Angled Spiralling Tape Inserts in a Heat Exchanger Annulus

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The purpose of this paper was to determine the single phase heat transfer and pressure drop chara-cteristics of an angled spi-ralling tape inserted into the annulus of a tube-in-tube heat exchanger. Experimental meas-ure-ments were taken on four set-ups: a normal tube-in-tube heat exchanger used as a refer-ence and three heat ex-changers with different angled spiralling tape in-serts. From the results correlations were developed that can be used to predict and com-pare heat transfer and pressure drop characteristics. It was concluded that the angled spiralling tape inserts resulted in an increase in the heat transfer and pressure drop characteristics.

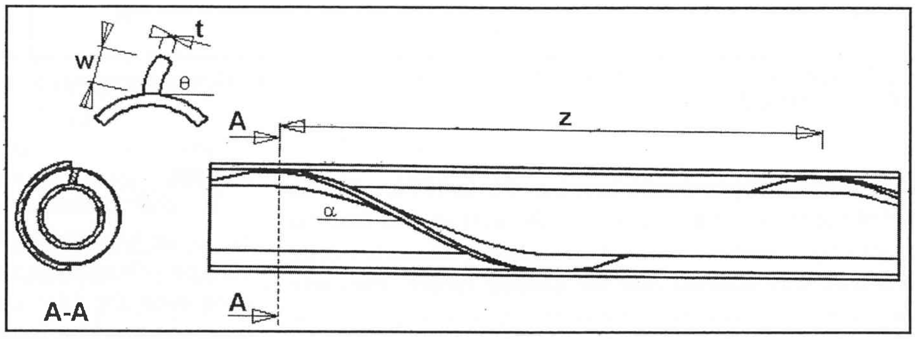

Figure 1: Schematic Representation of an Angled Spiralling Tape Heat Exchanger

NOMENCLATURE

A	area (m^2)
C	constant
CP	pressure drop coefficient
Cp	specific heat (J/kgK)
D	diameter for outer tube (m)
D_h	hydraulic diameter (m)
d	diameter for inner tube (m)
d_{eq}	equivalent diameter $[(D_i^2 - d_o^2)/d_o]$ (m)
EF	enhancement factor $[Nu_{Experimental} / Nu_{Plain\ Tube\text{-in-Tube}}]$
f	friction factor
$f()$	function of
$g()$	function of
h	heat transfer coefficient (W/m^{2}K)
k	thermal conductivity (W/mK)
L	length of test section (m)
l	length as shown in Fig. 2 (m)
$\dot{m}_w$	mass flow rate (kg/s)
Nu	Nusselt number
P	constant
PF	pressure drop penalty factor $[CP_{Experimental} / CP_{Plain\ Tube\text{-}in\text{-}Tube}]$
Pr	Prandtl number
Δp	pressure drop/difference (Pa)
Q	heat transfer rate (W)
Re_i	Reynolds number in the inner-tube based on the inner-diameter
Re_o	Reynolds number in the annulus based on the hydraulic diameter (D_h)
r	tube radius (m)
T	temperature (°C)
ΔT	temperature difference (°C)
t	tape thickness (m)
U	overall heat transfer coefficient (W/m^{2}K)
v	velocity (m/s)
w	tape height (m)
y	twist ratio $[Z/2D_i]$
Z	pitch (mm)

Greek Letters

α	tape angle (^{0})
θ	tape leaning angle (^{0})
ρ	density (kg/m^3)
μ	viscosity (Ns/m^2)

Subscripts

ave	average
c	cold water
cu	copper
h	hot water
i	inside/inner
$LMTD$	Logarithmic Mean Temperature Difference
o	outside/outer
w	wall

Introduction

Heat transfer enhancement is the process of improving the performance of a heat transfer system, generally by means of increasing the heat transfer coefficient. Bergles[1] gives a comprehensive survey of heat transfer enhancement. According to him heat transfer enhancement has been studied since the first documented study of heat transfer. For example, Newton suggested in 1701 "... not in a calm air, but in a wind that blew uniformly upon it...", which was an effective way of increasing the convective heat transfer. Bergles1 states that Joule was the

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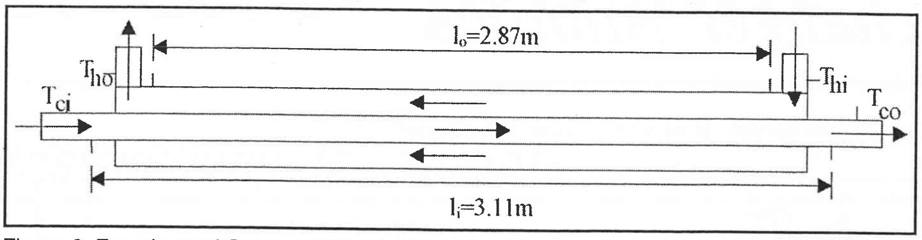

Figure 2: Experimental Set-up of Tube-in-Tube Heat Exchanger with Locations of Temperature and Pressure Measuring Points

first to use second-generation heat transfer in 1891, when he reported significant improvement in the overall heat transfer coefficient for in-tube condensation of steam, when a helically coiled wire was inserted into the cooling jacket. The only second-generation enhancement device that was used in the 1800s with any regularity was the twisted tape insert. In 1896 Whitham, who placed twisted tape inserts into the tubes of a fire-tube boiler, reported an increase of up to 18% in the boiler efficiency.

In many cases heat transfer enhancement in tubes may be supplemented by heat transfer augmentation on the outside wall of tubes, as for tube-in-tube heat exchangers. An application is in vapour compression hot-water heat pumps where an axially coiled tube-in-tube heat exchanger[2] is used for heating water in the condenser of a hot-water heat pump. The condensing refrigerant may typically flow in the inner tube and the water to be heated in a counterflow direction in the annulus. Heat transfer enhancement on the inner or outer tube decreases the temperature difference between the condensing refrigerant and water to be heated, which is an advantage as higher hot temperatures may be delivered.

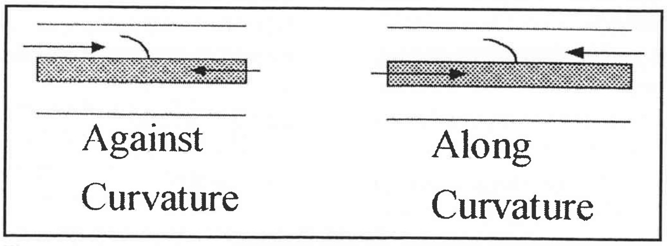

Figure 3: Flow in the Annulus Against and Along the Curvature of the Tape

In a review paper by Liebenberg et al.[3] it is shown that a lot of work has been done in heat transfer enhancement on the inner wall of tubes. However, in the case of water heating with refrigerants flowing in an inner tube and water through the annulus, heat transfer enhancement on the outside wall is important. The heat transfer from the refrigerant to the water through the tube is influenced by three components, namely: the thermal convection resistance of the condensing refrigerant on the inside of the inner tube ($1/h_iA_i$), the conduction resistance of the tube wall ($ln(r_o/r_i)/(2\pi kL)$) and the convection resistance ($1/h_oA_o$) of the water in the annulus.

The tube wall is thin and has a high thermal conductivity (k) as it is generally manufactured from copper or stainless steel. The result is that the wall's thermal resistance is negligibly small in comparison with the two convection resistance terms. The heat transfer coefficient of condensing refrigerant (h_i) is usually relatively large in comparison with the convection coefficient (h_o) of the water in the annulus. Therefore, the thermal resistance is the highest on the annulus side of the wall and that is the reason why heat transfer enhancement on the outside wall of the inner tube can make an important contribution to deliver higher hot-water temperatures.

Previous work to improve the heat transfer coefficient in annuli was done by Van der Vyver and Meyer[4]. They used a round tube inside a twisted square tube. The function of the twisted square tube was to induce a rotational component in the annulus, which increased the heat transfer coefficient by up to 50%. At the same time the friction factor increased by up to 9 percent.

Another method used by Herman and Meyer[5] was to use a spiralling tube inside the annulus of a tube-in-tube heat exchanger. Three tubes were thus used: Except for the inner and outer tube, the third tube was spiralled in the annulus of the other two and also formed a flow passage. The mechanism of heat transfer enhancement was thus also to cause swirl movement of the flow in the annulus. The enhancement increased the heat transfer coefficient in the annulus by an average of 600 per cent and the friction factor by 84 per cent.

A third method was investigated by Coetzee et al.[6] and Dirker et al.[7], which is simpler, is the use of thin wires in the annulus that are spiralled around the inner tube. Several studies [8,9] were done to determine the spiral angle and the optimum wire thickness. Recently Krüger[10] investigated the mechanism of the heat transfer enhancement by making use of computational fluid dynamics and found that there is a strong correlation between the wire angle and the rotational velocity in the annulus. She concluded that the heat transfer enhancement is primarily caused by the flow rotation and not by an increase in turbulence of the flow over the wires.

Most of the methods of enhanced heat transfer use the same mechanism to enhance heat transfer, and that is to force the flow in the annulus to rotate. Bergles[1] has found that devices that induce swirl flow and turbulence in the flowing fluid are particularly attractive enhancement techniques for forced convective systems. Another method that makes use of this principle was recently patented by Meyer and Coetzee[11]. An angled spiralling tape as shown in Fig. 1 is used in the annulus to induce swirl. This method of heat transfer enhancement was specifically developed for hot-water heating in heat pumps, although many more applications exist.

In the case of water heating with a heat pump for which the heat transfer enhancement technique was developed, the refrigerant will flow in the inner tube and the water in the counterflow direction in the annulus. The purpose of this paper is to determine experimentally the heat transfer and pressure drop characteristics of single phase water in the annulus of the angled spiralling tape heat exchanger and to develop correlations that describe each of these characteristics.

Experimental Method

Facility

The experimental set-up consisted of a closed system, using

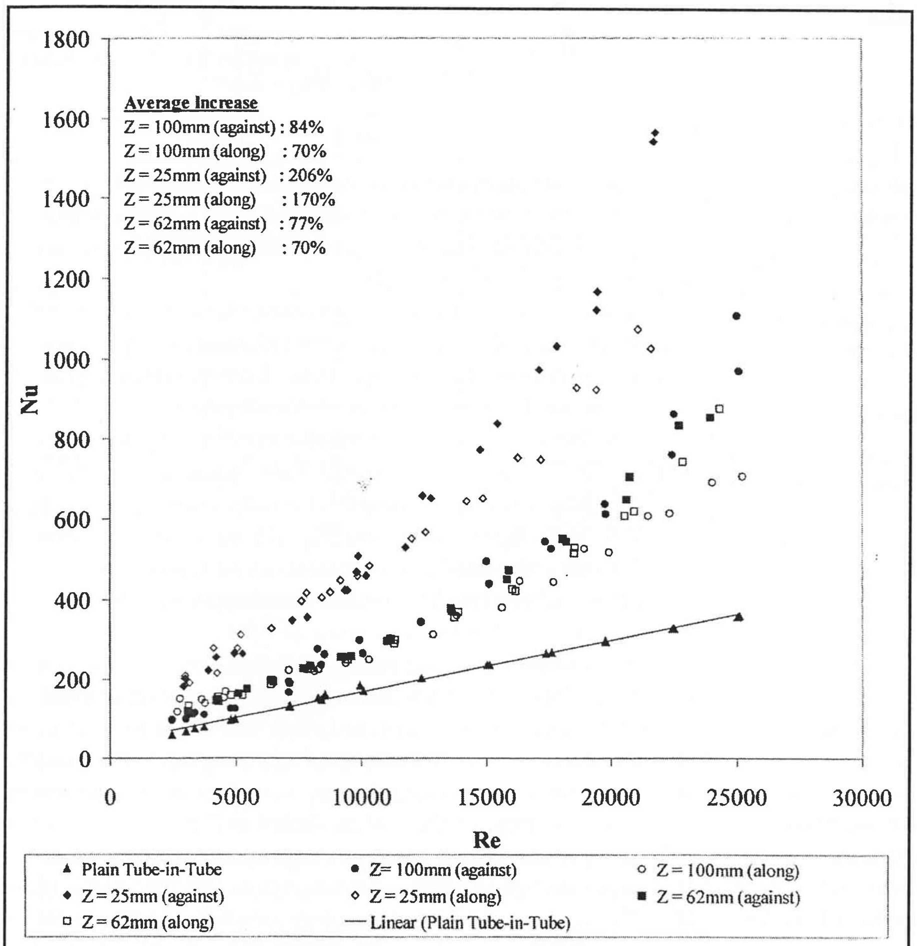

Figure 4: Nusselt Number Comparison for Flow Against and Along the Curvature of the Tape

Foam insulation with a thickness of (± 10 mm was used to minimize heat losses to the atmosphere. Hot water flowed in the annulus of the heat exchanger, while cold water flowed in an opposite direction in the inner tube. Also shown in Fig. 2 is a schematic representation of the temperature and pressure drop attachment points of the test section through which the water flows. Two mercury manometers were connected to the experimental set-up to measure the pressure drop over the annulus and inner tube. The distance over which the pressure drop was measured for the inner tube (l_i) was ± 3.11 m, while for the annulus (l_o) the distance was ± 2.87 m.

Four calibrated K-Type thermocouples (± 0.1°C accuracy) were installed on the surface of the hot- and cold-water, inlet and outlet points. The thermocouples were attached tightly onto the tube surfaces with insulation tape. Insulation was placed over the thermocouples to minimize heat transfer to the atmosphere.

The flow through the heat exchanger was calculated by means of two normal positive displacement meters (± 2% inaccuracy) installed at the hot- and cold-water outlets. Using a stopwatch and the amount of water that had flowed through the system over the given time, the mass flow could be determined accurately. Rotameters (± 4% inaccuracy at full scale) were used for setting the flow to different flow rates. As the accuracy of the rotameters was not as high as with measuring flow with the positive displacement meters, the rotameters were only used for adjusting the flow to approximate predetermined values. The rotameters were also used to confirm the calculations from the positive displacement meters.

both a hot-water and cold-water cycle. The heat exchanger was tested with water in both the inner tube and annulus, as it is not necessary to use refrigerant in the inner tube for the determination of the annulus characteristics. Both the hot-water and cold-water systems consisted of the following: the test section (Fig. 2), a reservoir, pump, two valves, a positive displacement meter, a flow control valve and a rotameter. The cold-water tank was connected to a chiller to cool the water, while the hot-water tank had an electric resistance heater connected to heat the water.

The tube-in-tube heat exchanger consisted of two concentric, hard-drawn, copper tubes, used primarily in the refrigeration industry. The outer tube had an outside diameter of 15.88 mm and an inside diameter of 14.26 mm. The inner tube had an outside diameter of 9.53 mm and an inside diameter of 8.11 mm. The total length of the heat exchange areas was 2.97 m. All the spiralling tape inserts were constructed from copper plate with a thickness of 1 mm. The tape inserts were cut to a height of 2.2 mm to give a bit of clearance between the annulus inner wall and the tape inserts. The first heat exchanger constructed had no spiralling tape inserts, while the other three heat exchangers had tape inserts with a pitch (Z in Fig. 1) of 25 mm, 62 mm and 100 mm respectively. These pitches correspond to twist ratios (y) of 0.731, 1.799 and 2.878 respectively.

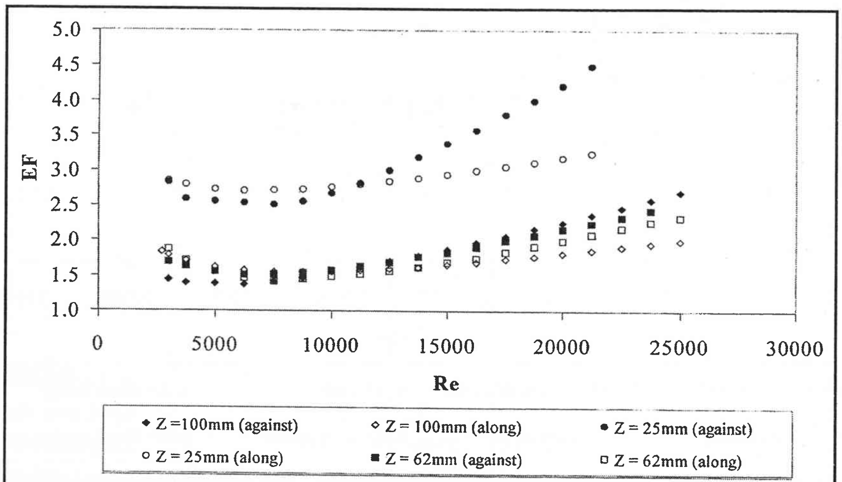

Figure 5: Enhancement Factor Comparison for Flow Against and Along the Curvature of the Tape

Experimental Procedure and Data Reduction

Heat Transfer

To obtain a reference base for comparison purposes the plain tube-in-tube heat exchanger was considered first. The heat transfer through both the inner tube and annulus was determined from inlet and outlet temperature measurements and mass flows ($Q = \dot{m}_w Cp\Delta T$). The average value (Q_{ave}) between the annulus heat transfer and inner tube heat transfer was considered as the correct value. The energy balance error was determined by comparing the annulus and inner tube heat transfer to this value, and resulted in errors below 5%.

With this heat transfer value the overall heat transfer coefficient can be determined from Eq. (1). The logarithmic mean temperature difference can be determined from the temperature measurements.

$$Q_{ave} = U_o A_o T_{LMTD} \tag{1}$$

where

$$\frac{1}{U_o A_o} = \frac{1}{A_i h_i} + \frac{\ln(d_o / d_i)}{2\pi k_{cu} L} + \frac{1}{A_o h_o} \tag{2}$$

The two heat transfer coefficients in Eq. (2) are not known and can be determined with Wilson plot-type methods. The modified Wilson plot technique described by Briggs and Young[12] was used. The Sieder-Tate equation[13] -format was used for both the inside heat transfer coefficient (Eq. (3)) and the outside heat transfer coefficient (Eq. (4)).

$$Nu_i = \frac{h_i d_i}{k_i} = C_i \operatorname{Re}_i^{0.8} \operatorname{Pr}_i^{0.333} \left(\frac{\mu}{\mu_w} \right)_i^{0.14} \tag{3}$$

$$Nu_o = \frac{h_o d_{eq}}{k_o} = C_o \operatorname{Re}_o^P \operatorname{Pr}_o^{0.333} \left(\frac{\mu}{\mu_w} \right)_o^{0.14} \tag{4}$$

Three sets of experiments were conducted where the Reynolds number in the inner tube was increased from 3 000 upwards, while the Reynolds number in the annulus was held constant at about 3 000, 4 500 and 7 000.

These results were plotted on a x-y coordinate system, from which the C_i and C_o multipliers were calculated for each individual experiment. The average value from the three experiments was used to determine the two multipliers.

The value of C_i was 0.01923 which corresponds well with the value of 0.027 of the original Sieder-Tate[13] equation. The value of C_o (the other multiplier) was 0.0726 and the value of P (Eq.(4)) was 0.8. With these values and Eq. (1) the theoretical heat transfer was calculated and compared to measurements. The average error between the theoretical and experimental results was 2.9% with a standard deviation of 1.9%.

When angled spiralling tape is considered, the heat transfer correlation for the inner tube given in Eq. (3) would still be valid. Therefore similar experiments as before were done for each of the three heat exchangers with angled spiralling tapes. Each one was considered with flow against the curvature of the tape and along the curvature of the tape as shown in Fig. 3.

During the experiments the flow rate through the inner tube was kept constant at a Reynolds number of approximately 16 000. The flow rate in the annulus was varied from a Reynolds number of 3 000 to approximately 25 000. From the temperature readings the heat transfer, logarithmic mean temperature difference and overall heat transfer coefficient could be determined. With the heat transfer coefficient of the inner tube known from Eq. (3), the heat transfer coefficient in the annulus could be determined.

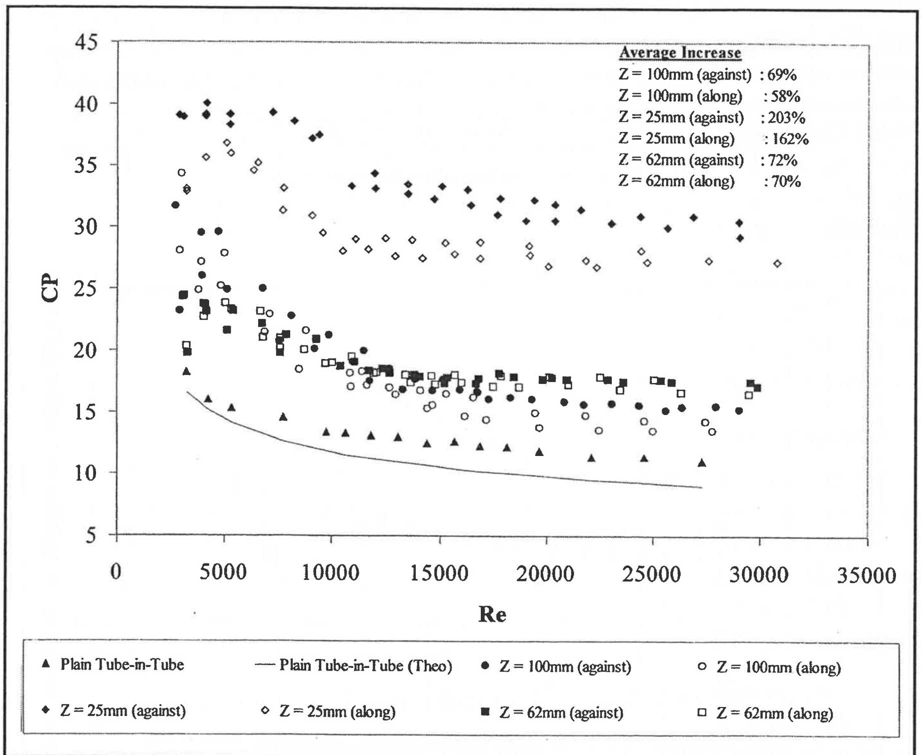

Figure 6: Pressure Drop Coefficient Comparison for Flow Against and Along the Curvature of the Tape

Pressure Drop

The pressure drops were determined from the differences of the manometers from which the pressure drop coefficient was determined from $CP = \Delta p/(1/2\rho v^2)$. For verification purposes the pressure drop coefficients for the inner tube and annulus were also determined analytically. The friction factor needed to determine the pressure drop coefficient was determined from the Swamee and Jain equation[13] for the inner tube. For the annulus the Petukhov equation[13] was used to calculate the friction factor. The Petukhov equation was based on the hydraulic diameter.

Results

In Fig. 4 the heat transfer results for the three experimental set-ups are plotted against the Reynolds number. The Nusselt number results for the plain tube-in-tube heat exchanger are also shown for comparison purposes (Eq.

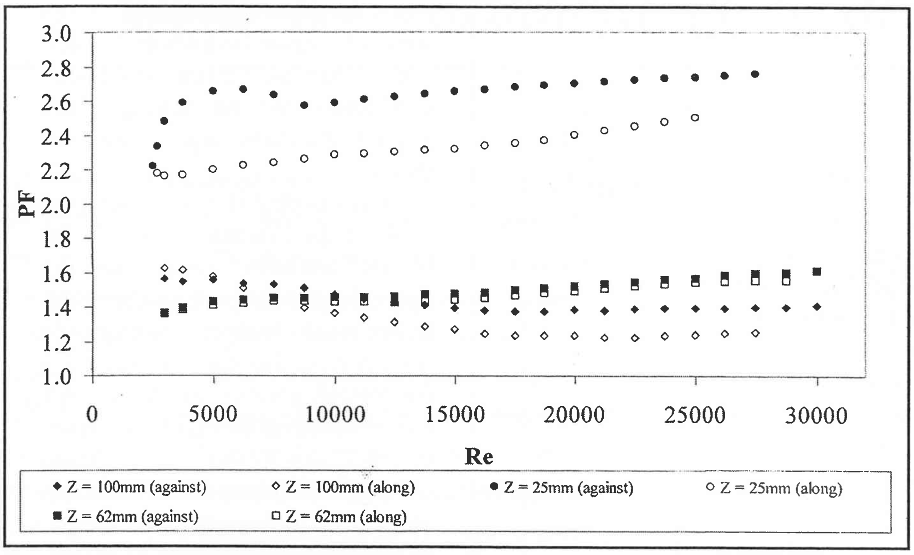

Figure 7: Pressure Factor Comparison for Flow Against and Along the Curvature of the Tape

(4)).

Since all the experiments were conducted for turbulent flow, the results lie in a Reynolds number range of 3 000 upwards. The experimental and theoretical results obtained from Eq. (4), for the plain tube-in-tube heat exchanger correlate well with each other. These results form an almost linear tendency, since the Nusselt number is directly proportional to $Re^{0.8}$. However, the results for the heat exchangers with angled spiralling tape inserts tend more to a parabolic correlation.

An average increase of 206% in the Nusselt numbers was noted from a plain tube-in-tube heat exchanger to a heat exchanger with a twist ratio of 0.731 (Z=25 mm) and flow against the curvature of the tape. The heat exchangers with a twist ratio of 1.799 (Z=62 mm) and 2.878 (Z=100 mm) produced very similar results. The average increase in the Nusselt number for these two set-ups was in the order of 80% for flow against the curvature of the tape and 70% for flow along the curvature.

In Fig. 5 the enhancement factor is plotted against the Reynolds number for all the conducted experiments. The enhancement factor was calculated as the ratio of the measured Nusselt number for a tape insert to the Nusselt number of a plain tube heat exchanger. The Nusselt number for the plain tube-in-tube heat exchanger was determined with linear interpolation, when a value was not available at the Reynolds number where measurements were taken for the configuration with tape inserts.

The enhancement factor decreased slightly to a minimum value for each experiment. From a Reynolds number of about 6 000 the enhancement factor increased with an increase in the Reynolds number. The highest increase in the enhancement factor was produced for the experimental set-up with a twist ratio of 0.731 (Z=25 mm) and flow against the curvature of the tape. The enhancement factor decreased from just below 3 to about 2.5, before increasing to about 4.5.

From Figs. 4 and 5, it was concluded that for Reynolds numbers below 11 000, the Nusselt numbers and thus enhancement factors for flow along the curvature of the tape were higher than for flow against the curvature for each experimental set-up. The situation switched for Reynolds numbers above 11 000, where the Nusselt numbers and enhancement factors for flow against the curvature of the tape were higher than for along the curvature.

Furthermore, the experimental set-ups with a twist ratio of 1.799 (Z= 62 mm) and 2.878 (Z= 100 mm) correlated well with each other, with only a slight difference between the average increases in Nusselt numbers and enhancement factors. A possible explanation for this correlation is that the performance follows a hyperbolic tendency as the pitch increases. From the results obtained it is assumed that the best performance with angled spiralling tape inserted into the annulus of a tube-in-tube heat exchanger is obtained at the smallest pitch of the tape. At the small pitch the flow through the annulus is induced with more swirl and turbulence by the tape inserts.

In Fig. 6 the pressure drop coefficients for the three experimental set-ups with twisted tape are given. Also included are the measured pressure drop coefficients for the plain tube-in-tube heat exchanger, as well as the theoretical estimates of the plain tube-in-tube heat exchanger.

As in the case of the heat transfer characteristics, all the experiments were conducted for turbulent flow. As the Reynolds number increases, the pressure drop coefficient values for the three experimental set-ups slightly decrease and then stay constant. The experimental and theoretical results for the plain tube-in-tube heat exchanger show some tendency to correlate well with each other. These results, as well as the results for the heat exchangers with angled spiralling tape inserts decreases sharply as the Reynolds number increases, before becoming almost constant.

The lowest average increase in the pressure drop coefficients for the heat exchangers with angled spiralling tape insert of 58% was obtained from the experimental set-up with twist ratio of 2.878 (Z= 100 mm) and flow along the curvature of the tape. This set-up resulted in similar results for the heat exchanger with a twist ratio of 1.799 (Z= 62 mm).

For Reynolds numbers below 8 000 the results start to scatter slightly from the power regression tendency. The probable cause is a measurement error of the manometer, since at these Reynolds numbers the pressure drop comprises of only a couple of millimetres in height.

The pressure factor for the three experimental set-ups was determined as the ratio of the measured pressure drop coefficient value for a tape insert to the pressure drop coefficient of a plain tube-in-tube heat exchanger. If necessary the pressure drop coefficient value for the plain tube heat exchanger was determined with linear interpolation, if a value was not available at the Reynolds number where measurements were taken for the configuration with tape inserts. The pressure factor comparison for all the experiments conducted are illustrated in Fig. 7.

The experimental set-up with a twist ratio of 2.878 (Z= 100 mm) and flow along the curvature of the tape resulted in the lowest increase in the pressure factor. The pressure factor starts at just above 1.6 and gradually decreases until just below 1.4. The other experimental set-ups resulted in an increase in the pressure factor.

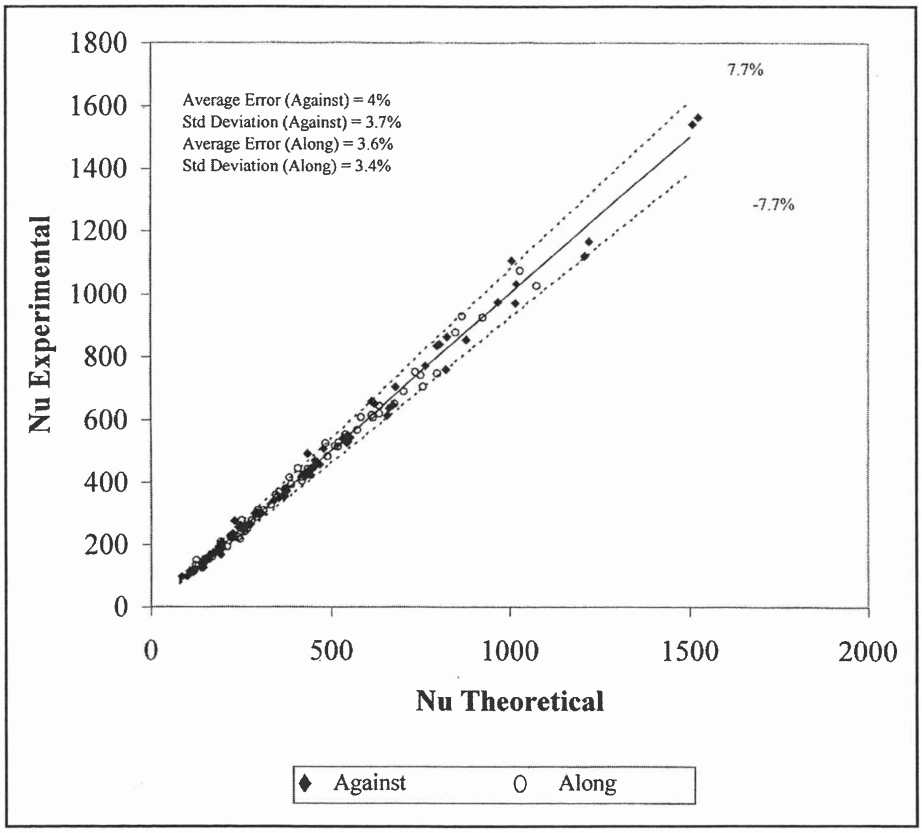

Figure 8: Error Calculation for the Nusselt Number Comparison for Flow Against and Along the Curvature of the Tape

The experimental set-ups with a pitch of 62 mm and 100 mm ($y = 1.799$ and $y = 2.878$) correlated relatively well with each other, with only a slight difference between the average increases in pressure drop coefficients and pressure factors. A possible explanation for this correlation is that the performance follows a hyperbolic tendency as the pitch increases. It can be assumed that the best performance for the pressure drop coefficients with angled spiralling tape inserted into the annulus of a tube-in-tube heat exchanger is obtained at the largest pitch of 100 mm.

Comparison of Results

With the heat transfer and pressure drop characteristics determined, the experimental set-ups were investigated as a whole. In Table 1, a comparison of all the experiments conducted on the heat exchangers with angled spiralling tape inserts are shown.

Only a slight difference is achieved between the average increase in the Nuss-elt number results and the average increase in the pressure drop coefficient results compared to those of a smooth tube. The highest increase in the Nusselt number results of 206% was achieved for the experimental set-up with a pitch of 25 mm ($y = 0.731$) and flow against the curvature of the tape. This set-up also resulted in the highest increase in the pressure drop coefficients of 203% as can be expected. The experimental set-up with the lowest increase in the pressure drop coefficient values was the set-up with a pitch of 100 mm ($y = 2.878$) and flow along the curvature of the tape, which produced an average increase of 58%. An average increase in the Nusselt number results of 70% was achieved.

Similar conclusions can be drawn from the enhancement factor and pressure factor results. The highest increase in the enhancement factor results is noted for the experimental set-up with a pitch of 25 mm ($y = 0.731$). The enhancement factor starts at about 2.5, slightly decreases and then rises to about 4.5. The pressure factor, however, starts at about 2.3, increases until levelling off at just above 2.5. Even though the enhancement factor and pressure factor begin at relatively the same values, a large difference results at the end.

This is the case for all other experimental set-ups, except the set-up with a pitch of 25 mm and flow along the curvature of the tape. For this set-up the enhancement factor and pressure factor stay relatively parallel to each other, with the enhancement factor starting at just below 3 and increasing to just above 3.

From Table 1 it can be concluded that the experimental set-ups with pitches of 62 mm ($y = 1.799$) and 100 mm ($y = 2.878$) resulted in very similar results for both the heat transfer and pressure drop characteristics. Only a slight difference between the average increases in pressure drop coefficients and Nusselt numbers could be noted, which resulted in similar results for the enhancement factors and pressure factors. An explanation for this is that the performance of the heat exchangers follows a hyperbolic tendency as the pitch increases. Thus the best performance for the Nusselt

Experimental Set-up	Pitch Z (mm)	Twist Ratio y	Curvature Flow	Increase			
				Nu	*CP*	*EF*	*PF*
1	100	2.878	Against	84%	69%	1.5 - 2.5	± 1.5
			Along	70%	58%	± 1.75	↑ 1.5 - ↓ 1.5
2	25	0.731	Against	206%	203%	2.5 - 4.5	2.25 - ↑ 2.5
			Along	170%	162%	↓ 3 - ↑ 3	2.25 -↓ 2.5
3	62	1.799	Against	77%	72%	.75 - 2.2	± 1.5
			Along	70%	70%	↓ 2 - ↑ 2	± 1.5

± : Results remain in vicinity; ↑ : Results are just above indicated value;
↓ : Results are just below indicated value.

Table 1: Heat Transfer and Pressure Drop Characteristics Comparison

numbers is achieved at a small pitch and the best performance for the pressure drop coefficients is achieved at a large pitch. This tendency could have been anticipated.

From the experimental results it is concluded that the experimental set-up with angled spiralling tape with a pitch of 25 mm (y = 0.731) produced the optimum performance. The highest increase in the Nusselt numbers over a plain tube-in-tube heat exchanger was achieved with this set-up. Unfortunately the highest increase in the pressure drop coefficients was also obtained with it. From the enhancement factor and pressure factor results, it was determined that a higher increase in the enhancement factor was achieved related to the increase in the pressure factor. This was not the case for the other two experimental set-ups studied.

Correlations Development

The procedure to develop the heat transfer coefficient correlation and friction factor correlation is similar. The experimental data was first plotted against the Reynolds number. Using curve fittings on these graphs, equations for each of the twist ratios studied was determined.

It was concluded that these equations incorporated functions dependent on the twist ratios. Using matrix algebra these functions were solved. Inserting these functions back into the equations resulted in the correlations for the prediction of the heat transfer and pressure drop characteristics. A detailed explanation is illustrated in Coetzee[14].

Heat Transfer Coefficient

Equation (4) was determined to predict the heat transfer coefficient of a plain tube-in-tube heat exchanger. Since this equation is in the form of the Sieder-Tate equation, only a correlation to the tape geometry needs to be added. Thus, this equation forms the basis to develop an equation to predict the heat transfer coefficient for a heat exchanger with angled spiralling tape inserts. Eq. (5) depicts the final heat transfer correlation.

$$Nu = 0.0726\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.333}\left(\frac{\mu}{\mu_w}\right)^{0.14}\left[f_1(y)\mathrm{Re}^2 + f_2(y)\mathrm{Re} + f_3(y)\right] \qquad (5)$$

where for flow against the curvature of the tape:

$$f_1(y) = 2.256 \times 10^{-9} y^2 - 10.989 \times 10^{-9} y + 16.03 \times 10^{-9} \qquad (6)$$

$$f_2(y) = -21.04 \times 10^{-6} y^2 + 125.27 \times 10^{-6} y - 211.33 \times 10^{-6} \qquad (7)$$

$$f_3(y) = 0.449y^2 - 2.329y + 4.503 \qquad (8)$$

and for flow along the curvature of the tape, the functions are given by:

$$f_1(y) = -0.3969 \times 10^{-9} y^2 + 1.233 \times 10^{-9} y + 3.369 \times 10^{-9} \qquad (9)$$

$$f_2(y) = 4.866 \times 10^{-6} y^2 - 26.27 \times 10^{-6} y - 54.17 \times 10^{-6} \qquad (10)$$

$$f_3(y) = 0.449y^2 - 1.969y + 4.021 \qquad (11)$$

The comparison of the experimental Nusselt numbers and the Nusselt numbers obtained from Eq. (5) for flow against and along the curvature of the tape is illustrated in Fig. 8.

It was determined that the correlation results in an average error of 4% with a standard deviation of 3.7% for flow against the curvature of the tape. In the case of flow along the curvature of the tape an average error of 3.6% was determined with a standard deviation of 3.4%.

Friction Factor

Equation (12) depicts the final friction factor correlation developed for tube-in-tube heat exchangers with angled spiralling tape inserts.

$$f = g_1(y)\,\mathrm{Re}^{g_2(y)} \qquad (12)$$

where for flow against the curvature of the tape:

$$g_1(y) = 0.3618y^2 - 1.047y + 0.9186 \qquad (13)$$

$$g_2(y) = -0.0669y^2 + 0.1656y - 0.2282 \qquad (14)$$

and for flow along the curvature of the tape, the functions are given by:

$$g_1(y) = 0.6283y^2 - 1.6519y + 1.1939 \qquad (15)$$

$$g_2(y) = -0.0797y^2 + 0.1814y - 0.2392 \qquad (16)$$

The comparison of the experimental friction factors and the friction factors obtained from Eq. (12) for flow against the curvature of the tape is illustrated in Fig. 9.

The correlation results in an average error of 3.7% with a standard deviation of 3.7% for flow against the curvature of the tape. In the case of flow along the curvature of the tape an average error of 4.3% was determined with a standard deviation of 3.3%.

Conclusion

To study the heat transfer and pressure drop characteristics of angled spiralling tape augmentation devices, four experimental set-ups were constructed. Three tube-in-tube heat exchangers were manufactured with angled spiralling tape in the annulus with a pitch of 25 mm, 62 mm and 100 mm. The fourth experimental set-up was a plain tube-in-tube heat exchanger to be used for comparison purposes.

It was determined that the heat exchanger with a pitch of 25 mm and with flow against the curvature of the tape resulted in the highest increase in the Nusselt number, with an average increase of 206%. This heat exchanger also had the best enhancement factor increase. As penalty this heat exchanger also had the highest increase in the pressure drop, with an average increase of 203%. The heat exchanger with the lowest increase in the pressure drop, was for a pitch of 100 mm and flow along the curvature of the tape. The average increase was 58%.

The experimental set-ups with a pitch of 62 mm and 100 mm correlated well with each other for both the heat transfer and pressure drop results. A possible explanation for this correlation is that the performance follows a hyperbolic tendency as the pitch increases. From the results obtained it is assumed that the best heat transfer performance is obtained at the smallest pitch of the tape. The opposite is assumed for the best pressure drop performance, where the lowest increase should be obtained at the highest pitch of the tape.

This heat exchanger with a pitch of 25 mm resulted in the highest increase in the Nusselt numbers over a plain tube-in-tube heat exchanger. Unfortunately, an increase in the pressure drop coefficients close to the Nusselt number increase was also obtained. However, a higher increase in the enhancement factor was achieved than the increase in the pressure factor. This was not the case for the other experimental set-ups, thus,

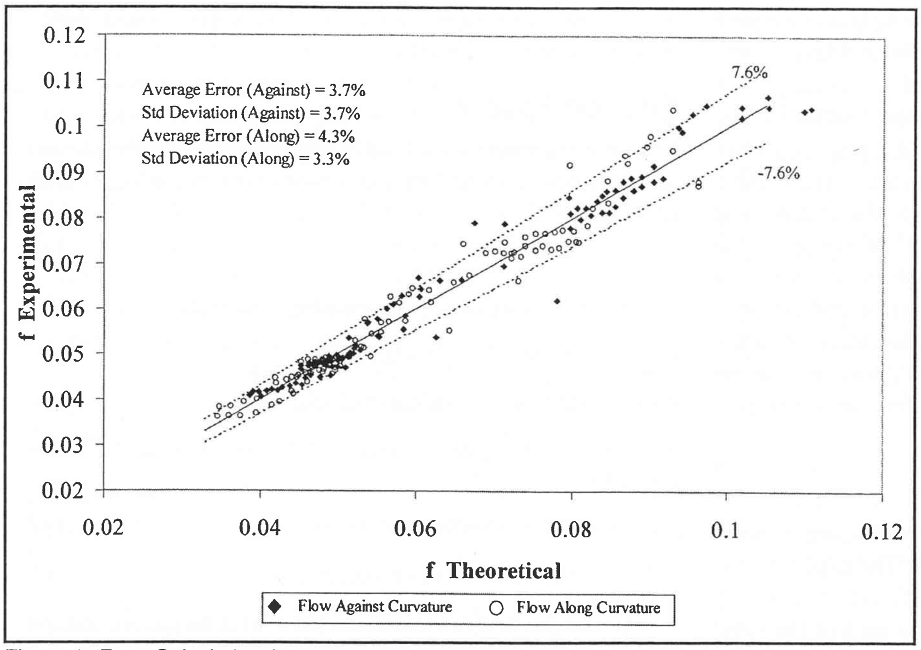

Figure 9: Error Calculation for the Friction Factor Comparison for Flow Against and Along the Curvature of the Tape

it is concluded that the heat exchanger with a pitch of 25 mm produced the optimum performance.

With the experimental results concluded, a correlation to predict the heat transfer coefficient and friction factor for a heat exchanger with angled spiralling tape inserts was developed. Both the correlations developed resulted in average errors below 8% for flow against and along the curvature of the tape. Since the experimental results correlated well with the values predicted by the equations, it is concluded that the correlations are accurate enough to predict the heat transfer and pressure drop characteristics for a heat exchanger with angled spiralling tape inserted into the annulus.

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