Heat transfer augmentation of a spiralled tube inside the annulus of a tube-in-tube heat exchanger

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The potential of a simple and inexpensive method of heat augmentation that could be manufactured relatively easily by small manufacturing companies is investigated. It consists of a tube that is turned spirally and tightly around the outer surface of the inside tube of a tube-in-tube heat exchanger. The aim is to increase the heat transfer between the flow in the inner tube and the flow in the annulus by increasing the heat transfer coefficient in the annulus. This investigation was conducted with measurements taken at five different twist angles: the first without a spiralled tube and the rest twisted at angles corresponding to a pitch 1D, 2D, 3D, and 4D. The Nusselt number and friction factor in the annulus of each case study were measured as functions of the Reynolds number, while the Prandtl number was kept constant, and for each heat exchanger configuration an equation was deduced that describes this functional relationship. It was found in general that the geometry considered is appropriate for heat transfer enhancement and that the highest heat transfer per unit pumping power occurs at a pitch of 4D. It is recommended that more work should be conducted on this geometry, specifically at pitches of 4D and higher.

Nomenclature

a	length of major elliptical axes of the twisted tube [m]
A	cross-sectional flow area in the annulus [m^2]
A_i	inside area of inner tube [m^2]
A_o	outside area of inner tube [m^2]
b	length of minor elliptical axes of the twisted tube [m]
C_p	specific heat of water at constant pressure [J/kg.K]
C, C_1, C_2	constants used in Equations 2 and 3
D	outside diameter of inner tube [m]
D_h	hydraulic diameter of the annulus [m]
D_i	inside diameter of inner tube [m]
D_{oi}	inside diameter of outer tube [m]
f	Darcy-Weisbach friction factor $(2D_h\Delta p/\rho.V^2.L)$ [ ]
h_i	convection coefficient in the inner tube [W/m^2.K]
h_o	convection coefficient in the annulus [W/m^2.K]
k	thermal conductivity of inner tube [W/m^2.K]
k_f	thermal conductivity of water in the annulus at annulus bulk temperature [W/m^2.K]
L	length of inner and outer tubes [m]
m_1	constant in Equation 2
Nu	Nusselt number in the annulus (h_oD_h/k_f) [ ]
P	total length of the wetted perimeter in the annulus [m]
P_i	length of wetted perimeter of inner tube on the outside diameter [m]
Pr	Prandtl number in the annulus at bulk temperature $(C_p\mu/k_f)$ [ ]
P_s	length of wetted perimeter of twisted tube measured on outside of the tube [m]
Δp	pressure drop of water in the annulus [Pa]
Q	heat transfer to the annulus [W]
t	pitch of twisted tube [m]
Re	Reynolds number for the annulus $(\rho VD_h/\mu)$
U_o	overall heat transfer coefficient based on A_o for the annulus [W/m^2.K]
V	velocity of water in annulus [m/s]
Wp	pump power [W]
ρ	density of water in annulus at bulk temperature [kg/m^3]
μ	viscosity of water in annulus at bulk temperature [Ns/m^2]

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Introduction

Techniques[1] applied to improve the efficiency of heat exchangers or to augment heat transfer can be classified as passive methods which require no direct application of external power, or as active schemes which require external power. Examples of passive techniques include rough surfaces, extended surfaces, displaced promoters, and vortex devices. Examples of active techniques include mechanical aids, surface vibration, fluid vibration, and electrostatic fields. The effectiveness of a given augmentation technique depends largely on the mode of heat transfer, which may range from single-phase free convection to dispersed-flow film boiling, or on the type of heat exchanger to which it is applied.

Several examples of passive tube-in-tube heat exchangers with augmentation techniques are described in technical literature.[1-7] Rough surfaces of the spiral-repeated rib variety are widely used to improve in-tube heat transfer with water, as in flooded chillers. The roughness may be produced by spirally indenting the outer wall, forming the inner wall, or inserting coils. Internal fins in tubes, longitudinal or spiral, can be produced by extrusion or forming, with a substantial increase in the surface area. Twisted tape can be inserted as original equipment or as a retrofit device and is one of the most important members of passive heat transfer augmentation.[8]

Although all these techniques of heat transfer enhancement exist, they are usually not economically viable for small companies manufacturing original equipment (i.e. chillers, heat pumps, air-conditioners, etc.). The reason is that the manufacturing processes are too complicated and expensive because of the low production volumes.

An exception is twisted tape which is one of the simplest of the various augmentation devices and can be fabricated in any moderately equipped workshop. Two other exceptions were published recently.[9-10] Van den Vyver and Meyer[9] investigated the heat transfer augmentation in the annulus of a heat exchanger consisting of a round tube inside a twisted square tube whilst, in a preliminary study, Swanepoel and Meyer[10] investigated the heat transfer augmentation by means of spiral wires in the annulus of tube-in-tube heat exchangers. It was concluded in both cases that the heat transfer can be substantially increased.

The purpose of this paper is to investigate the potential of another very simple and inexpensive method of heat augmentation which could be used by small manufacturing companies. The heat exchanger is similar to a tube-in-tube heat exchanger but with an additional tube with a relatively small diameter spiralled tightly around the inner tube in the annulus. The aim is therefore to increase the heat transfer in the annulus by swirl flow. The aim is, however, not only to increase the heat transfer in the annulus by swirl flow but also to increase the cross flow area, since some of the flow will not only be through the annulus but also through the spiralled tube. The geometry considered in this study of which different twists are examined is shown in Figure 1. The advantage of this method when compared to spiralled thin wires is that no flow can occur through a wire since it blocks the flow. The disadvantage, however, of using a spiralled tube is that although the flow is not blocked, similar tube diameters of similar diameters of thin wires are not readily available, causing a higher pressure drop than with thin wires.

In this paper, the heat transfer characteristics and pressure drop are determined for the said method of heat transfer enhancement in the annulus. The heat exchange considered will be limited to single-phase (water-to-water) experiments and only parallel flow will be considered. The outline of this paper is as follows: in the next section the experimental set-up is described; this is followed by a description of the method used to determine the convection coefficient and pressure drop in the annulus; thereafter the results of the different experimental case studies are given. These case studies are discussed, whereafter the paper is concluded.

Experimental set-up

The heat exchangers consist of three standard refrigerant soft-drawn copper tube sizes. They consist of a 9.53 mm ($\frac{3}{8}''$) tube inside a 19.05 mm ($\frac{3}{4}''$) tube. The twisted tube inside the annulus is a 6.35 mm tube ($\frac{1}{4}''$) which was twisted tightly by hand at a fixed angle around the inner tube before the outer tube was placed over it. The twisting was conducted by hand and no special tools were used. Care was taken to ensure that the twisting of the tube was uniform and that the inlet and outlet of the twisted tube were fully open. It is estimated that the tube angle was kept constant within $\pm 10°$. It was found that the geometry of the twisted tube changed from round to approximately elliptical and that the minor axis of the ellipse decreased with an increase in twist angle.

The thermal conductivity[10] of the copper tubes is taken as 386 W/m.K. All heat exchanger lengths are 3 m, insulated on the outside from the atmosphere with 50 mm fibreglass wool. The assumption is therefore made that the insulation ensures an adiabatic heat exchanger. Five heat exchanger configurations were constructed: the first without any twisted tube and the other four at twist angles of 18°, 45°, 59° and 67°. These angles correspond at a pitch of 1D, 2D, 3D, and 4D, respectively. A typical example showing the pitch between two spiralled tubes is shown in Figure 1.

For each heat exchanger configuration considered, four K-type thermocouples were fixed onto the inlet and outlet surfaces of the two tubes by insulation tape. The thermocouples were connected to a Fluke microprocessor-based digital thermometer with an accuracy of $\pm 0.1°C$. The thermocouples' readings were used for measuring the inlet and outlet water temperatures on the inside tube and in the annulus. The hot water flowed through the inside tube and the cold water through both the annulus and twisted tube.

The water was heated with a heat pump and stored in a 200 litre insulated storage tank at a temperature of approximately 50° to 55°C. The hot water was connected to the inside tube of the heat exchanger and circulated with a pump from the bottom of the tank through the heat exchanger and back to the top of the storage tank. The flow rate of the hot water was measured by disconnecting the return pipe at the top of the hot-water tank and measuring the time to fill a 5 litre calibrated bucket, without changing the height of the outlet during experiments. During this time the change in the water level and the inlet pressure at the pump were negligible. This measurement was conducted regularly to ensure that the flow rate of the hot water through the annulus remained constant. Tap water at an average inlet temperature of 13° to 15°C flowed through the annulus of the heat exchanger to drain. The flow rate could be adjusted with a globe valve.

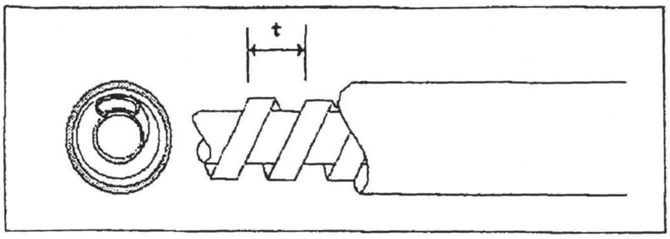

Figure 1 Schematic representation of the tube twisted spirally at a pitch of 1D around the outer surface of the inside tube of a tube-in-tube heat exchanger

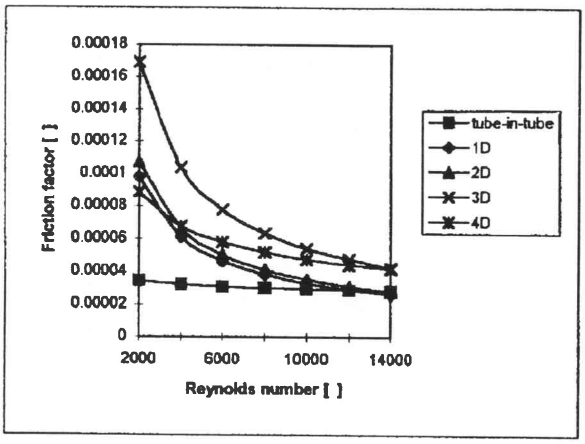

Figure 4 The friction factor in the annulus at different twisted tube pitches as a function of the Reynolds number in the annulus

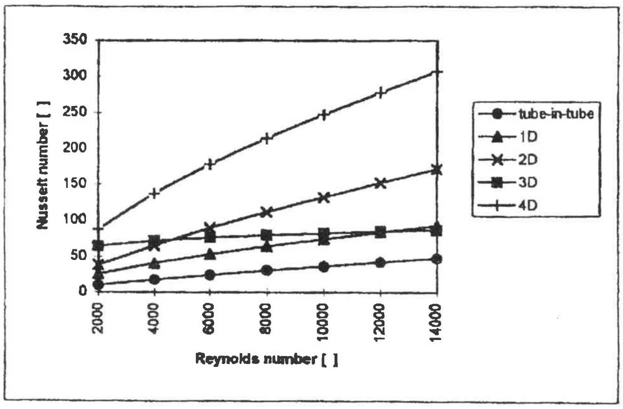

Figure 2 The Nusselt number in the annulus at different twisted tube pitches as a function of the Reynolds number in the annulus

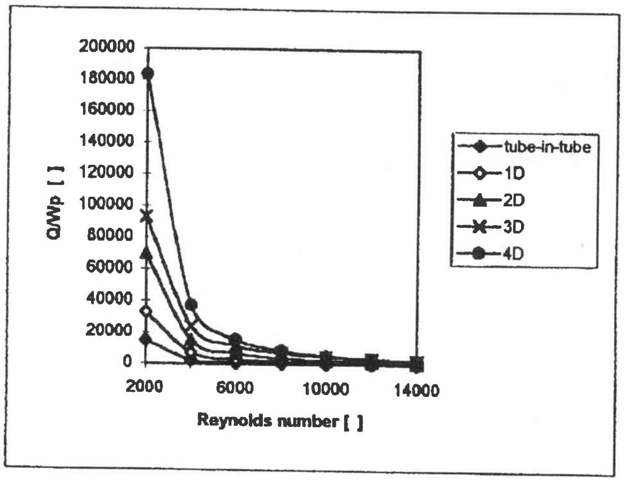

Figure 5 Heat transfer rate per unit pumping power versus Reynolds number in the annulus as a function of different twisted tube pitches

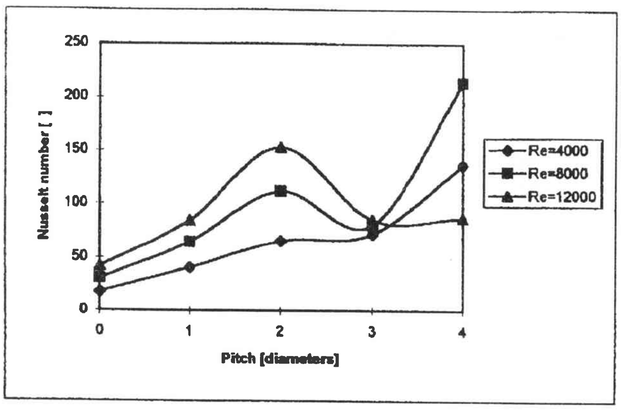

Figure 3 Nusselt number as a function of twisted tube pitch for different Reynolds numbers

The flow rate of the water was determined by taking the time it took to fill a calibrated 10 litre container, without changing the height of the outlet during experiments. The pressure drop through the annulus was measured with a mercury U-tube manometer connected to the inlet and outlet of the annulus.

Convection coefficient in the annulus

For each heat exchanger configuration the mass flow of the hot water through the inner tube was kept constant. Different mass flows were, however, used for different configurations. The flow varied between 200 and 500 kg/h. The mass flow of the cold water through the annulus varied between 30 and 200 kg/h. At the lower mass flow the Reynolds number was laminar although most of the experiments were conducted in the turbulent flow region which is usually of more importance in practice. For each experiment the inlet and outlet temperatures of the water in the inner tube and annulus were measured, as well as the mass flow of the water through the annulus. To ensure steady state conditions sufficient time was allowed to pass before measurements were taken. With each experiment the heat transfer to the annulus was calculated and energy conservation to the heat transfer from the inner tube was checked. When the heat transfer to the annulus and from the inner tube were compared, all errors attributed to measuring inaccuracies were less than 10%.

With the heat transfer known the following procedure was used to determine the convection coefficient in the annulus. With the inlet and outlet temperatures known, the logarithmic mean temperature difference was calculated and then the overall heat transfer coefficient based on the outside area of the inner tube, using

$$U_o = \frac{1}{\left\{\left(\frac{A_o}{A_i}\right)\left(\frac{1}{h_i}\right) + A_o \ln\frac{\left(\frac{D_o}{D_i}\right)}{(2\pi\, k\, L)} + \left(\frac{1}{h_o}\right)\right\}} \qquad (1)$$

With the overall heat transfer coefficient in Equation 1 known, the only two unknowns were the two heat transfer coefficients. If the coefficient in the inner tube could be calculated the heat transfer coefficient in the annulus could be determined.

The inside convection coefficient was calculated using the expression of Petukhov,[11] evaluated at the film temperature. The average bulk temperature of the hot and cold water in the inner tube and annulus were of the order of 40°C and 28°C, respectively. The average wall temperature was then approximately 34°C. Therefore, the temperature difference between the inside wall temperature of the inner tube and the water bulk temperature in the inner tube, was in the region of 6°C. For this temperature difference the maximum changes in water viscosities, thermal conductivity, densities, and specific heats were 12.6%, 0.3%, and 0.1%, respectively. From these values it can be shown from a sensitivity analysis that, should the film temperature be assumed to be equal to the bulk temperature, the maximum influence it will have on the calculation of the convection coefficient is less than 4%. Therefore the approximation, that the film temperature in the Petukhov equation is equal to the bulk temperature, is made in this study.

Although the Dittus Boelter equation[11] would have been much easier to use, the maximum standard deviation between experimental data and predicted values is of the order of 25%[12] to 40%.[13] The accuracy of the Petukhov equation[12] is ±10%, but the inside wall temperature must be known for the calculation of the film properties. Therefore, the need for the above approximation arises from the difficulty of measuring the inside wall temperature. To ensure that this approach gives correct convection coefficients in the annulus, five sets of ten measurements were done at arbitrary Reynolds numbers on a tube-in-tube heat exchanger (without a twisted tube in the annulus) and compared to theoretically predicted heat transfer coefficients. The measured and predicted values according to the method of Petukhov[11] as used in this study are given in Van den Vyver and Meyer.[9] The data correspond well since the average absolute error between the measurements and predictions was ±6.5%; therefore, the apparatus instrumentation and calculation of parameters were ratified.

Results

The Nusselt number and friction factor data points of the measurements in each configuration were used in a commercially available curve-fit program to generate equations as a function of the Reynolds number similar to the Dittus Boelter format:

$$Nu = C_1 Re^{m_1}\,\mathrm{Pr}^{\,n} \text{ with } n = 0.4 \qquad (2)$$

and

$$\Delta p = C_2 Re^{C} \qquad (3)$$

Both the Reynolds number and the Nusselt number in Equations 2 and 3 are based on the hydraulic diameter of the annulus taking the twisted tube into account. The hydraulic diameter was estimated with the well-known relationship

$$D_h = 4\frac{A}{P} \qquad (4)$$

where

$$A = \frac{\pi}{4}\left[(D_{oi}{}^2 - D^2) - ab\right] \qquad (5)$$

The second term in Equation 5 represents the area of the twisted tube in the annulus. This area was found by approximating it to the area of an ellipse; the shape formed after being twisted around the inner tube. The lengths of the major (a) and minor (b) axes were measured for each different configuration. The length of the wetted perimeter (P) in Equation 4 is

$$P = \pi D_{oi} + P_s + P_i \qquad (6)$$

The wetted perimeter of the twisted tube (P_s) and inner tube (P_i) was measured by opening the tube-in-tube heat exchanger. The reason it was measured and not calculated was that it was found that the wetted perimeter of the twisted tube is less than the perimeter of an ellipse as the contact between the twisted tube and inner tube is not a small contact point. The hydraulic diameters for the spiralled tubes corresponding to a pitch of 1D, 2D, 3D, and 4D were 1.9 mm, 2.8 mm, 4.1 mm, 4.0 mm, and 3.8 mm, respectively.

The reason Equation 2 was not used in the same format as the Petukhov formula is that the format is easier to work with. The constants for Equations 2 and 3 are given in Table 1. It was found in all cases that the maximum standard deviations between the Nusselt number equations and measurements are less than 11% and less than 7% for the pressure drop values. The results of the equations are presented in Figures 2 to 4 at a Prandtl number of 7.3.

Table 1 Constants used in Equations 2 and 3

Pitch diameters	C	C_1	C_2	m_1
1	1.329	1.713E-01	2.601E-03	6.598E-01
2	1.315	1.098E-01	1.044E-03	7.771E-01
3	1.296	2.002E+01	5.763E-04	1.541E-01
4	1.618	6.463E-01	2.761E-05	6.456E-01

It can be seen from Figure 2 that the Nusselt number increases with Reynolds number as can be expected and the Nusselt number increases in general as the pitch of the twisted tube is increased, except at a pitch of 3D. This can be seen more clearly if the Nusselt number is plotted against the pitch of the twisted tube (Figure 3) at Reynolds numbers of 4 000, 8 000 and 12 000. This phenomenon could not be explained in this study (even after repeating the experiments) and it is recommended that it should be investigated further with a numerical method or with more experiments. The reason it was not investigated in more detail is that it would have no influence on the general conclusions of this study. It can be concluded that the highest heat transfer is obtained with a pitch of 4D for the twisted tube. If the Nusselt number at a specific Reynolds number of this geometry is compared to other geometries investigated[9,10,14] it can be concluded that it outperforms the other geometries.

Figure 4 shows the friction factor. The tendency of the results is as expected except at low Reynolds numbers of approximately 2 000 (again at a pitch of 3D). The friction factor is the lowest for a tube-in-tube heat exchanger without a twisted tube and the highest with a pitch of 4D. At high Reynolds numbers (approximately 14 000 and more) the friction factor seems to convert to a value of approximately 0.00004 where it is no longer a function of Reynolds number or twisted tube pitch.

In comparing the performance of the heat exchangers with different twist angles to the heat exchanger without any twisted tube, it is necessary to take into account both the heat transfer increase and the corresponding pressure drop increase. Figure 5 shows the performance data of the four heat exchangers with twisted tubes at different angles as well as the tube-in-tube heat exchanger without any twisted tube in the ratio of the pumping power to the rate of heat transfer as an evaluation index.[14] The rate of heat transfer was evaluated from the temperature rise through the annulus and the pumping power from the product of the volumetric flow rate through the annulus and the pressure drop across the test section.

The results show that all twist angles increase the heat transfer rate per unit pumping power although it decreases with Reynolds number. It can be deduced that a twist angle equal to a pitch of 4D should be used when a round tube is used in annuli of tube-in-tube heat exchangers. It is recommended that more experiments should be conducted at higher pitches (5D and higher) to obtain the pitch where the optimum heat transfer enhancement takes place. This was not done in this study owing to time constraints and owing to the fact that the higher the pitch the more difficult it gets to manufacture the heat exchangers.

Conclusions

In this study the influence of twist angle of a heat exchanger consisting of a twisted tube inside the annulus of a tube-in-tube heat exchanger was investigated. Five geometries were investigated, one without any twist and the rest with twist angles corresponding to pitches of 1D, 2D, 3D, and 4D, respectively. It was found in general that the heat transfer increases as the pitch and Reynolds number increase, except at a pitch of 3D, where results that cannot be explained were obtained. The heat transfer rate per unit pumping power was used as an evaluation index. It was concluded that all twist angles increase the heat transfer rate per unit pumping power although the rate decreases as the Reynolds number increases. The highest heat transfer per unit pumping power was obtained at a pitch of 4D. It is recommended that more work should be conducted, firstly, to explain the phenomena at a pitch of 3D and, secondly, to obtain the optimum pitch which should be at a pitch higher than 4D.

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