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Effect of tube-tape clearance on heat transfer for fully developed turbulent flow in a horizontal isothermal tube

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This paper presents the effect of tube–tape clearance on the heat transfer for a fully developed turbulent flow in an isothermal tube. Tests were performed on 15 different tapes; they include 3 twisted tape ratios each with five different widths. Results have demonstrated that as the tube–tape clearance decreases, the heat transfer enhancement increases. For the case of twist ratio of 3.6, the tape widths have resulted in almost 17% difference in the enhancement. On the other hand, for the cases of twist ratios of 5.4 and 7.1, the enhancements were 9% and 4%, respectively. For practical design of thermal systems, operating under turbulent flow condition, small twist ratio and tight-fit tape are desirable in order to obtain a reasonably high heat transfer enhancement. The heat transfer for twist ratio of 3.6 and loose-fit tape shows nearly the same enhancement as the case of the tight-fit tape for the same twist ratio. Such behavior indicates the presence of an optimum tape width, and such width is a function of the twist ratio and Reynolds number.

Introduction

Twisted tape inserts are widely used in industry as a means to increase the heat transfer coefficient inside tubes. A considerable heat transfer enhancement can be achieved when using twisted tape inserts at a reasonable increase in pressure drop. Because of their low cost and ease of manufacture installation, they can be used to produce compact heat exchangers and to upgrade the thermal performance of existing shell-and-tube heat exchangers.

A literature review has demonstrated the existence of an extensive research regarding the use of twisted tape inserts for heat transfer augmentation in tubes. Numerical solutions for laminar and turbulent flows were reported by Date (1974) and Date and Saha (1990), respectively, for the case of uniform heat flux boundary condition. Also Hong and Bergles (1976), Marner and Bergles (1978), and Sukhatme et al. (1987) showed experimentally the effect of twisted tape on heat transfer and pressure drop under uniform heat flux condition in laminar flow in tubes. On the other hand, Duplessis and Kröger (1983); Monheit (1987), and Marner and Bergles (1989) conducted a numerical and an experimental studies, respectively, for the constant wall temperature boundary condition under laminar flow. Lopina and Bergles (1969) developed correlations for turbulent flow conditions that are widely used in the heat exchanger industries. More recently, Manglik and Bergles (parts I and II 1993) performed an extensive literature review (over thirty citations) together with their experimental findings. They have developed practical and more general correlations that can be used for practical design to estimate heat transfer and pressure drop in laminar and turbulent flows.

In an effort to produce more general and practical correlations for twisted tape insert turbulent flows, Manglik and Bergles (part II 1993) have pointed out clearly that there is much discrepancy in the existing data, and the existing correlations are based on few and scattered datapoints in most cases. Hence, they took the initiative to conduct more experiments to produce more data, and, based upon that, they came up with more general and practical correlations for heat transfer and pressure drop.

Although the previous investigations are important and comprehensive, it was found that the literature regarding the influence of twisted tape width on the heat transfer and pressure drop is scarce. Lopina and Bergles (1969) had developed a practical correlation that includes the influence of tape width on heat transfer in turbulent flows. They reported an increase in the Nusselt number as much as 20% for the tight-fit tape over that of the loose-fit tape. On the other hand, Manglik and Bergles (part II 1993), have reported that the gain in the heat transfer for the snug-fit tape is negligible as compared to the loose-fit tape. Recently, Ayub and Al-Fahed (1993) have conducted extensive experimental research on the influence of twisted tape width on the pressure drop. Their findings indicated the possibility of the existence of an optimum tape width, which might be a function of both the twist ratio and the Reynolds numbers.

As stated before, there is a need for more twisted-tape insert data. Also there is a lack of information in the literature related to the influence of tape width on heat transfer. These motivated the present work. In this study heat transfer data were collected for fifteen tapes (three twist ratios, each with five different widths). The physical dimensions of the tapes are shown in Table 1. The results of the tight-fit cases are correlated and compared with

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Table 1 Twisted tape specifications

	Tape width (mm)				
$Y = H/D_i$	10.8	11.4	12.0	12.6	13.2
3.6	A	B	C	D	E
5.4	F	G	H	I	J
7.1	K	L	M	N	O

previous works. In addition, the influence of the twisted tape width on heat transfer is analyzed.

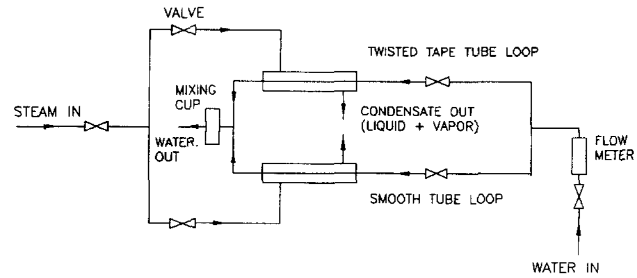

Figure 1 Schematic diagram of the heat transfer facility

Test facility and procedure

The test facility is shown in Figure 1. It includes two test sections of a single tube-in-tube heat exchanger. Each test section consists of a smooth copper tube with an outside diameter of 15.9 mm and a wall thickness of 0.95 mm. Each test section has a nominal length of 1000 mm and is preceded by a calming length of 1400 mm. Saturated steam was introduced in the annulus of each section and was used as an isothermal source. Chilled water was introduced in the test tubes and was used as the working fluid. The whole system was insulated by wrapping insulation around all the exposed parts of the system.

The other components of the facility include a flow meter, thermocouples, and a data acquisition system. Two thermocouples were installed across each test section to measure the inlet and outlet temperatures of the water. Another thermocouple was installed in a mixing cup at the outlet. It was found that the outlet and the mixing cup thermocouples readings differ by only ± 0.10 K. This served as a validity check for the thermocouples. Also, five thermocouples were inserted in the annulus of each test section to monitor the steam temperature to ensure that saturation existed during the test runs; hence, an isothermal condition in the annulus was established.

The wall temperature of each test section was monitored by 15 thermocouples installed on the outside of the tube wall (see Figure 2a). Such thermocouples were installed in five axial locations along the wall of the tube. At each location, three thermocouples were installed 120° apart around the circumference of the tube. Each thermocouple was soldered in a groove inside the wall. The groove, which was filed in the outer wall of the tube, has a 0.5-mm depth. After soldering the wall thermocouples, an insulating high temperature silicon was put on top of the place where the thermocouples were soldered to insulate them from direct exposure to the steam in the annulus. All the thermocouples were 30-gauge copper constantan.

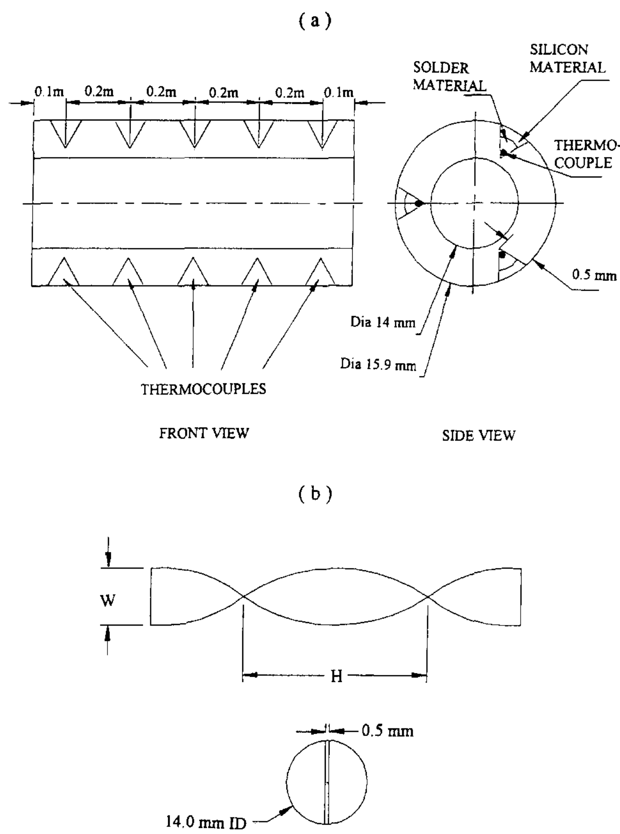

Figure 2 (a) Tube wall thermocouples; (b) Twisted tape insert

Notation

A_i	inside surface area of tube
C_p	specific heat
D_i	inside diameter of tube
f	fannig friction factor
H	pitch, based on 180°
h_i	inside mean heat transfer coefficient
k	thermal conductivity for water
$\dot{m}$	mass flow rate for water
Nu_m	mean Nusselt number
Pr	Prandtl number
Q	rate of heat transfer
Re	Reynolds number
T_i	water inlet temperature
T_o	water outlet temperature
T_w	local wall temperature
W	tape width
Y	tape twist ratio, H/D_i

Greek

δ	twisted tape thickness
μ	dynamic viscosity

Subscripts

b	at bulk temperature
w	tube wall
$\dot{m}$	average value

The data, collected by a data acquisition system, revealed that most of the test runs needed 10 minutes to attain a steady-state condition. In this work, a waiting period of 15 minutes was used for all runs following each flow change. Before collecting the data, the thermocouple readings were checked against a calibrated thermometer to ensure that the thermocouples were responding accurately. The average uncertainty was observed to be less than ± 0.10 K.

Fifteen different twisted tape inserts, as shown in Figure 2b, were used in this study and these are: three twist ratios each with five different widths. The twisted tape was fixed at both ends inside the tube using strings to keep it in place and to prevent it from sliding when the flow passes through. To fix the twisted tape inside the tube, holes were drilled at the center ends of it, then fine strings were passed through the holes and through the tube wall. Following that, the tape ends were centered, and the strings were tightened and fixed to the tube wall. Such a procedure did not prevent the twisted tape in the loose cases to deviate slightly from its center position. However, this simulates the actual insertion of a twisted tape in heat exchanger applications.

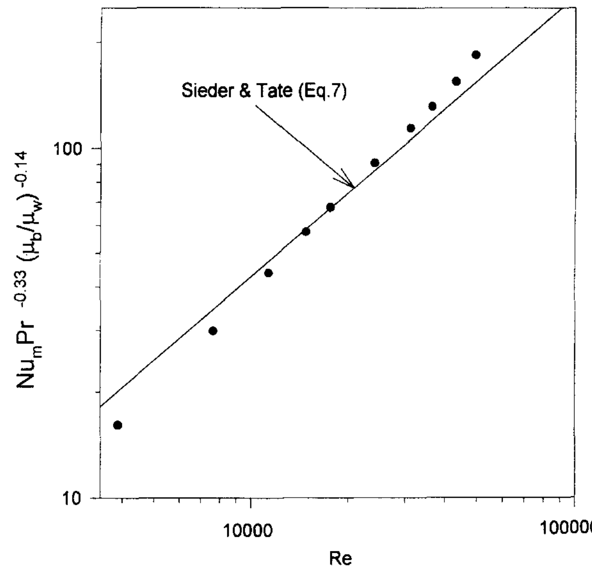

Figure 3 Verification of the plain tube case

Data reduction equations

The average inside heat transfer coefficient and the mean Nusselt number for the plain and the twisted tape insert cases were evaluated as follows:

$$Q = \dot{m}C_p(T_o - T_i) \tag{1}$$

$$Q = h_i A_i(\bar{T}_w - T_b) \tag{2}$$

where

$$T_b = (T_i + T_o)/2 \tag{3}$$

and

$$\bar{T}_w = \sum T_w/15 \tag{4}$$

T_w is the local wall temperature of the tube.

It must be noted that the local wall temperature T_w was measured within the tube wall (at a position of 0.45 mm from the inner wall). Analysis has demonstrated that such a measurement technique for the wall temperature resulted in a maximum error of 2% in the Nusselt number, because the wall resistance was neglected. Furthermore, if we are interested in comparing the plain tube and or the twisted-tape heat transfer, then such an error would not influence the results (Chakroun et al. 1993).

The average inside heat transfer coefficient and the mean Nusselt number were determined by:

$$h_i = \dot{m}C_p(T_o - T_i)/A_i(\bar{T}_w - T_b) \tag{5}$$

$$\mathrm{Nu}_m = h_i D_i/k \tag{6}$$

All properties were evaluated at the mean bulk temperature.

The uncertainty in the determined Nusselt number was estimated based on ANSI/ASME (1986) together with the procedure used by Coleman and Steele (1989). The overall uncertainty for the Nusselt number was found to range from 4 to 6%, depending on the flow conditions.

Results and discussion

The heat transfer data for the plain tube were collected first. Such data served as a check for the validity of the experimental setup and measurement techniques over the whole range of flow conditions. Figure 3 presents the data of mean Nusselt number *versus* Reynolds number, together with those evaluated from Sieder and Tate (1936) correlation

$$\mathrm{Nu}_m = 0.027\ \mathrm{Pr}^{0.33}\ \mathrm{Re}^{0.8}(\mu_b/\mu_w)^{0.14} \tag{7}$$

The plain tube data are shown to agree well with the Sieder and Tate (1936) correlation, which clearly indicates the validity of the setup and procedure. Following that, tests were performed on the 15 tapes and data were collected. Prior to investigating the influence of the tape width on the heat transfer, the data for the tight-fit tapes were compared with some known correlation. That was done to make sure that the twisted tape experimental technique was valid. Data obtained for the tight-fit twisted tapes were plotted together with the correlation taken from Manglik and Bergles (Part II 1993)

$$\mathrm{Nu}_m = \mathrm{Nu}_{my=\infty}(1.0 + 0.769/Y) \tag{8}$$

where, $\mathrm{Nu}_{my} = \infty$ is the mean Nusselt number for the straight tape case, and is expressed as follows:

$$\mathrm{Nu}_{my=\infty} = 0.023\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{0.4}(Z)^{0.8}((\pi + 2 - 2\delta/D_i)/Z)^{0.2} \times (\mu_b/\mu_w)^{0.18} \tag{9}$$

where

$$Z = \pi/(\pi - 4\delta/D_i) \tag{10}$$

Substituting for $\delta = 0.5$ mm and $D_i = 14.0$ mm into Equation 10 and using the resulting value for Z into Equation 9, we obtain

$$\mathrm{Nu}_{my=\infty} = 0.02651\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{0.4}(\mu_b/\mu_w)^{0.18} \tag{11}$$

Using Equations 8 and 11, we can obtain the following expression for Nu_m

$$\mathrm{Nu}_m = 0.02651\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{0.4}(\mu_b/\mu_w)^{0.18}(1.0 + 0.769/Y) \tag{12}$$

Equation 12 together with the heat transfer data obtained from this work, for $Y = 3.6$, 5.4, and 7.1, are shown in Figure 4. It can be seen that the experimental data obtained from this work agree reasonably well with Equation 12. Such validation and the plain tube validation that was performed previously gave a reasonable confidence in the setup and the experimental techniques used in this work.

At this stage, the effect of the twisted tape width on the heat transfer was investigated. The heat transfer data for the three different twist ratios ($Y = 3.6$, 5.4 and 7.1) and five tape widths ($W = 10.8$ mm, 11.4, 12, 12.6, and 13.2) are shown in Figures

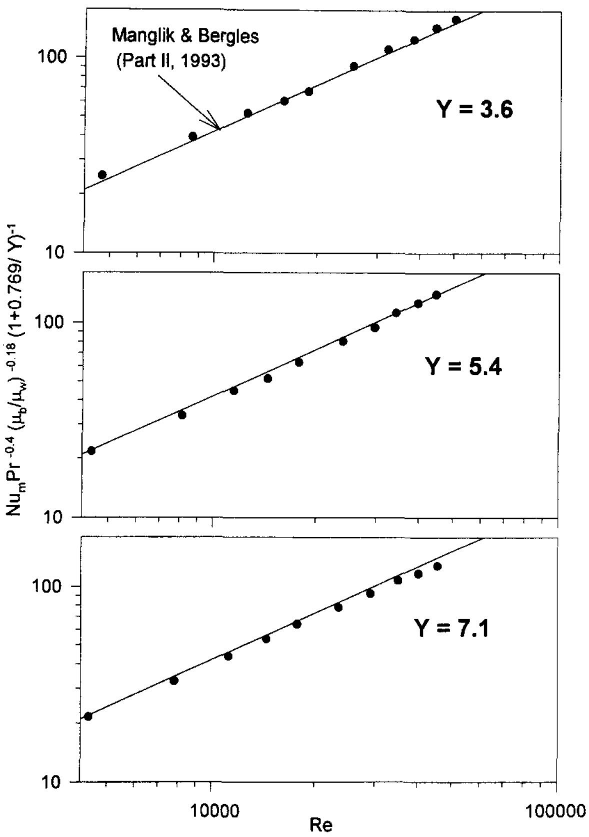

Figure 4 Comparison of heat transfer data for tight fit twisted tape inserts with Equation 12

5–7. In these figures, the heat transfer data are shown together with that obtained for the plain tube. In all cases, except that of $Y = 3.6$ and $W = 10.8$ mm, the data show that the heat transfer increases with the width of the tape. This increase reaches a maximum value for the tight-fit tape ($W = 13.2$ mm). Such behavior is seen at all Reynolds numbers, and it is more pronounced at higher values.

Figure 5 shows a considerable increase in the heat transfer for the case of twist ratio equal to 3.6 as compared to the plain tube.

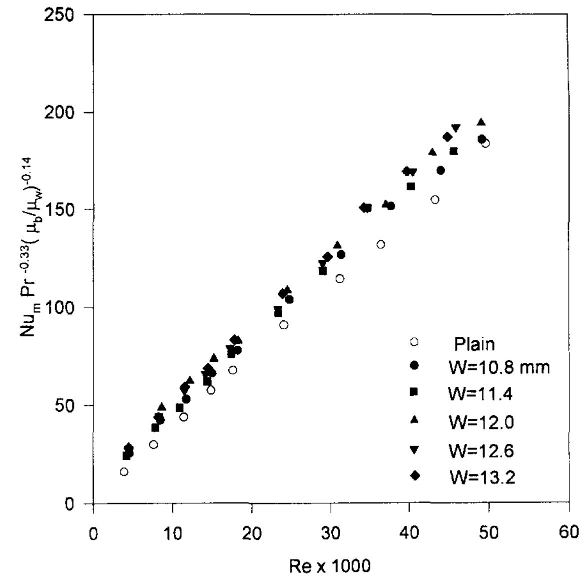

Figure 6 Heat transfer data for $Y = 5.4$ and plain tube

For the case of $Y = 3.6$, and $W = 13.2$ mm, the heat transfer gain was as much as 22% at Reynolds number of 50,000. On the other hand, for the case of $Y = 3.6$ and $W = 11.4$ mm, the heat transfer gain was nearly 5%. Hence, for these cases, the tape widths have resulted in 17% difference in the enhancement.

Figures 6 and 7 display similar behavior to that of Figure 5, however the gain in the heat transfer is observed to be less. For the cases of $Y = 5.4$ and 7.1, the tape widths resulted in differences in the heat transfer enhancements equal to 9% and 4%, respectively. Overall, there was a trend in the influence of the tape width on the heat transfer, that is for a given twist ratio, the heat transfer increases as the tape width increases. Such behavior could be explained as follows. As the twist ratio decreases, the swirl produced by the presence of the tape dominates; hence, the heat transfer gain as compared to the plain tube also increases. Moreover, as the tape width increases, more blockage is introduced, which results in an increase in the swirl velocity, and in turn increases the heat transfer.

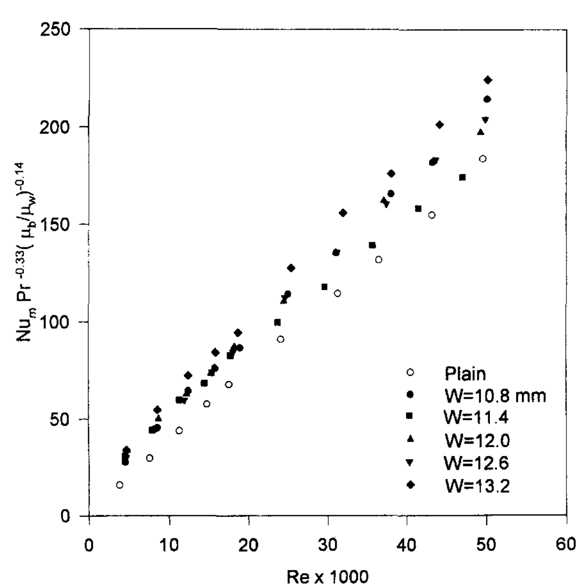

Figure 5 Heat transfer data for $Y = 3.6$ and plain tube

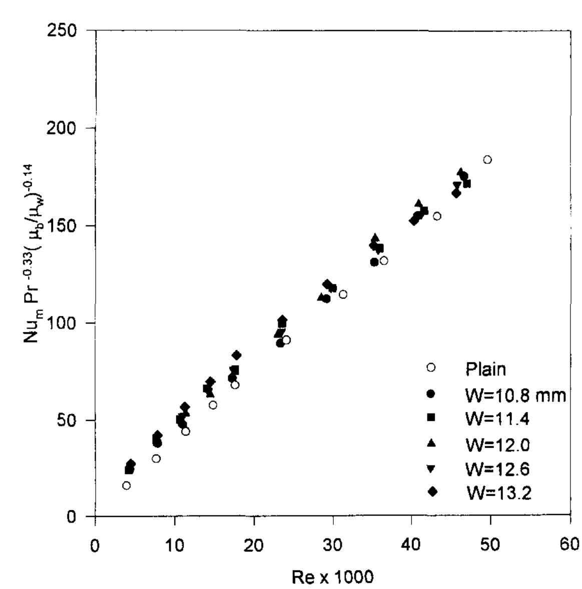

Figure 7 Heat transfer data for $Y = 7.1$ and plain tube

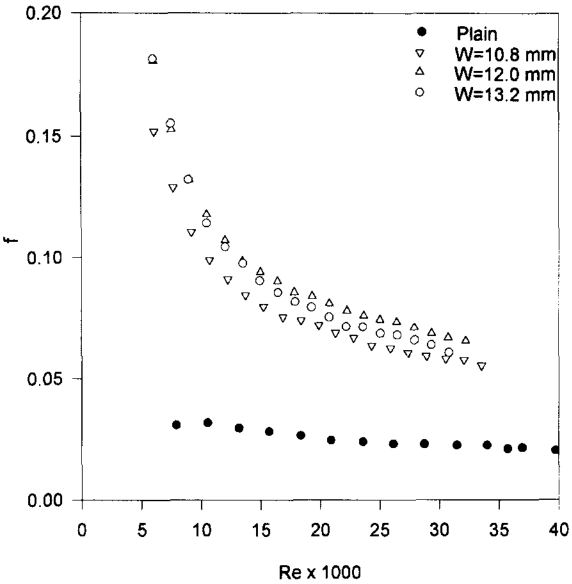

Figure 8 Friction factor for $Y = 3.6$ and plain tube (Ayub and Al-Fahed 1993)

Based on the previous analysis and discussion, we would expect that the tape width of 10.8 mm (loose fit), for the case of $Y = 3.6$, would result in the minimum enhancement for this particular case. However, the data show otherwise; that is, that the heat transfer enhancement is nearly the same as that of tape width of $W = 13.2$ mm (tight fit). Such behavior could be explained as follows. For the case of a small twist ratio ($Y = 3.6$), the swirl velocity was high; hence, if the clearance between the tape and the inner wall of the horizontal tube was relatively big, this results in a large mass of fluid circulating around the tape, and because of that, a large enhancement was observed.

It is worth mentioning that Ayub and Al-Fahed (1993) have reported that for some Reynolds number, the pressure drop increases with the tape width up to a certain value, after which the pressure drop starts to decrease. To complement the heat transfer results reported here, the fricion factor data for the twist ratio of 3.6 and tape widths of 10.8 mm, 12.0, and 13.2 from Ayub and Al-Fahed (1993) are reported in Figure 8. Plain tube friction data are also shown in the same figure.

From Figure 8, it is seen that the friction factor for $Y = 3.6$ is about 2.5–4.0 times that of the plain tube depending on the Reynolds number. From the results shown, it can also be seen that for Reynolds number less than 8000, the friction factor increases as the tape width increases; however, for higher values of Reynolds number, the friction factor increases with the tape width, then it decreases. The friction factor for tape width of 12.0 mm is higher than that of 10.8 mm (loose fit); however, the friction factor of the 13.2 mm tape (tight fit) is lower than that of 12.0 mm. The behavior of the friction factor for $Y = 3.6$ was also seen in the cases of $Y = 5.4$, and $Y = 7.1$ (for complete details, see Ayub and Al-Fahed 1993). The behavior of the friction factor reported here is not surprising. Because the tube–tape clearance decreases with the increase of the width of the tape, the fluid that circulates in the gap between the tape and the inner wall of the tube will have higher velocity, which in turn, changes the flow field and produces more pressure drop. On the other hand, when the tape is tight-fit ($W = 13.2$ mm), the circulation is not as much; therefore, the pressure drop decreases.

The result shows that the increase in heat transfer and the pressure drop are strongly influenced by the helical motion induced by the twisted tape. As the twist ratio decreases, the swirl flow increases, which in turn, results in an increase in the heat transfer. In addition, small twist ratio and tight-fit tape results in a reasonable pressure drop. Therefore, a small twist ratio and a tight-fit tape (small tube–tape clearance) would produce a high heat transfer at a reasonable pressure drop.

Conclusions

An experimental study was performed on 15 twisted tape inserts in a horizontal tube using water as the test fluid. Three twist ratios ($Y = 3.6$, 5.4, and 7.1) and five widths ($W = 10.8$ mm, 11.4, 12.0, 12.6, 13.2) were investigated. The heat transfer results for all the twist ratios and tight-fit twisted tapes ($W = 13.2$ mm) have shown a good agreement with previous results.

The influence of the twisted tape width on the heat transfer has been demonstrated. As the tape width increases, the heat transfer increases. On the other hand, for the small twist ratio ($Y = 3.6$) and tape width ($W = 10.8$ mm) an interesting behavior was noticed: such a case resulted in an increase in the heat transfer nearly as much as the tight-fit tape for the same twist ratio. This indicates that there might be an optimum tape width, which is possibly a function of the twist ratio and Reynolds number. A numerical investigation is in progress to substantiate those findings.

For practical design of thermal systems that operate on turbulent flows, the decision for the choice of the twist ratio and tape width must be weighed carefully. Results have demonstrated that the twist ratio equal to 3.6 has produced a reasonable heat transfer enhancement. On the other hand, twist ratios equal to 5.4 and 7.1 have shown a small enhancement. In addition, tight-fit tapes resulted in a high heat transfer enhancement as compared to loose-fit tapes. Thus, it seems desirable to use a small twist ratio and a tight-fit tape to obtain a reasonable high heat transfer enhancement at the expense of a reasonable pressure loss.

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