Heat transfer augmentation in the annulus of a heat exchanger consisting of a round tube inside a twisted square tube

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(Received February 1997; Final version June 1997)

The potential of a very simple and inexpensive method of heat augmentation that could be used by small manufacturing companies is investigated. It consists of an aluminium heat exchanger comprising a round tube inside a twisted square tube. 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 on five different heat exchanger configurations: the first without any twist and the rest with the square tube twisted at angles of 45°, 60°, 90°, and 105° per metre, respectively. The Nusselt number and friction factor in the annulus of each case study were measured as functions of the Reynolds number, while the Prandtl numbers were kept constant; and for each heat exchanger configuration an equation was deduced that describes this functional relationship. The relationships are shown in graphical format. It was found in general that the heat transfer and pressure drop increase with twist angle and Reynolds number. Furthermore that for a twist angle of 60° and a Reynolds number lower than 4 000, the heat transfer rate per unit pumping power is the highest while for higher Reynolds numbers a twist angle of 90° gives the highest heat transfer per unit pumping power.

Nomenclature

A_i	inside area of inner tube [m^2]
A_o	outside area of inner tube [m^2]
C_p	specific heat of water at constant pressure [J/kg.K]
C_1, C_2	constants used in Equations 4 and 5
D	hydraulic diameter of the annulus before twisting [m]
f	Darcy-Weisbach friction factor ($2D\Delta p/\rho.V^2.L$)
h_i	convection coefficient in inner tube [W/m^2.K]
h_o	convection coefficient in annulus [W/m^2.K]
k	thermal conductivity of inner tube [W/m.K]
k_f	thermal conductivity of water in annulus at annulus bulk temperature [W/m.K]
L	length of inner and outer tubes [m]
m	mass flow of water in annulus [kg/s]
m_1	Nusselt number in annulus (h_oD/k_f)
Pr	Prandtl number in annulus at bulk temperature ($C_p\mu/k_f$)
Δp	pressure drop of water in annulus [Pa]
Q	heat transfer to annulus [W]
r_i	inside radius of inner tube [m]
r_o	outside radius of inner tube [m]
t	inside width of square tube [m]
Re	Reynolds number for annulus ($\rho VD/\mu$)
T_i	annulus inlet temperature [°C]
T_o	annulus outlet temperature[°C]
ΔT	logarithmic mean temperature difference [°C]
U_o	overall heat transfer coefficient based on A_o for 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

The process of improving the performance of a heat transfer system is referred to as heat transfer augmentation, enhancement or intensification. 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 outputs. The only exception is twisted tape which is one of the simplest of the various augmentation devices and can be fabricated in any moderately equipped workshop.

The purpose of this paper is to investigate the potential of one of many very simple and inexpensive methods of heat augmentation that could be used by small manufacturing companies. The heat exchanger is similar to a tube-in-tube heat exchanger with the inside tube round as usual but with the outside tube square and twisted. The aim is therefore to increase the heat transfer in the annulus by swirl flow. The intensity of the swirl is influenced by the twist of the outside square tube. The geometry considered in this study of which different twists are examined is shown in Figure 1.

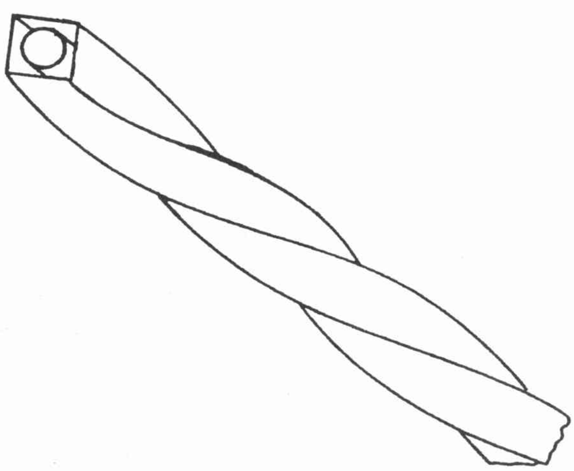

Figure 1 Schematic representation of the round tube inside a twisted square tube heat exchanger (twisting not to scale)

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 following section the experimental set-up is described; followed by a description of the method used to determine the convection coefficient and pressure drop in the annulus; whereafter the results of the different experimental case studies are given. These case studies are discussed, and the paper is thereafter concluded.

Experimental set-up

The heat exchangers consist of extruded, round tubes, inside extruded, twisted square tubes. Both tubes are made of aluminium and the thermal conductivity of the aluminium is taken as 204 W/m.K.[9] The inside and outside diameters of the round tube are 9.5 mm and 12.6 mm, respectively. The outer dimensions of the square tubing before twisting are 19 mm by 19 mm and the wall thickness is 1.6 mm. All heat exchanger lengths are 3 m, insulated on the outside from the atmosphere with 50 mm of fibreglass-wool. The assumption is therefore made that the insulation ensures an adiabatic heat exchanger.

The external square tube was twisted by hand with a shifting spanner while the tube was clamped in a vice. Twisting was at small angles at regular intervals of 50 mm along the axis of the tube. Four wires of 1 mm in diameter each, which were not removed afterwards, were placed longitudinally in the annulus to keep the inside tube in position. Care was taken to ensure that the twisting of the tube was uniform. Five heat exchanger configurations were constructed. The first without any twist and the other four at twist angles of 45°, 60°, 90°, and 105° per metre, respectively. The angle of 105° was the maximum that could be achieved before the square tube started to crack and the square structure collapsed.

For each heat exchanger configuration considered four K-type thermocouples were fixed firmly 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 ±0.1°C. The thermocouples' readings were used for measuring the inlet and outlet water temperatures in the inside tube and in the annulus. The hot water flowed through the inside tube and cold water through the annulus.

Hot 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 was 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. 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 external tube.

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 in practice is usually of more importance. 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 enough time was allowed to pass before measurements were taken. With every experiment the heat transfer to the annulus was calculated by

$$Q = mC_p\,(T_i - T_o) \tag{1}$$

and compared for energy conservation to the heat transfer from the inner tube, which was calculated in a similar manner as Equation 1. When the heat transfer to the annulus and from the inner tube were compared, all errors attributed to measuring inaccuracies were less than 7%.

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, by

$$Q = U_o A_o \Delta T \tag{2}$$

where the overall heat transfer coefficient was

$$U_o = 1/\{(A_o/A_i)\,(1/h_i) + A_o \ln{(r_o/r_i)}\,/\,(2\pi kL) + (1/h_o)\} \tag{3}$$

With the overall heat transfer coefficient known in Equation 3, 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 thus be determined.

The inside convection coefficient was calculated from the expression of Petukhov,[9] 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 should therefore have been 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 conductivities, densities and specific heats are 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%. The approximation is therefore made in this study that the film temperature in the Petukhov equation is equal to the bulk temperature.

Although the Dittus Boelter equation[9] would have been easier to use, the maximum standard deviation between experimental data and predicted values is of the order of 25%[10] to 40%.[11] The accuracy of the Petukhov equation is 10%[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 (the twisted square tube was therefore replaced by a round tube) and compared to theoretically predicted heat transfer coefficients. The measured and predicted values according to the method of Petukhov[9] as used in this study are given in Figure 2. 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 twisted square annulus were used in a commercially available curve-fit programme to generate equations as a function of the Reynolds number similar to the Dittus Boelter format:

$$Nu = C_1 Re^{m1}\,\mathrm{Pr}^{\,n} \text{ with } n = 0.4 \tag{4}$$

and

$$\Delta p = C_2 Re^{1.483} \tag{5}$$

Both the Reynolds numbers in Equations 4 and 5 are based on the hydraulic diameter of the untwisted configuration and can be calculated as

$$D = (t^2 - \pi {r_o}^2)\,/\,(t + 0.5\pi r_o) \tag{6}$$

Table 1 Constants used in Equations 3 and 4

Twist [deg./m]	C_1	m_1	C_2
0	0.02772	0.7549	4.741E – 02
45	0.12403	0.6048	4.978E – 02
60	0.13154	0.6106	5.215E – 02
90	0.02389	0.8277	5.689E – 02
105	0.03034	0.8060	6.163E – 02

The reason Equation 4 was not deduced in the same format as the Petukhov formula is that the format is easier to work with. The constants for Equations 4 and 5 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 3 and 4 at a Prandtl number of 7.3.

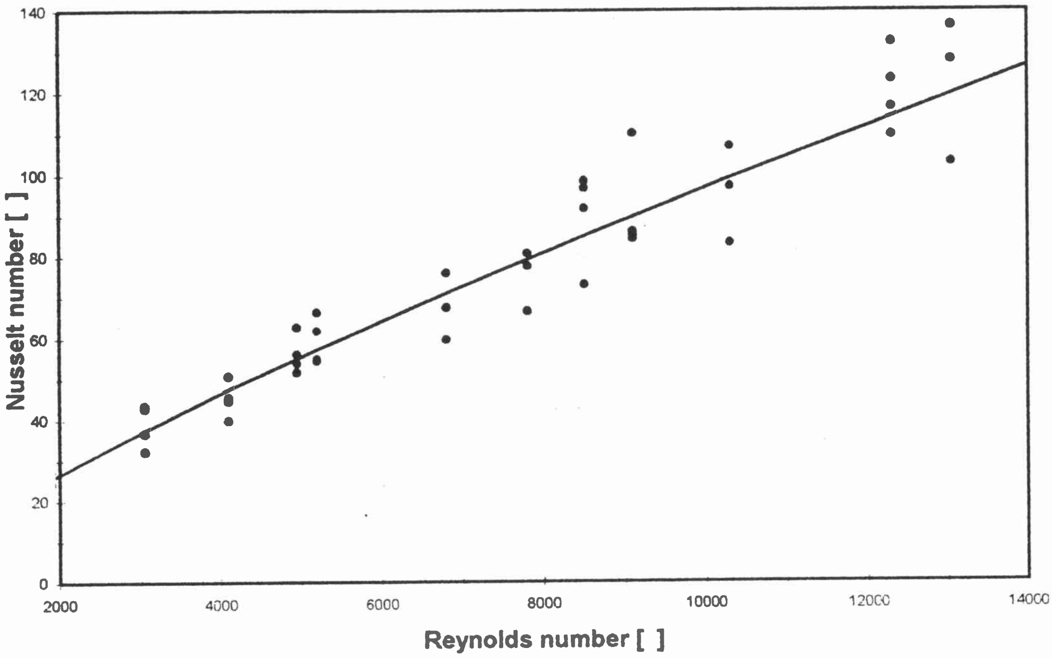

Figure 2 A comparison between measurements (•) in the annulus and the prediction method (solid line) of Petukhov[9] for a tube-in-tube geometry

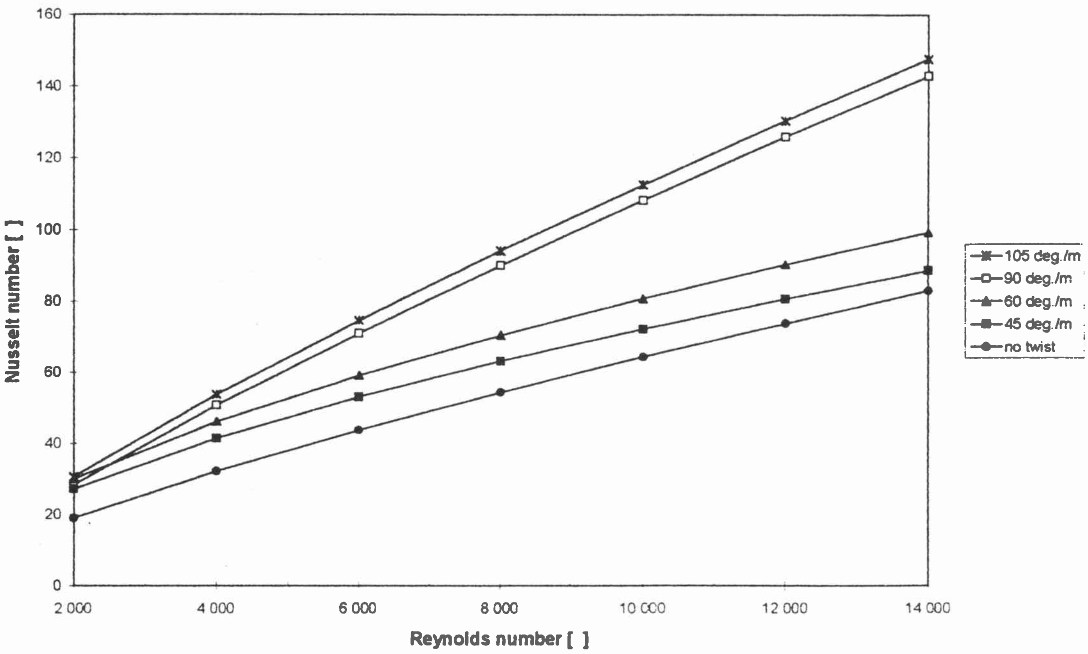

Figure 3 The Nusselt number in the annulus at different twist angles as a function of the Reynolds number in the annulus

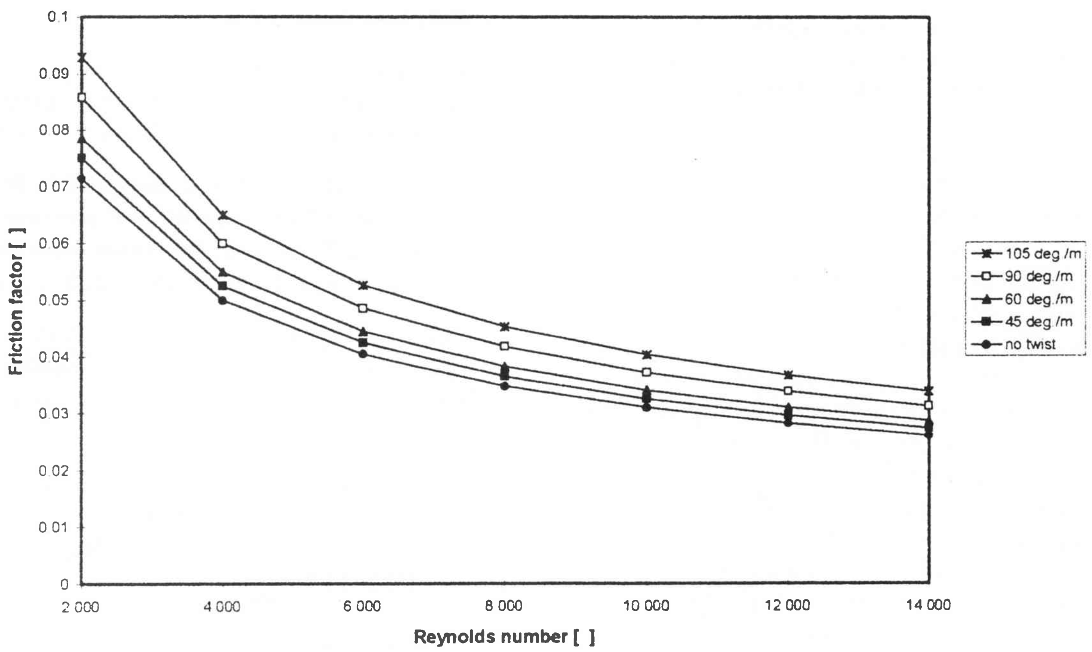

Figure 4 The friction factor in the annulus at different twist angles as a function of the Reynolds number in the annulus

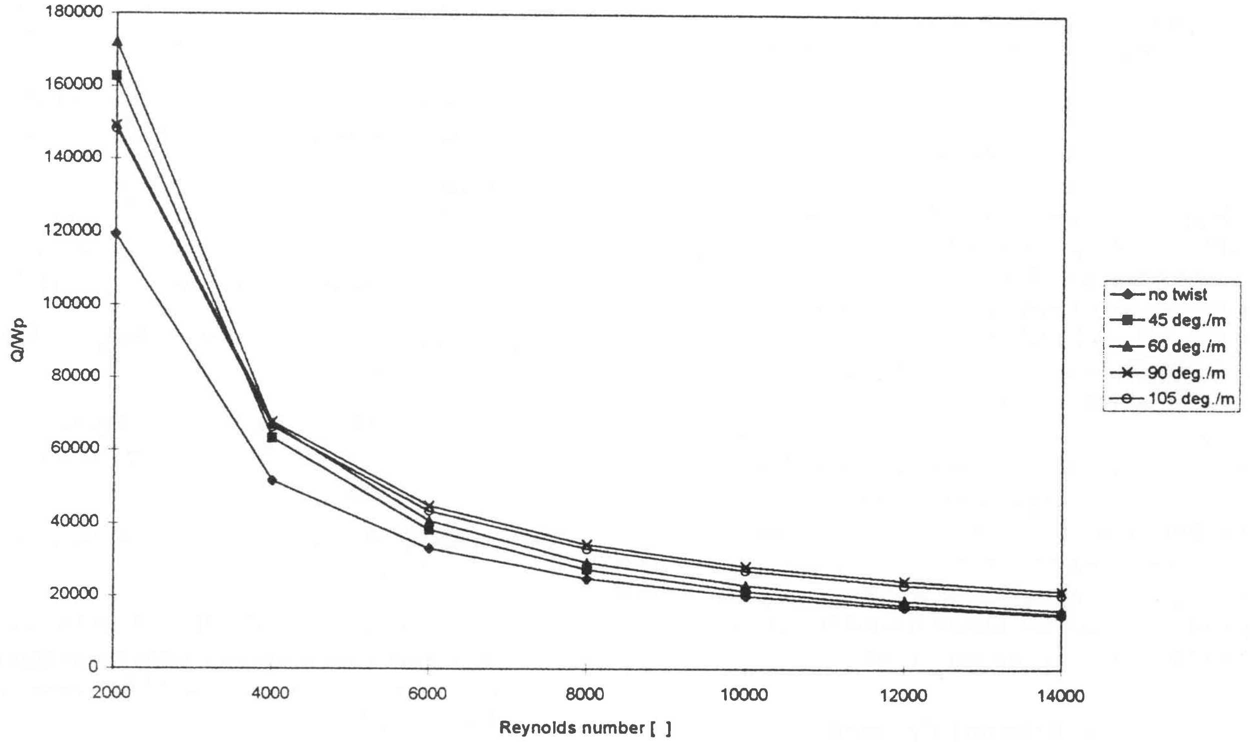

Figure 5 Heat transfer rate per unit pumping power versus Reynolds number in the annulus as a function of different twist angles

In Figures 3 and 4 it can be seen that the Nusselt number and pressure drop increase with twist angle as could be expected. The only exception is for the Nusselt number at a Reynolds number lower than 2 800 where a twist angle of 60° generates higher heat transfer coefficients than at 90°. This phenomenon could possibly be attributed to the transition from laminar to turbulent flow which appears to happen faster at a twist angle of 60° than at 90°. Note also that the gradients for the Nusselt number as a function of the Reynolds number for twist angles of 90° and 105° are steeper than for the other twist angles, especially at high Reynolds numbers.

In comparing the performance of the heat exchangers with different twist angles to the heat exchanger with no twist, 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 twisted heat exchangers as well as the no-twist heat exchanger in the form of the ratio of the pumping power to the rate of heat transfer as an evaluation index.[12] 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. If the Reynolds number is lower than 4 000 the heat transfer rate per unit pumping power is the highest for a twist angle of 60° while for higher Reynolds numbers a twist angle of 90° gives the highest values.

Conclusions

In this study the influence of twist angle of a heat exchanger consisting of a round tube inside a twisted square tube was investigated. Five geometries were investigated, one without any twist and the rest with twist angles of 45°, 60°, 90°, and 105°, respectively. It was found in general that the heat transfer and pressure drop increase with twist angle and Reynolds number. The heat transfer rate per unit pumping power was used as an evaluation index. It was concluded that all twist angles increased the heat transfer rate per unit pumping power although it decreased as the Reynolds number increased. It was also found that at Reynolds numbers lower than 4 000 the heat transfer rate per unit pumping power was the highest for a twist angle of 60° while for higher Reynolds numbers a twist angle of 90° gave the highest values.

Acknowledgement

This project was a final-year project in the Department of Mechanical and Manufacturing Engineering at the Rand Afrikaans University.

References

1. Bergles AE. *Techniques to augment heat transfer.* Handbook of heat transfer applications. McGraw-Hill, New York, 1985.
2. ASHRAE 1993. *ASHRAE Handbook: Fundamentals.*, American Society of Heating, Refrigeration and Air-conditioning Engineers, Atlanta, GA, USA, 1993.
3. Schlager LM, Pate MB & Bergles AE. Evaporation in condensation heat transfer and pressure drop in horizontal, 12.7 mm microfin tubes with refrigerant 22. *Journal of heat transfer*, 1990, **112**, pp.1041–1047.
4. Zimparov VD & Vulchanov NL. Performance evaluation criteria for enhanced heat transfer surfaces. *International journal of heat and mass transfer*, 1994, **37**(2), pp.1807–1816.
5. Ravigurujan TS & Bergles. Prandtl number influence on heat transfer enhancement in turbulent flow of water at low temperatures. *Journal of heat transfer*, 1995, **117**, pp. 276-281.
6. Bergles AE. Some perspectives on enhanced heat transfer – second generation heat transfer technology. *Journal of heat transfer*, 1988, **110**, pp.1082–1096.
7. Li HM, Ye KS, Tan YK & Deng SJ. Investigation on tubeside flow visualization, friction factors and heat transfer characteristics of helical- ridging tubes. *Heat transfer 1982.* Proceedings of the 7th International Heat transfer conference, Hemisphere Publishing Corporation, Washington, DC, 1982, **3**, pp.75–80.
8. Agarwall SK & Raja Rao M. Heat transfer augmentation for the flow of a viscous liquid in circular tubes using twisted tape inserts. *International journal of heat and mass transfer*, 1996, **39**(17), pp. 3547-3557.
9. Holman JP. *Heat transfer*, 7th edn. McGraw-Hill, London, 1992.
10. Incropera FP and De Witt DP. *Fundamentals of heat and mass transfer*, 3rd edn. John Wiley & Sons, Singapore, 1990.
11. Bejan A. *Heat transfer.* John Wiley & Sons, Singapore, 1993.
12. Said SA & Azer NZ. Heat transfer and pressure drop during condensation inside horizontal tubes with twisted-tape inserts. *ASHRAE Transactions*, 1983, **89**(1a), pp.96–113.