Research Note: Preliminary investigation of heat transfer augmentation by means of spiral wires in the annulus of tube-in-tube heat exchangers

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Introduction

Techniques[1] applied to improve the efficiency of heat exchangers or to augment heat transfer can be classified either 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 flow 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 or the type of heat exchanger to which it is applied.

Several examples of tube-in-tube heat exchangers with augmentation techniques are described in the subject literature.[1-4] 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 by 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 strips can be inserted as original equipment or as retrofit devices.

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.

It is the purpose in this paper to investigate the potential of one of many very simple and inexpensive methods of heat transfer augmentation that can be used by small manufacturing companies. The method of heat transfer augmentation is shown in Figure 1. The aim is to increase the heat transfer between the flow in the inner tube and the flow in the annulus.

In this paper the heat transfer characteristics, or Nusselt numbers, are determined experimentally for the said method of heat transfer enhancement. Although the lower limit of the experiments is in the laminar region, most of the results are for turbulent flow. It is, however, a preliminary study – a first quick look with no pretensions to comprehensiveness. Therefore experimental results for only eight enhancement configurations are given, as well as one pressure drop result. The heat exchange will be limited to water-to-water applications and only parallel counterflow will be considered.

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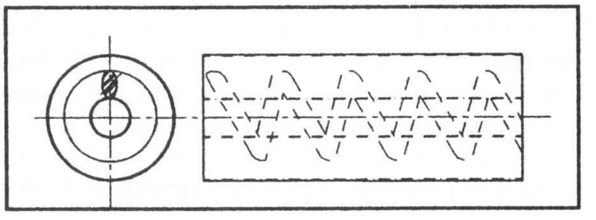

Figure 1 Schematic representation of wires turned spirally around the outer surface of the inside tube of a tube-in-tube heat exchanger

Experimental set-up

The tube-in-tube heat exchangers consist of two soft-drawn refrigeration tubes made of copper (thermal conductivity, k = 386 W/mK). The inside and outside diameters of the inside tube are 4.93 mm and 6.35 mm, respectively, and 11.18 mm and 12.70 mm for the outside tube. (In Imperial dimensions this would be a quarter-inch tube inside a half-inch tube.)

In total nine heat exchangers were constructed. One without any wires in the annulus and eight with wires turned tightly by hand, spirally around the inner tube. The wires were not fastened permanently to the tubes; they stayed in position through friction. The eight heat exchangers were made up by using different wire thicknesses (0.5 and 1 mm), twist angles[3] (30° and 60°) and number of wires (one or two wires). When two wires were used the second wire was turned in the middle of the pitch of the first wire. All heat exchanger lengths were 3.014 m, insulated from the atmosphere with 50 mm of fibreglass. For each heat exchanger configuration, four K-type thermocouples were silver-soldered onto the inlet and outlet surfaces of the two tubes. The thermocouples were connected to a Fluke microprocessor-based digital thermometer with an accuracy of ±0.1% of the reading.

[3]The angle is the smallest angle between the wire centre line and the longitudinal axis of the inner tube.

Water was heated by means of a heat pump and stored in a 200ℓinsulated tank at a temperature of approximately 53°C. The hot water was connected to the inside tube of the heat exchanger and circulated by a pump through the heat exchanger, a variable area flowmeter, and back into the hot water tank. The inaccuracy of the flowmeter is $\pm 3\%$ of the reading. Tap water at an average inlet temperature of 21°to 22°C flowed through the annulus of the heat exchanger in an opposite direction than the hot water to a drain. The flow rate could be adjusted by means of a globe valve. The flow rate of the water was determined by taking the time to fill a calibrated 10ℓcontainer, without changing the height of the outlet during experiments. The pressure drop through the annulus was measured using a mercury U-tube manometer connected to the inlet and outlet of the annulus. Pressure drops were measured only on two heat exchangers.

Convection coefficient in the annulus

For each heat exchanger the mass flow of hot water through the inner tube was kept constant at 67 kg/h while the mass flow through the annulus varied between 160 and 900 kg/h. With the lower mass flow the Reynolds number was laminar, but close to the critical Reynolds number. Most of the experiments were, however, conducted in the turbulent flow region.

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. Regular checks were carried out to ensure that the mass flow of the hot water through the inner tube was constant. With every experiment the heat transfer to the annulus was calculated and compared for energy conservation to the heat transfer from the inner tube. When compared, the errors attributed to measuring inaccuracies were less than 12%.

The following procedure was used to determine the convection coefficient in the annulus. With the inlet and outlet temperatures known, the log mean temperature difference was calculated. With the heat transfer known from the mass flow and temperature measurements the overall heat transfer coefficient was calculated. By calculating the heat transfer coefficient in the inner tube the heat transfer coefficient in the annulus could be determined.

The inside convection coefficient was calculated from the expression of Petukhov,[6] evaluated at the film temperature. The average bulk temperatures 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, between the inside wall temperature of the inner tube and the water bulk temperature in the inner tube, the temperature difference 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%, 2.5%, 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. The need for this approximation arises from the difficulty of measuring the inside wall temperature of a tube-in-tube heat exchanger.

Verification of experimental procedure

The inside convection coefficients were calculated as a function of ten arbitrarily chosen Reynolds numbers. This experiment was repeated five times on different occasions without any wires in the annulus. Thereafter the Nusselt numbers in the annulus were calculated for the different Reynolds numbers using the procedure described in the previous sections. The fifty data points of the five experiments were used in a commercially available curve-fit program to generate an equation of the Nusselt (Nu) number as a function of the Reynolds (Re) and Prandtl (Pr) numbers. The equation based on the hydraulic diameter of the annulus with a standard deviation of 6.5% is given as

$$Nu = 0.0275 Re^{0.8} Pr^{0.4} \qquad (1)$$

The reason equation (1) was not written in the same format as the Petukhov formula is that the format of equation (1) is easier to work with. Since there are no wires in the annulus these 'measured' convection coefficients or Nusselt numbers in the annulus, given by equation (1), can be compared to theoretical values predicted by the Dittus Boelter[6] formula. By comparing these theoretical values with the measurements presented as equation (1), it is found that the error is less than 20%, which is acceptable since the Dittus Boelter equation gives answers only to within ±25% of measurements.

Results

The same procedure followed in the previous section can therefore be followed to determine similar equations for the Nusselt numbers in the annulus with wires around the inside tube. The maximum standard deviation of the measurements with the derived equations was 35% and the average standard deviation was 23%. The results are presented in Figures 2 and 3 at a Prandtl number of 5.49. In Figure 2 the results are given for the wire(s) at 30° while the results for 60° are given in Figure 3. In both figures the case of no wires in the annulus is also included for comparison purposes.

If Figure 2 is considered it can be concluded that the Nusselt number or convection coefficient in the annulus increases with an increase in the number of wires but does not necessarily always increase with wire thickness from 0.5 to 1.0 mm. The convection coefficient increases by a factor of 2.3 from the case of no wires to the case of two wires at 30 with a thickness of 0.5 mm. The same tendency is shown in Figure 3 where the Nusselt number in the annulus increases by 2.8 from the case of no wires to the case of two wires at 60 with a thickness of 0.5 mm. It therefore seems that higher convection coefficients occur for two wires of 0.5 mm at 60°, than at 30°.

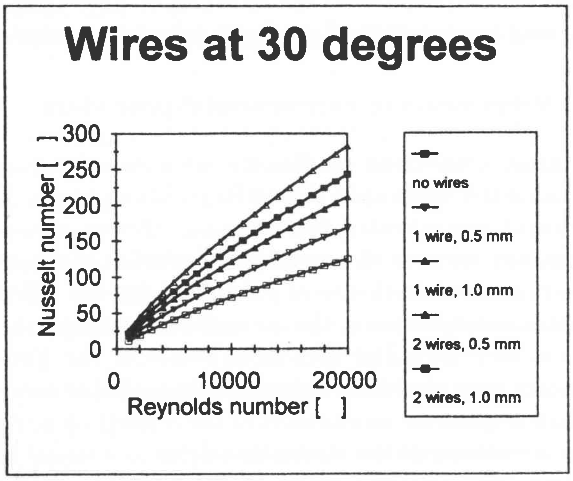

Figure 2 The Nusselt number in the annulus as a function of the Reynolds number with the wires at 30°

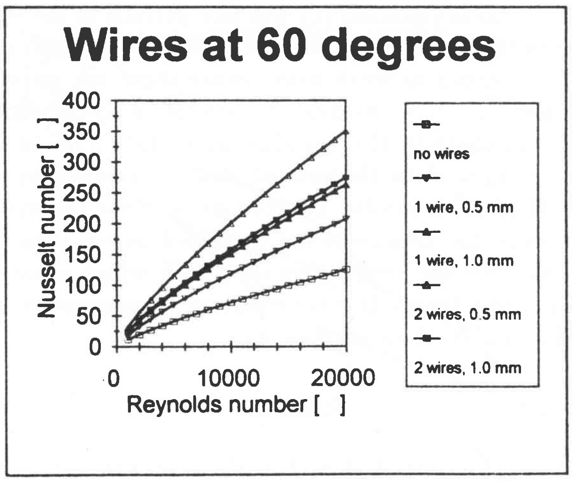

Figure 3 The Nusselt number in the annulus as a function of the Reynolds number with the wires at 60°

The results in Figures 2 and 3 show a potential in the use of wires in the annulus of tube-in-tube heat exchangers. However, more results are needed to make definite recommendations on the optimum number of wires, angle and thickness. In this regard more experiments and case studies will have to be conducted and pressure drops should also be measured. In this regard two experiments were conducted. The pressure drops in the annulus were measured as a function of the Reynolds number for the case without wires and the case with two 1.0 mm thick wires at 60°. For this case the increase in heat transfer is a factor of 2.2 while the pressure drop increases by a factor of 1.8.

Conclusion

In this study the influence of wires in the annulus of a tube-in-tube heat exchanger on the heat transfer was investigated. Although results were found to show a substantial increase in heat transfer, not enough results are given to select an optimum configuration on the number, angle and thickness of the wires. It is therefore recommended that more case studies are considered and that for each case study the pressure drop is also measured. The value of this work lies in the contribution made to show the potential of this technique and that more research should be conducted on this subject.

Acknowledgement

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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**, pp.1807–1816.

5. Mills AF. *Heat transfer.* Irwin, Homewood, IL, USA, 1992.

6. Holman JP. *Heat transfer.* 7th edn. McGraw-Hill, London, 1992.