878.48 - 0.6525 \cdot T)}\,[\mathrm{K}]^{-1}. \quad (7)$$

These equations were developed [8] using the data (Table 1) supplied by the manufacturer—M/S Indian Oil Corporation Ltd, Bombay. Friction factors were calculated using

$$f = \frac{\Delta P}{(4L_p/D) \cdot (1/2 \cdot \rho \cdot V^2)}. \quad (8)$$

Nusselt numbers. For heating runs, the heat transfer coefficient h_i was calculated from

$$\text{heat duty, } Q = m_t C_p (T_2 - T_1) = h_i A_i (\Delta T_i)_m, \quad (9)$$

where

$$A_i = \pi D L_h = 0.1763\ \mathrm{m^2} \quad (10)$$

$$(\Delta T_i)_m = \frac{(T_w - T_1) - (T_w - T_2)}{\ln[(T_w - T_1)/(T_w - T_2)]} \quad (11)$$

and

$$T_w = 1/8\,(T_{12} + T_{13} + \cdots + T_{19}). \quad (12)$$

For cooling runs, the overall heat transfer coefficient U_i was determined from

$$\text{heat duty, } Q = m_t C_p (T_2 - T_3) = U_i A_i (\mathrm{LMTD}) \quad (13)$$

where

Table 1. Properties of Servotherm oil (medium grade)

T (°C)	ρ [kg m^{-3}]	C_p [J kg^{-1} K^{-1}]	(μ/ρ) [m^2 s^{-1}]	k [W m^{-1} K^{-1}]
37.8	853.0	2009	35.00	0.1298
93.3	—	2119	—	0.1255
98.9	814.0	—	5.30	—
148.9	783.0	2428	2.35	—
204.4	745.1	2595	1.33	0.1177
260.0	708.0	2847	—	0.1091

$$(\mathrm{LMTD}) = \frac{(T_2 - T_7) - (T_3 - T_6)}{\ln[(T_2 - T_7)/(T_3 - T_6)]}. \quad (14)$$

The tube-side heat transfer coefficient h_i was then found out using

$$1/h_i = (1/U_i) - [(x_w/k_w)(D/D_m) + (D/D_0 h_0)], \quad (15)$$

where the annulus side heat transfer coefficient h_0 was determined by using the equation of Monrad and Pelton [23]

$$Nu = (h_0 D_e/k) = 0.020(Re)^{0.8}(Pr)^{1/3}(D_2/D_1)^{0.53}. \quad (16)$$

More recently, Gnielinski [24] has suggested the following equation for calculating the value of h_0:

$$Nu = (h_0 D_e/k) = \left(\frac{Pr}{Pr_w}\right)^{0.11}\left[\frac{(f/8)(Re-1000)Pr}{1+12.7(f/8)^{1/2}(Pr^{1/3}-1)}\right] \times\left[1+\left(\frac{D_h}{L_h}\right)^{2/3}\right]\left[0.86\left(\frac{D_2}{D_1}\right)^{0.16}\right] \quad (17)$$

where $f = (1.82 \log_{10} Re - 1.64)^{-2}$.

For the present case, it was seen that the values of h_0 obtained by using equations (16) and (17) agree to within $\pm 4\%$. The net effect on the value of h_i was found to be less than 1%.

The inside tube wall temperature for the cooler (for finding out μ_w) was determined by equating

$$U_i(\mathrm{LMTD}) = h_i(\Delta T_i)_m, \quad (18)$$

where

$$(\Delta T_i)_m = \frac{(T_2 - T_{wi}) - (T_3 - T_{wi})}{\ln[(T_2 - T_{wi})/(T_3 - T_{wi})]}. \quad (19)$$

Solving for T_{wi} gives

$$T_{wi} = T_3 - [(T_2 - T_3)/(e^K - 1)] \quad (20)$$

where

$$K = (T_2 - T_3)/(\Delta T_i)_m \quad (21)$$

and $(\Delta T_i)_m = (U_i/h_i)\cdot(\mathrm{LMTD})$ from equation (18).

RESULTS AND DISCUSSIONS

Friction results

The isothermal friction factors for the plain tube and the tube with twisted tape inserts are shown in Fig. 2. Of the plain tube values, 86% were found to within $\pm 5.0\%$ of the analytical equation $f_0 = 16/Re_0$, for both the heater and the cooler, for Reynolds numbers less than 2000. When compared to the predictions of Shah and London [6], the isothermal friction factors for tubes with twisted tape inserts were found to be 21–53% greater for $y = 2.41$, for $Re_s \leqslant 2000$. However, for $y = 4.84$, the experimental values were higher by a maximum of 9% only. The under-prediction by the correlations of Shah and London increases with the increase in Reynolds number and/or reduction in the value of y, i.e. for higher degrees of swirl.

The isothermal friction factors were also compared with the predictions of Du Plessis and Kroger [7] (Fig. 3), and were found to be within +18 to −5% of the predicted values, for $Re_s \leqslant 2000$. The experimental

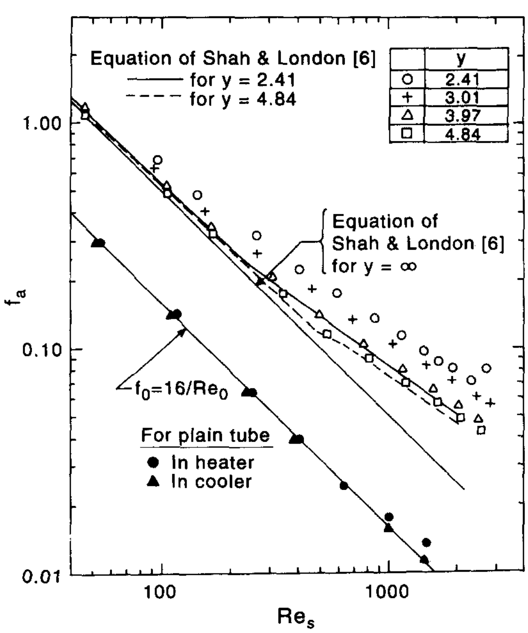


Fig. 2. Isothermal friction factors for flow of Servotherm oil.

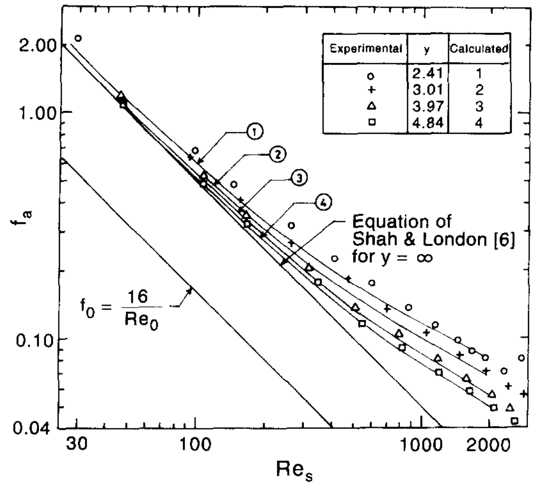


Fig. 3. Comparison of experimental isothermal friction factors with the correlation of Du Plessis and Kroger [7].

values were found to be predicted to within $\pm 10\%$ by the correlation suggested by Manglik and Bergles [20].

The isothermal friction factors were found to be 3.13–9.71 times the plain tube values (for Reynolds number $\leqslant 2100$). They were found to decrease with the increase in Reynolds number, but the rate of decrease reduces at higher Reynolds numbers, thereby indicating a corresponding increase in the swirl flow contribution. This happens at lower Reynolds numbers for tapes of lower twist ratio, thereby indicating an early onset of swirl flow in these cases.

Nonisothermal friction factors (Figs. 4–6), in general, were found to be lower for heating and higher for cooling, when compared to the isothermal values. The difference between the isothermal and nonisothermal friction factors was maximum for the plain tube (Fig. 4) when compared to the tubes with twisted tape inserts (Figs. 5 and 6). For the plain tubes, for heating, they were found to be about 32% lower than the isothermal friction factors, while for cooling they were found to be 17–21% higher. The isothermal and

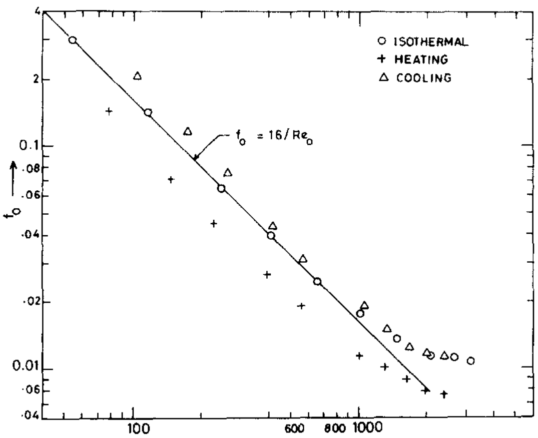


Fig. 4. Nonisothermal friction factors for flow of Servotherm oil in plain tubes.

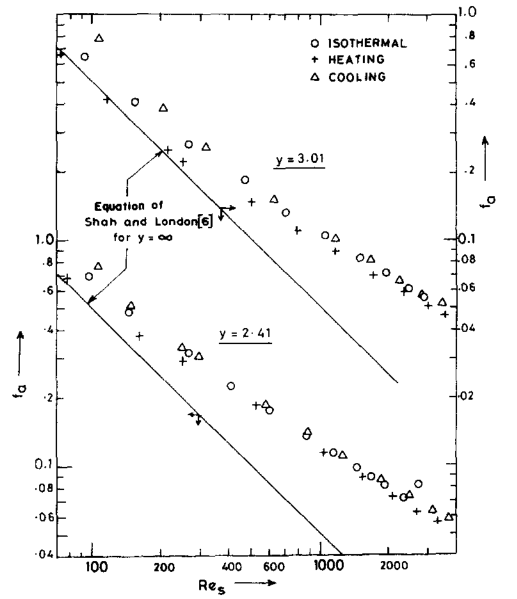


Fig. 5. Nonisothermal friction factors for flow of Servotherm oil in tubes fitted with tapes of $y = 2.41$ and 3.01.

the nonisothermal values, for the plain tubes, could be brought together by the correlation

$$(f_0) = (f_0)_{\text{non-iso}}(\mu_b/\mu_w)^m, \tag{22}$$

where $m = 0.22$ for heating and 0.12 for cooling.

It may be noted that Marner and Bergles [13] also correlated their plain tube friction data for polybutene-20 (for $Re_0 = 15$–575) by a similar approach, using $m = 0.25$ for heating, and 0.12 for cooling.

For the case of tubes with twisted tape inserts, the viscosity ratio effect for both heating and cooling was less pronounced when compared to the case of plain tubes. This was due to the induced swirl flow because

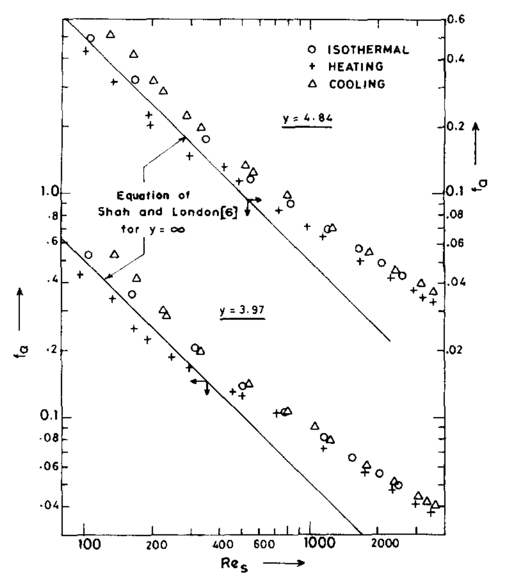


Fig. 6. Nonisothermal friction factors for flow of Servotherm oil in tubes fitted with tapes of $y = 3.97$ and 4.84.

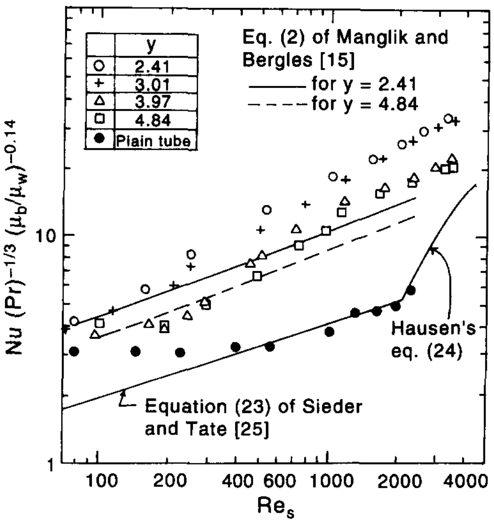


Fig. 7. Heat transfer results for heating of Servotherm oil.

of the presence of twisted tape. The effect of heat transfer was found to be minimum for the tape of $y = 2.41$. At Reynolds numbers greater than about 250 ($Re_s/y \cong 100$), the isothermal and nonisothermal friction factors (Fig. 6) almost merge together due to high degree of swirl flow. On analysis of the friction factors for the tubes with twisted tape inserts, in terms of equation (22), the value of the exponent m was found to be 0.12 for $Re_s \leqslant 250$ and 0.06 for $Re_s \geqslant 250$, for both heating and cooling.

Heat transfer results

Nusselt numbers for the plain tube and the tube with twisted tape inserts, in the form of $Nu/(Pr)^{1/3}(\mu_b/\mu_w)^{0.14}$ vs Reynolds number, are shown in Figs. 7 and 8, for heating and cooling, respectively. The plain tube forced convection equations of Seider and Tate [25], equation (23), and Hausen [26], equation (24), are also included in these figures.

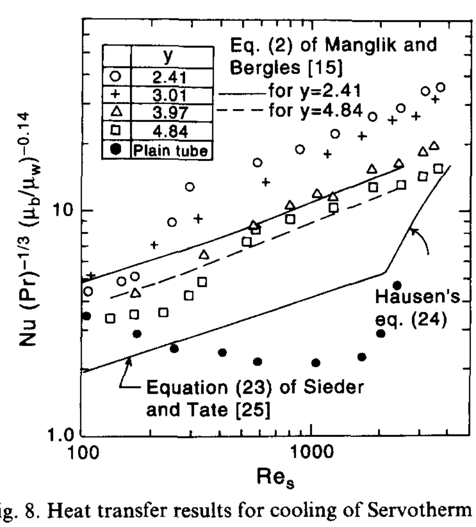


Fig. 8. Heat transfer results for cooling of Servotherm oil.

$$Nu_0 = 2.016(Gz)^{1/3}(\mu_b/\mu_w)^{0.14} \quad (23)$$

(valid for $Gz \geqslant 100$, $Re_0 \leqslant 2100$)

$$Nu_0 = 0.116(Re_0^{2/3}-125)Pr^{1/3} \times \{1+(D/L_h)^{2/3}\}(\mu_b/\mu_w)^{0.14} \quad (24)$$

(valid for $Re_0 = 2100$–$10\,000$).

Correlation of Manglik and Bergles [15], equation (2), for twisted tapes of $y = 2.41$ and 4.84 is also shown in Figs. 7 and 8, for comparison.

For heating of the test liquid in the plain tube (Fig. 7), the Nusselt numbers were found to be higher than those predicted by the forced convection equation (23) for Reynolds numbers $\leqslant 250$, because of the effect of natural convection. However, for higher Reynolds numbers the Nusselt numbers were found to agree within $\pm 8\%$ with the forced convection equations (23) and (24). For cooling of the test liquid also (Fig. 8), some natural convection was observed for Reynolds numbers less than around 200. But the Nusselt numbers were found to be much below the forced convection equations for Reynolds numbers beyond 200. This might be mainly due to the 'cold-wall' insulation effect, which is most prominent in the case of viscous coolers. The slow moving, high viscosity fluid layers near the wall (because of lower temperature) tend to act as an insulator in the heat transfer process, which serves to impede an already poor heat transfer mechanism. This phenomenon was also observed by Oliver and Jenson [27] in their 'two-layer model' for viscous coolers. They ascribed the low heat transfer coefficients to the stratification of layers of widely differing viscosities. It is interesting to note that the effect of stratification increases continuously until a Reynolds number of about 1600, after which it starts declining sharply.

At low flow rates, for both heating and cooling, where the effect of natural convection was observed, the experimental Nusselt numbers were compared with the correlation of Depew and August [28], equation (25), for mixed convection.

$$Nu_0 = 1.75[Gz+0.12(Gz \cdot Gr^{1/3} \cdot Pr^{0.36})^{0.88}]^{1/3}(\mu_b/\mu_w)^{0.14}. \quad (25)$$

The experimental values were found to be within +7 to −32% of the predicted values. It may be pertinent to note here that the correlation of Depew and August was found to predict their own data with short tubes and the data of many other investigations within ±40%.

For the case of tubes with twisted tape inserts, the increase in heat transfer coefficient was found to be maximum for $y = 2.41$ in the entire range of Reynolds numbers studied, for heating as well as for cooling of Servotherm oil. It reduces for higher values of twist ratio. Therefore a lower value of the twist ratio is always preferable to realize maximum enhancement in heat transfer. The Nusselt numbers were found to be 2.28–5.35 times the plain tube forced convection values, for $y = 2.41$.

The correlation of Manglik and Bergles, equation (2), predicts Nusselt numbers in fairly close agreement with the experimental values for low swirl flow conditions, resulting from lower Reynolds numbers and/or higher twist ratios. However for intense swirl flow, obtained by using lower twist ratio and/or higher Reynolds number ($Re_s/y \geqslant 100$), their correlation underpredicts the experimental Nusselt numbers by 15–94%. It may be pertinent to note that their correlation, equation (2), was developed based on experimental data with only one twist ratio ($y = 5.4$), which corresponds to a relatively low degree of swirl flow condition, and the numerical results for $y = \infty$. The experimental Nusselt numbers were also compared with the latest correlation developed by Manglik and Bergles [20] and were found to be predicted within a wide band of $\pm 55\%$.

The effect of twist ratio was found to be somewhat more pronounced in the case of cooling (Fig. 8) when compared to the heating results (Fig. 7), as evidenced by the higher increase in heat transfer coefficients with the reduction in the value of y. The increase in swirl, represented by a reduction in the twist ratio, could thus have a more significant effect on the viscous fluid layers near the wall, in the case of cooling. The effect of 'cold-wall' insulation, so prominent for the plain tube, was almost totally eliminated in the case of tubes fitted with twisted tapes.

CORRELATION OF EXPERIMENTAL RESULTS

As discussed in the previous sections, the available correlations of Shah and London [6] and that of Du Plessis and Kroger [7] for the isothermal friction factors and that of Manglik and Bergles [15, 20] for UWT heat transfer results, do not fully satisfy the present experimental results for Servotherm oil. Further, most of these correlations are valid for Reynolds numbers $\leqslant 2000$ only,whereas the present investigations were carried out for Reynolds numbers up to 4000. Therefore correlations were developed for the prediction of isothermal friction factors and Nusselt numbers for the heating and cooling of Servotherm oil.

Isothermal friction factors

Since the correlations of Shah and London [6] underpredict the present friction data, especially for lower twist ratios and for higher Reynolds numbers, an attempt was made to correlate the isothermal friction factors in terms of Reynolds number and the twist ratio of the twisted tapes. The following correlations were obtained for different ranges of the parameter (Re_s/y):

(1) $(Re_s/y) = 9$–30,

$$(f_a \cdot Re_S \cdot y^{0.28}) = 75.74(Re_S/y)^{0.0216} \quad (26)$$

(2) $(Re_s/y) = 30$–100,

$$(f_a \cdot Re_S \cdot y^{0.28}) = 47.79(Re_S/y)^{0.1553} \quad (27)$$

(3) $(Re_s/y) = 100$–1000,

$$(f_a \cdot Re_S \cdot y^{0.28}) = 19.48(Re_S/y)^{0.3481}. \quad (28)$$

The effect of (Re_s/y) clearly increases as its value is increased from 9 to 1000. Each of these ranges of (Re_s/y) represents a particular degree of swirl flow, arranged in the order of increasing intensity of secondary motion. The correlations (26) and (28) were combined by the method of Churchill and Usagi [29] to give a composite equation for $(Re_s/y) = 9$–1000, which is given below

$$(f_a \cdot Re_S \cdot y^{0.28}) = [\{75.74(Re_S/y)^{0.0216}\}^{10} + \{19.48(Re_S/y)^{0.3481}\}^{10}]^{0.10}. \quad (29)$$

The correlation (29) was found to predict 98% of the experimental data within $\pm 4.5\%$ (for $Re_s/y = 9$–1000, $Re_s = 30$–3000) and the overall deviation was within $\pm 8.0\%$. It is important to note that these empirical correlations were developed based on the range of twist ratio ($y = 2.4$–5.0) which is most important in heat transfer enhancement.

The developed correlation (29) predicts the data of Marner and Bergles [30] for polybutene-20 ($y = 5.4$) within $\pm 10\%$. Reddy's [17] data for Servotherm oil (for $y \leqslant 5.0$), was also predicted well within $\pm 15\%$.

Nusselt numbers

As expected, the Nusselt numbers were found to depend on Reynolds number, Prandtl number, tape twist ratio and the viscosity ratio. As discussed earlier, the effect of twist ratio was found to be somewhat more pronounced in the case of cooling than for heating. The correlations (30) and (31) developed for the present experimental results for heating and cooling of Servotherm oil, for $y = 2.41$–4.84 and $Re_s = 70$–4000, are as follows:

$$Nu_a = 0.725(Re_S)^{0.568}(y)^{-0.788}(Pr)^{1/3}(\mu_b/\mu_w)^{0.14} \quad (30)$$

for heating

$$Nu_a = 1.365(Re_S)^{0.517}(y)^{-1.05}(Pr)^{1/3}(\mu_b/\mu_w)^{0.14} \quad (31)$$

for cooling.

Of the experimental Nusselt numbers 91% for heating and 84% for cooling were found to agree within $\pm 15\%$ of the above equations, over the entire range of Reynolds number studied. However for $Re_a \geqslant 300$, all the heat transfer results for heating and 93% for the case of cooling, were found to be within $\pm 15\%$ of the calculated values.

Therefore the correlations developed for isothermal friction factors (29) and the Nusselt numbers for heating (30) and for cooling (31) of Servotherm oil were found to predict the experimental values satisfactorily.

THERMAL PERFORMANCE OF TWISTED TAPES

Out of the eight performance evaluation criteria suggested by Bergles *et al.* [21], the first and the third ones are used to evaluate the performance ratios R_1 and R_3, for the present investigations. They were calculated as follows:

(1) corresponding to a particular heat transfer run using twisted tapes the Reynolds number, Prandtl number, viscosity ratio (μ_b/μ_w) and nonisothermal friction factors are known.

(2) Calculation of Re_0, the equivalent plain tube Reynolds number.

(a) for performance ratio R_1,

$$Re_0 = Re_a \tag{32}$$

(b) for the performance ratio R_3, Re_0 was calculated so as to satisfy

$$P_0 = P_a \tag{33}$$

(for equal pumping power).

So

$$(f_0)_{\text{non-iso}} \cdot Re_0^3 \cdot (A_t) = (f_a)_{\text{non-iso}} \cdot Re_a^3 \cdot (A_s) \tag{34}$$

(for fixed geometry),

where $(f_0)_{\text{non-iso}}$ and $(f_a)_{\text{non-iso}}$ are the nonisothermal friction factors in the plain tube and the tube with a twisted tape insert, corresponding to Re_0 and Re_a, respectively.

Equation (35) is obtained by rearrangement of equation (34), using the relationship between isothermal and nonisothermal friction factors equation (22) for the plain tube case.

$$(f_0)_{\text{iso}} \cdot Re_0^3 = (f_a)_{\text{non-iso}} \cdot Re_a^3 \cdot (A_s/A_t) \cdot (\mu_b/\mu_w)^m \tag{35}$$

where $m = 0.22$ for heating and 0.12 for cooling.

Re_0 was then calculated from a knowledge of the parameters on the right hand side (RHS) and $f_0 - Re_0$ relationship for the plain tube. For Re_0 less than 2000, $f_0 = 16/Re_0$ was used. For Re_0 ranging from 2000 to 3100, the actual experimental results of isothermal friction factors in the plain tube were used. When Re_0 value was however greater than 3100, Colburn's [31] equation (36) was used.

$$f_0 = 0.046(Re_0)^{-0.20} \tag{36}$$

(valid for $Re_0 = 3000-10^6$).

(3) The equivalent plain tube Nusselt number Nu_0 was then calculated at Re_0 and for the same Pr and (μ_b/μ_w) values as for the augmented case, using forced convection equations (23) and (24), respectively, for laminar and transition flow regions.

(4) The performance ratios R_1 and R_3 were then calculated using equations (37) and (38), respectively, based on the actual experimental data for Nu_a.

$$R_1 = (Nu_a/Nu_0)_{m_t,D,L_h,N,\Delta T,T_i} \tag{37}$$

$$R_3 = (Nu_a/Nu_0)_{P,D,L_h,N,\Delta T,T_i}. \tag{38}$$

The thermal performance ratios R_1 and R_3 are shown in Figs. 9 and 10, respectively for heating and cooling of Servotherm oil. It is important to note that the performance ratios were calculated based on forced convection equations (23) and (24) for the plain tube (i.e. for Nu_0). So in cases, where the natural convection is important, the performance ratios will reduce corresponding to the degree of natural convection in the plain tube. However for the case of cooling, where the heat transfer coefficients for $Re_0 > 200$ were significantly lower (Fig. 8) than those predicted by the forced convection equations, due to the 'cold-wall' insulation effect, the actual performance ratios would increase correspondingly.

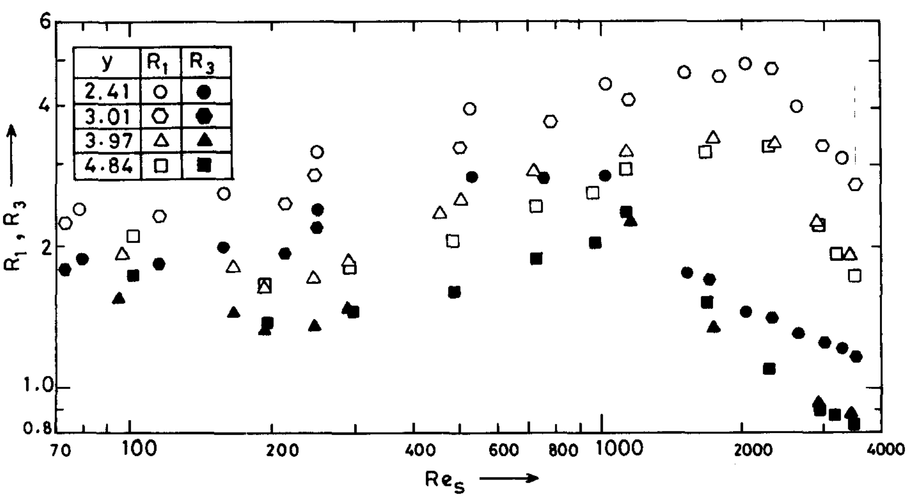


Fig. 9. Thermal performance ratios R_1 and R_3 for heating of Servotherm oil.

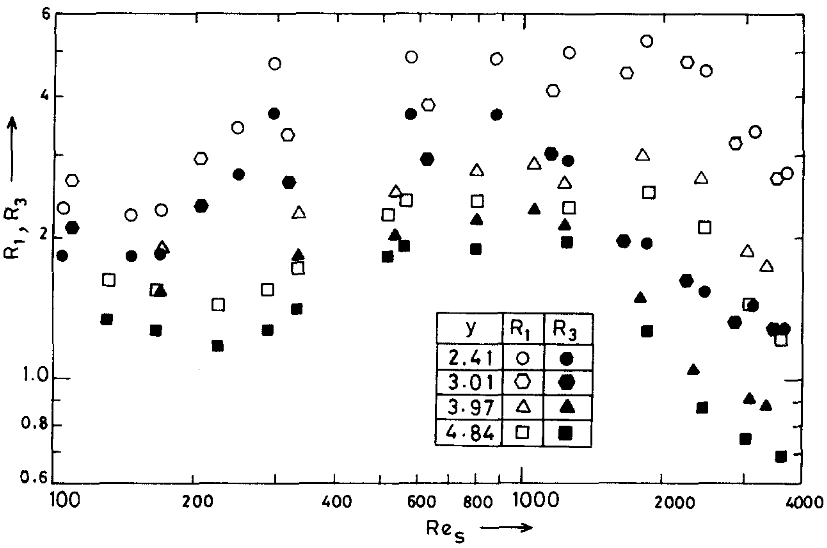


Fig. 10. Thermal performance ratios R_1 and R_3 for cooling of Servotherm oil.

The thermal performance ratio (R_1) was found to vary between 1.65 at low Reynolds numbers of around 200 (for $y = 4.84$) to as high as 5.1 (for $y = 2.41$) at Reynolds number of about 2000, for UWT heating of Servotherm oil (Fig. 9). The corresponding values for cooling (Fig. 10) were 1.45 and 5.35, respectively. The performance ratios were found to be highest for the minimum twist ratio in the entire region of Reynolds numbers studied. They were found to reduce as the twist ratio was increased. After a Reynolds number of about 2000, the R_1 values decline sharply. This might be explained as due to the appearance of turbulence, in the transition region ($Re_0 = 2100$–$10\,000$), in the case of plain tubes, resulting in increased value of Nu_0, equation (24), which in turn decreases the performance ratio R_1 ($= Nu_a/Nu_0$).

On constant pumping power basis, the thermohydraulic performance ratio R_3 was found to vary from 1.35 (for $y = 4.84$) at low Re_s of about 200 to around 2.85 (for $y = 2.41$) for $Re_s \cong 500$–1000, for the case of heating (Fig. 9). The corresponding values of R_3 ranged from 1.18 to 3.70 for the case of cooling (Fig. 10). The maximum R_3 values were realized at Reynolds numbers between 500 and 1000. After a Reynolds number of about 1000, R_3 declines sharply, reaching values even below 1.0 at very high flow rates. This happens because of the fact that around a Reynolds number of 1000, for the twisted tape case, the equivalent plain tube Reynolds number for the constraints of equal pumping power and fixed geometry (i.e. Re_0) becomes greater than 2100. For Re_0 greater than 2100, the Nu_0 increases as per Hausen's equation (24), thus reducing the performance ratio R_3.

CONCLUSIONS

Experiments were conducted to evaluate friction and heat transfer characteristics of twisted tapes for heating and cooling of Servotherm oil (medium grade) under uniform wall temperature. Isothermal friction factors were found to be 3.13 to 9.71 times the plain tube values for the laminar flow. The isothermal friction factors were correlated with the nonisothermal values in the form of equation (22). The value of m was found to be 0.25 for heating and 0.12 for cooling, for the plain tube and 0.12 for $Re_s \leqslant 250$ and 0.06 for $Re_s \geqslant 250$, for both heating and cooling, for the case of twisted tapes. On the basis of both the constant flow rate and constant pumping power, the twisted tape of minimum twist ratio ($y = 2.41$) performed the best over the entire range of Reynolds number, for both heating and cooling. Its performance ratio R_1 was found to vary from 2.28 to 5.35, while the ratio R_3 ranged between 1.21 and 3.70. The proposed correlations can be used to predict isothermal friction factors, equation (29), and the Nusselt numbers for UWT heating, equation (30), and cooling, equation (31), of viscous liquids using twisted tapes of $y \leqslant 5.0$.

Acknowledgements—This work was done by Dr S. K. Agarwal (on deputation cum study leave from Regional Engineering College, Rourkela) as a part of his doctoral programme during July, 1988 to February, 1992, at the Department of Chemical Engineering, Indian Institute of Technology, Bombay, under the guidance of Professor M. Raja Rao (retired since December, 1991). We are thankful to the authorities at I.I.T., Bombay for providing all the facilities for completing this work.

REFERENCES

1. A. E. Bergles, Enhancement of heat transfer, *Proceedings of the 6th International Heat Transfer Conference*, Toronto, Vol. 6, pp. 89–108 (1978).
2. A. E. Bergles and S. D. Joshi, Augmentation techniques for low Reynolds number in-tube flow. In *Low Reynolds Number Flow Heat Exchangers* (Edited by S. Kakac, R. K. Shah and A. E. Bergles), pp. 695–720. Hemisphere, Washington, DC (1983).
3. R. L. Webb, Enhancement of single-phase heat transfer.

In *Handbook of Single-phase Convective Heat Transfer* (Edited by S. Kakac, R. K. Shah and Win Aung), Chap. 17. Wiley-Interscience, New York (1987).
4. A. W. Date and J. R. Singham, Numerical prediction of friction and heat transfer characteristics of fully developed laminar flow in tubes containing twisted tapes, paper no. 72-HT-17, ASME, New York, (1972).
5. A. W. Date, Prediction of fully developed flow in a tube containing a twisted tape, *Int. J. Heat Mass Transfer* **17,** 845–859 (1974).
6. R. K. Shah and A. L. London, Laminar flow forced convection in ducts. In *Advances in Heat Transfer* (Edited by T. F. Irvine Jr and J. P. Hartnett), Vol. 1 (supplement), pp. 379–381 (1978).
7. J. P. Du Plessis and D. G. Kroger, Friction factor prediction for fully developed laminar twisted tape flow, *Int. J. Heat Mass Transfer* **27,** 2095–2100 (1984).
8. S. K. Agarwal, Heat transfer augmentation for the flow of a viscous liquid in circular tubes using twisted tape inserts, Ph.D. thesis, Department of Chemical Engineering, Indian Institute of Technology, Bombay (1991).
9. S. W. Hong and A. E. Bergles, Augmentation of laminar flow heat transfer in tubes by means of twisted tape inserts, *ASME J. Heat Transfer* **98,** 251–256 (1976).
10. S. P. Sukhatme, U. N. Gaitonde, C. S. Shidore, and R. S. Kuncolienkar, Forced convection heat transfer to a viscous liquid in laminar flow in a tube with a twisted tape, *Proceedings of the 9th Heat Mass Transfer Conference*, Bangalore, pp. B-1–B-7 (1987).
11. S. K. Saha, U. N. Gaitonde and A. W. Date, Heat transfer and pressure drop characteristics of laminar flow in a circular tube fitted with regularly spaced twisted tape elements, *Exp. Thermal Fluid Sci.* **2,** 310–322 (1989).
12. W. J. Marner and A. E. Bergles, Augmentation of tubeside laminar flow heat transfer by means of twisted tape inserts, static mixer inserts and internally finned tubes, *Proceedings of the 6th International Heat Transfer Conference*, Toronto, Vol. 2, pp. 583–588. Hemisphere, Washington, DC (1978).
13. W. J. Marner and A. E. Bergles, Augmentation of highly viscous laminar tube side heat transfer by means of a twisted tape insert and an internally finned tube. In *Advances in Enhanced Heat Transfer* (Edited by S. M. Shenkman, J. E. O'Brien, I. S. Habib and J. A. Kohler), pp. 19–28. ASME, New York (1985).
14. R. M. Manglik and A. E. Bergles, Laminar flow heat transfer in a semi-circular tube with uniform wall temperature, *Int. J. Heat Mass Transfer* **31,** 625–636 (1988).
15. R. M. Manglik and A. E. Bergles, A correlation for laminar flow enhanced heat transfer in uniform wall temperature circular tubes with twisted tape inserts. In *Advances in Enhanced Heat Transfer, 24th National Heat Transfer Conference*, Pittsburgh, PA, Vol. HTD-68, pp. 19–25 (1987).
16. M. Monheit, Experimental evaluation of the convective characteristics of tubes with twisted tape inserts. In *Advances in Enhanced Heat Transfer, 24th National Heat Transfer Conference*, Pittsburgh, PA, Vol. HTD-68, pp. 11–18 (1987).
17. A. P. Reddy, Studies on the augmentation of forced convection heat transfer to a viscous liquid in laminar flow in a smooth tube fitted with regularly spaced twisted tape elements, M. Tech. dissertation, Department of Chemical Engineering, Indian Institute of Technology, Bombay (1988).
18. R. S. Van Royan and D. G. Kroger, Laminar flow heat transfer in internally finned tubes with twisted tape inserts, *Proceedings of the 6th International Heat Transfer Conference*, Vol. 2, pp. 577–581. Hemisphere, Washington, DC (1978).
19. C. Shivkumar and M. Raja Rao, Studies on compound augmentation of laminar flow heat transfer to generalized power law fluids in spirally corrugated tubes by means of twisted tape inserts, *Proceedings of the 25th National Heat Transfer Conference*, Houston, TX, Vol. 1, pp. 685–695 (1988).
20. R. M. Manglik and A. E. Bergles, Heat transfer enhancement of intube flows in process heat exchangers by means of twisted tape inserts. In *Heat Transfer Laboratory*, Vol. HTL8. Department of Mechanical Engineering, Aeronautical Engineering and Mechanics, Rensselaer Polytechnic Institute, Troy, NY (1991).
21. A. E. Bergles, A. R. Blumenkrantz and J. Taborek, Performance evaluation criteria for enhanced heat transfer surfaces, *Proceedings of the 5th International Heat Transfer Conference*, Tokyo, Vol. 2, pp. 239–243 (1974).
22. W. J. Marner, A. E. Bergles and J. M. Chenoweth, On presentation of performance data for enhanced tubes in shell and tube heat exchangers, *ASME J. Heat Transfer* **105,** 358–365 (1983).
23. C. C. Monrad and J. F. Pelton, Heat transfer by convection in annular spaces, *Trans. A.I.Ch.E.* **38,** 593–611 (1942).
24. V. Gnielinski, Forced convection in ducts. In *Heat Exchanger Design Handbook*, Vol. 2, Chap. 2.5.1. Hemisphere, Washington, DC (1983).
25. E. N. Seider and C. E. Tate, Heat transfer and pressure drop of liquids in tubes, *Ind. Engng Chem.* **28,** 1429–1435 (1936).
26. H. Hausen, Darstellung des Warmeübergangs in Rohren durch Verallgemeineste Potenz-beziehungen. *Z. Ver. deut. Ing., Beih. Verfahrenstechnik* **4,** 91–98 (1943).
27. D. R. Oliver and V. G. Jenson, Heat transfer to pseudoplastic fluids in laminar flow in horizontal tubes, *Chem. Engng Sci.* **19,** 115–129 (1964).
28. C. A. Depew and S. E. August, Heat transfer due to combined free and forced convection in a horizontal and isothermal tube, *ASME J. Heat Transfer* **93,** 380–384 (1971).
29. S. W. Churchill and R. Usagi, A general expression for the correlation of rates of heat transfer and other phenomena, *A.I.Ch.E. J.* **18,** 1121–1128 (1972).
30. W. J. Marner and A. E. Bergles, Augmentation of highly viscous laminar heat transfer inside tubes with constant wall temperature, *Expl Thermal Fluid Sci.* **2,** 252–267 (1989).
31. A. P. Colburn, A method of correlating forced convection heat transfer data and a comparison with fluid friction, *Trans. A.I.Ch.E.* **29,** 174–210 (1933).