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HEAT TRANSFER ENHANCEMENT BY A COMBINATION OF SPIRALLY CORRUGATED TUBES WITH A TWISTED TAPES PART 1 - HYDRAULIC RESISTANCE AND HEAT TRANSFER COEFFICIENTS

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ABSTRACT

Heat transfer and isothermal friction pressure drop results have been experimentally obtained for two single-start spirally corrugated tubes combined with five twisted-tape inserts with different relative pitches in the range of Reynolds number $10^4 - 7\times10^4$. The characteristic parameters of the tubes are height to diameter ratio, e/Di=0.0371 and 0.0261; pitch to height ratio, p/e=11.6 and 15.9; and relative pitch, H/Di= 15.68-5.80. Significantly, higher friction factor and inside heat-transfer coefficients than those of the smooth tube under the same operating conditions have been observed.

NOMENCLATURE

D	tube diameter, m
e	rib height, m
p	pitch of corrugation, m
H	pitch of twist, m
h	heat transfer coefficient, W/(m^{2}K)

Greek symbols

β	helix angle of rib (deg)

Dimensionless groups

β_*	$\beta/90$
f	Fanning friction factor
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

Subscripts

i	inside
o	outside
R	rough tube
S	smooth tube

INTRODUCTION

The characteristics of modern heat exchangers can be significantly improved through various techniques and technologies for enhancement of heat transfer, as a result of which highly efficient energy systems are built [1,2]. These technologies can be applied to existing heat exchangers or to the creation of new ones.

The methods used to increase the efficiency of the devices are active and passive. Enhancement of heat transfer in passive methods that are most commonly studied and used can be simple or compound.

The choice of a particular technique depends on many factors, namely: thermo-hydrodynamic characteristics; production costs; pollution and the possibility of cleaning; availability of required material; specific practical application.

Some of the existing methods for enhancing heat transfer in a single-phase, fully developed turbulent flow in a round tube are one of the two types: a) methods in which the inner surface of the tube is roughened, e.g., with repeated or helical ribbing, by sanding, or with internal fins, turbulizers, obtained by cold rolling of smooth tubes - cross-rolled or spiral-rolled (spirally corrugated); and b) flow-turning devices such a twisted tapes, discs, augers or various inserts located in the tube.

It is well known that two or more of the existing techniques can be utilized simultaneously to produce an enhancement larger than that produced by only one technique. Interactions between different enhancement methods contribute to greater values of the heat transfer coefficients compared to the sum of the corresponding values for the individual techniques used alone. Preliminary studies in compound passive enhancement techniques were encouraging: rough tube wall with twisted tape inserts [3] and grooved rough tube with twisted tape [4].

Many previous surveys [5,6] indicated that the corrugated tubes are among the most effective and practical methods for enhancing single-phase heat transfer in tubes. It is reasonable to assume that a combination of a corrugated tube with twisted tape would be superior to either passive surface technique acting alone.

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EXPERIMENTAL RESULTS

This study reports on the part of a comprehensive experimental program to see whether or not heat transfer can be enhanced by the multiplicative effect of a corrugated tube combined with a twisted tape and how the characteristic parameters of the tube affect the friction factor and inside heat transfer coefficient.

In this part of the experimental program, thermo-hydrodynamic characteristics of two single-start spirally corrugated tubes of varying geometries combined with four twisted tapes with different pitches have been studied. The smooth tube had an outside diameter of 16.0 mm with wall thickness of 1.0 mm before the rolling operation. A smooth tube was used for standardizing the experimental set-up and for comparing the enhancement in heat transfer and fluid friction. The appearance of the tubes and tapes are shown in Fig. 1. The characteristic parameters – outside encase diameter D_o, inside diameter D_i, pitch of corrugation *p*, height of corrugation *e*, spiral angle β and dimensionless parameters e/D_i, p/e and $\beta_* = \beta/90$, are presented in Table 1 and Fig. 2.

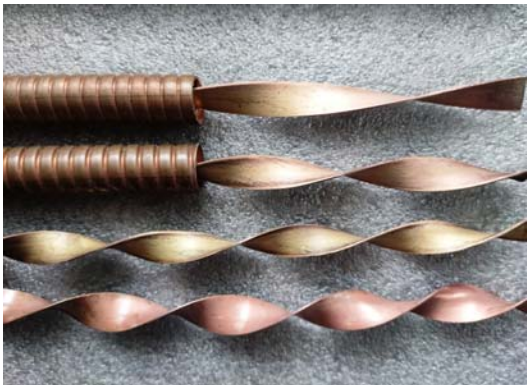

Fig. 1

Table 1 *Geometrical parameters of the corrugated tubes*

Tube №	D_o mm	D_i mm	*p* mm	*e* mm	β (°)	e/D_i -	*p/e* -	β* -
340	15.56	13.39	5.77	0.497	82.2	0.0371	11.6	0.913
360	15.67	13.78	5.80	0.359	82.5	0.0261	15.9	0.917

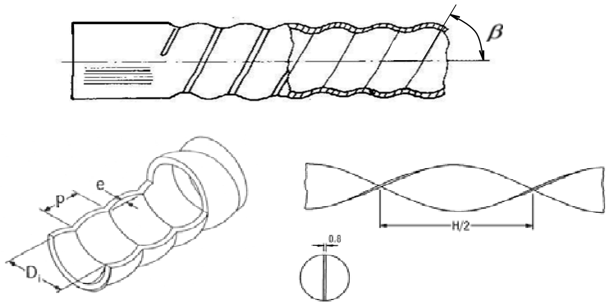

Fig. 2. Characteristic parameters of the corrugated tube and the twisted tape

With the length of the tape in 360° twist defined as H, four different tapes were used: H = 210, 168, 106, 80 mm.

Full details of the experimental set-up consisting of two heat transfer tubes 1200 mm long in a conventional recalculating system can be found in [7]. The pressure difference across the heat transfer section was measured and friction factors were calculated. The heating section was heated by condensing steam generated into a boiler and water was used as the cooling fluid. The flow rate of the test fluid, the tube wall temperatures, inlet and outlet water temperatures and the temperature of the saturated steam were measured.

Coefficient of hydraulic resistance

The isothermal pressure drop studies were conducted at different temperatures of the water in the range from 72 to 92°C in all tubes over a range of Reynolds number $10^4 < \mathrm{Re} < 7\times10^4$. The pressure drop data for the tubes were converted to Fanning friction factors as shown in Fig. 3 and 4.

The experimentally determined coefficients f for the smooth tube correlated with the Blasius equation

$$f_S = 0.079\, Re^{-0.25}, \quad (1)$$

with a standard deviation of ±3%, which is proof of the accuracy of the collected experimental information.

A characteristic feature of the flow is that even at high flow rates, the friction factor continues to decrease with the increase of Re although not so rapidly as in the smooth tube. The isothermal friction coefficients for straight flow and swirl flow in the corrugated tubes increase when the relative pitch H/D_i decreases.

The friction factors for all the tubes investigated are related to the Reynolds number by the following equation, which has been widely accepted in the literature

$$f_R = c_f\, Re^m . \quad (2)$$

The values of c_f and m were determined for each tube and Eq. (2) represents the data within 2% standard deviation. The values for c_f and m are listed in Table 2 and the experimentally obtained results are shown in Fig. 3 and 4.

Table 2 *The values of c_f and m (Eq.(2))*

№	H/D_i	c_f	m
340	-	0.043	-0.052
341	15.68	0.172	-0.121
342	12.56	0.210	-0.135
343	7.96	0.177	-0.106
344	5.98	0.229	-0.119
360	-	0.072	-0.155
361	15.24	0.161	-0.167
362	12.20	0.121	-0.137
363	7.74	0.200	-0.173
364	5.80	0.254	-0.187

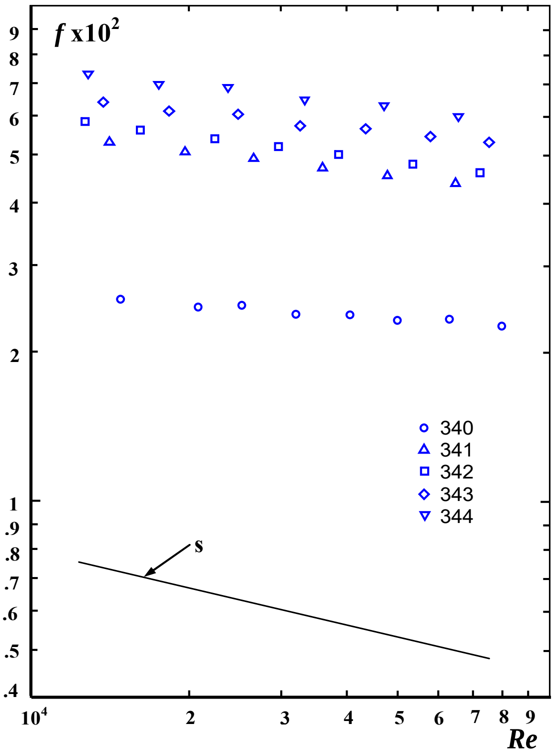

Fig. 3. Friction factor vs. Reynolds number

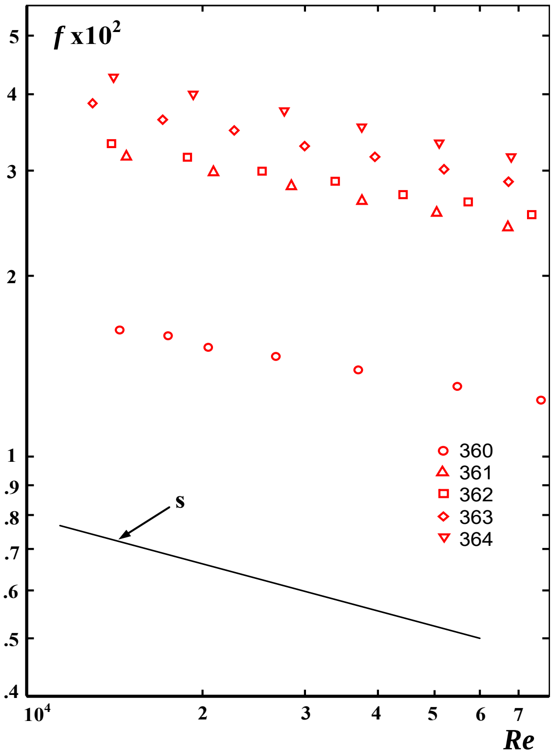

Fig. 4. Friction factor vs. Reynolds number

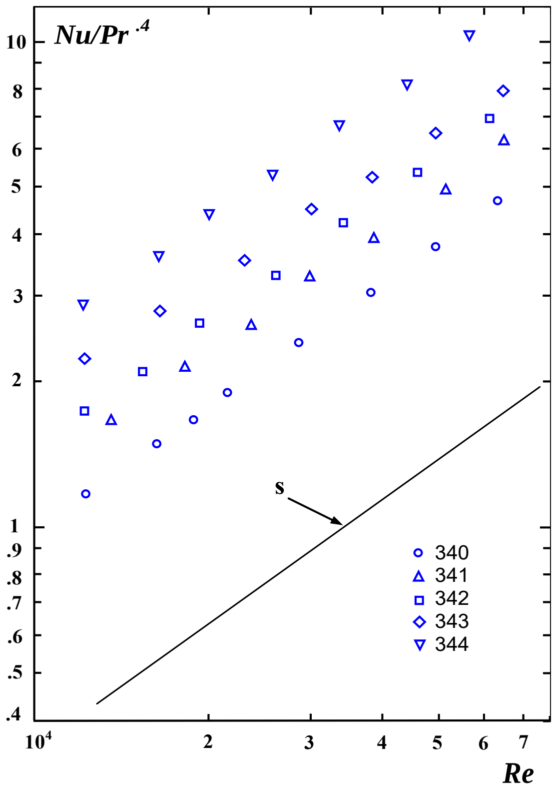

Fig. 5. Heat transfer coefficients vs. Reynolds number

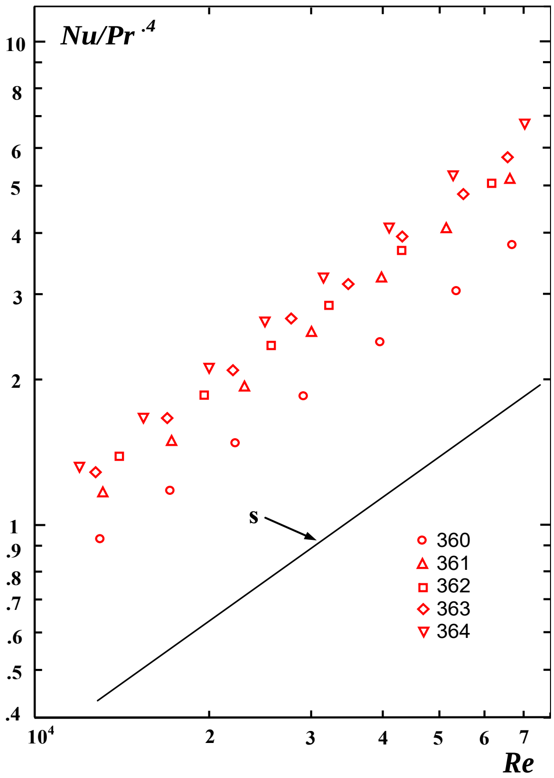

Fig. 6. Heat transfer coefficients vs. Reynolds number

Heat transfer coefficient of cooling water

Heat transfer studies in spirally corrugated tubes were carried out to obtain values for the water side heat transfer coefficients h_i and the steam condensing coefficients h_o. Since the metal wall temperatures were measured, the individual film coefficients were determined. The smooth tube heat transfer coefficients were found to agree with ±3% of the Dittus–Boelter correlation

$$Nu_S = 0.023\,Re^{0.8}\,Pr^{0.4} \qquad (3)$$

in the whole range $10^4 < Re < 7\times10^4$.

Figs. 5 and 6 show the heat transfer data in the form $Nu_i\,Pr^{-0.4}$ as a function of the Reynolds number Re and can be presented with by the following equation

$$Nu_R\,Pr^{-0.4} = c_h\,Re^n. \qquad (4)$$

The values of the constants c_h and n are presented in Table 3, as Eq. (4) correlates the experimental results with a standard deviation of ±5%.

Table 3 *The values of c_h and n (Eq.(4))*

№	c_h	n
340	0.044	0.839
341	0.062	0.830
342	0.052	0.864
343	0.080	0.846
344	0.126	0.823
360	0.035	0.830
361	0.020	0.915
362	0.038	0.857
363	0.028	0.893
364	0.026	0.910

ANALYSIS OF RESULTS

The analysis of the information presented in Fig. 5 and 6 show that when combining spirally corrugated tubes with twisted tapes, the heat transfer coefficients h_i are significantly increased compared to those of spirally corrugated and even more than those of the smooth tube. This increase occurs when e / D_i increases and H / D_i decreases and is most pronounced when combining spirally corrugated tubes with a twisted tape with the smallest step. The coefficients of hydraulic resistance for spirally corrugated tubes with tape also increase significantly, with the largest increase observed in the combination with tape with the smallest step. However, as can be seen from Fig. 3 and 4 the flow regime does not reach the area of "hydraulically rough" mode. Resistance coefficient f continues to decrease at an increase of *Re*, albeit more weakly than the smooth tube.

The values of ratios for Nu_R / Nu_S and f_R / f_S in the studied range of *Re* are presented in Table 4.

Table 4 *The values of ratios*

№	Nu_R / Nu_S	f_R / f_S
340	2.85 – 3.30	3.36 – 4.80
341	3.17 – 4.32	7.16 – 8.28
342	3.50 – 5.30	7.74 – 9.33
343	5.12 – 5.85	8.62 – 11.5
344	7.03 – 6.83	9.69 – 12.25
360	2.22 – 2.34	2.78 – 2.42
361	2.56 – 3.45	4.30 – 4.66
362	2.62 – 3.84	4.32 – 5.00
363	2.65 – 3.78	5.07 – 5.78
364	3.23 – 3.66	5.75 – 6.44

It can be seen that the highest value of Nu_R / Nu_S (more than 7.0) is for combination 344 with the lowest value of H / D_i. The highest values of f_R / f_S (more than 12.0) were obtained for the same combination.

It is obvious that the thermo-hydrodynamic characteristics of the tube are a major factor in the selection. For this, however, a criterion is needed to give a quick assessment of the properties of one turbulizer compared to another. Such a criterion for rapid evaluation was first formulated by the Bergles [8]

$$\left(h_R / h_S\right)_{P,D,L} = \text{f (Re,Pr, turbulizer geometry)}. \quad (5)$$

The turbulizer, which has a higher coefficient h_i at the same pump energy consumption, a certain inner diameter of the pipe and a specific fluid flowing in it, is more efficient in terms of energy savings and operating costs.

A suitable expression for this criterion is the idea of Sano and Usui [9] that the increase of h_i can be expressed by the dissipation of energy per unit mass of fluid flowing in the heat exchanger, which leads to the criterion

$$i_E = \frac{Nu_R / Nu_S}{\left(f_R / f_S\right)^{0.291}} = f\left(\mathrm{Re}_R\right). \quad (6)$$

It should be emphasized that the use of the criterion Eq. (6) can only serve as a preliminary assessment of the properties of a turbulizer. He cannot give an overall estimate of the profit that would be realized by replacing standard smooth pipes with those in which turbulizers are created. This assessment is made through other criteria.

The values of i_E for each of the tested tubes are presented in Table 5. From the presented results it is noticed that the values of i_E for tubes 341 and 342 are lower than those of tube 340, for the low values of *Re*, while the others - 343 and 344 have higher values in the whole range. The results are similar for tubes 361, 362 and 363, compared to 364. From all combinations it can be seen that for tube 344, i_E reaches the highest value 3.63. It can also be seen that tube 340 is better than 360, both without and in combination with tapes.

Table 5 *The values of i_E*

№	i_E	№	i_E
340	2.00 – 2.09	360	1.88 – 1.81
341	1.79 - 2.33	361	1.67 - 2.20
342	1.93 - 2.77	362	1.71 - 2.40
343	2.73 - 2.87	363	1.65 - 2.27
344	3.63 - 3.29	364	1.94 - 2.13

CONCLUSIONS

Experimental results show that the combination of spirally corrugated tubes with a spirally twisted tape with a larger pitch ($H / D_i > 7.96$) deteriorates the energy efficiency. The conclusion that can be made is that the application of twisted tapes in combination with turbulizers does not always lead to good results. An accurate assessment of both the geometrical characteristics of the turbulizer and the pitch of the tape with which it is combined is required.

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