Experimental studies of a double-pipe helical heat exchanger

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Abstract

An experimental study of a double-pipe helical heat exchanger was performed. Two heat exchanger sizes and both parallel flow and counterflow configurations were tested. Flow rates in the inner tube and in the annulus were varied and temperature data recorded. Overall heat transfer coefficients were calculated and heat transfer coefficients in the inner tube and the annulus were determined using Wilson plots. Nusselt numbers were calculated for the inner tube and the annulus. The inner Nusselt number was compared to the literature values. Though the boundary conditions were different, a reasonable comparison was found. The Nusselt number in the annulus was compared to the numerical data. The experimental data fit well with the numerical for the larger heat exchanger. But, there were some differences between the numerical and experimental data for the smaller coil; however these differences may have been due to the nature of the Wilson plots. Overall, for the most part the results confirmed the validation of previous numerical work.

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1. Introduction

Several studies have indicated that helically coiled tubes are superior to straight tubes when employed in heat transfer applications [1,5,11,15]. The centrifugal force due to the curvature of the tube results in the development of secondary flows (flows perpendicular to the axial direction) which assist in mixing the fluid and enhance the heat transfer. In straight tube heat exchangers there is little mixing in the laminar flow regime, thus the application of curved tubes in laminar flow heat exchange processes can be highly beneficial. These situations can arise in the food processing industry for the heating and cooling of either highly viscous liquid food, such as pastes or purees, or for products that are sensitive to high shear stresses.

The majority of the studies involving helical coils and heat exchange have focused on two major boundary conditions; constant wall temperature and constant heat flux [17,12]. However, these boundary conditions are not present in most fluid-to-fluid heat exchangers. Haraburda [4] described a method of calculating the shell side heat transfer coefficients for a coil-in-shell type heat exchanger; however, the calculations were based on empirical relations for flow over a bank of non-staggered circular tubes. Patil et al. [7] describe a method for designing a coil-in-shell heat exchanger similar to that of Haraburda [4].

The heat exchanger proposed in this work is unlike the coil-in-shell heat exchangers of Patil et al. [7] and Haraburda [4] in that the shell is replaced by a coiled tube. This changes the flows of the two fluids from being perpendicular to parallel flow or counterflow. This configuration results in secondary flows in both the inner tube and in the annulus, as both sections are curved and subjected to centrifugal forces. Furthermore,
the coil-in-shell tube could, theoretically, have regions in the shell next to the coil where there is poor circulation. This problem could be avoided by using a double-pipe configuration. Karahalios [6] and Petrakis and Karahalios [8–10] have studied the fluid flow and heat transfer in a curved pipe with a solid core. They showed that the size of the core affected the flow in the annulus, with flows approaching parabolic for large cores and the flow being skewed towards the outer wall [10]. Karahalios [6] studied the heat transfer in a curved annulus; however, a constant temperature gradient on both the outer and inner walls of the annulus was used as the thermal boundary condition. These boundary conditions are not appropriate for the heat exchanger proposed in this work, where the temperature of the inner wall of the annulus will be dictated partly by the fluid temperature in the inner tube. Furthermore, the outer wall of the annulus will be insulated. This type of heat exchanger was numerically studied by Rennie [13], and was partly validated by using literature data. However, the wall boundary conditions for heat transfer were different from literature data, and there were no comparable annulus values. Thus the numerical results need to be validated with experiments using a double-pipe helical heat exchanger.

2. Objective

The objectives of this study were:

1. Design, build, and instrument two double-pipe helical heat exchangers, the difference between the two being the size of the inner tube.
2. Experimentally evaluate the heat transfer characteristics of a double-pipe helical heat exchanger for both parallel flow and counterflow configurations.
3. Compare the experimental results with the work of Rennie [13].

3. Materials and methods

3.1. Heat exchanger

The heat exchanger was constructed from copper tubing and standard copper connections. The outer tube of the heat exchanger had an outer diameter of 15.9 mm and a wall thickness of 0.8 mm. The inner tube had an outer diameter of either 9.5 mm or 6.4 mm, both with wall thickness of 0.8 mm. The end connections are shown in Fig. 1, which were constructed from standard copper tees and reducers. Each coil had a radius of curvature (measured from the centre of the inner tube) of 235.9 mm. Small holes were drilled in the outer tube and tapped so that set screws could be used to ensure that the inner tube was centred prior to the final soldering of the end connections, which then held the inner tube in place. After soldering the set screws were removed and the holes covered so that the fluid flow in the annulus would not be disturbed. The heat exchanger consisted of one loop.

3.2. Experimental apparatus

The heat exchanger was tested in the setup show in Fig. 2. Cold tap water was used for the fluid flowing in the annulus. A large reservoir was used and a small submersible pump (#523086, Little Giant Pump...
Company, Oklahoma City, OK) was used to provide flow to the annulus. The water in the annulus was not circulated, and additional water was periodically added to the reservoir. However, this did not significantly change the depth of water in the reservoir and hence the flow rate was not compromised. The flow was controlled by a flowmeter (Model 4L53, Cole Parmer, Vernon Hills, IL) with an attached metering valve, allowing flows to be controlled and measured between 100 and 1500 cm$^3$/min. Hot water for the inner tube was provided with an Isotemp Refrigerated Circulator (model 1013, Fisher Scientific, Pittsburgh, PA) set at 60 °C. This water was circulated via the internal pump. The flow rate for the inner tube was controlled by an identical flowmeter and metering valve as described for the annulus flow. Flexible PVC tubing was used for all the connections. Type-K thermocouples (Omega Engineering, Stanford, CT) were inserted into the flexible PVC tubing to measure the inlet and outlet temperatures for both fluids. Temperature data was recorded using a data acquisition/switch unit (Model 34970A, Agilent, Palo Alto, CA) connected to a computer.

### 3.3. Experimental procedure

Flow rates in the annulus and in the inner tube were varied. The following five levels were used: 100, 300, 500, 700, and 900 cm$^3$/min. All possible combinations of these flow rates in both the annulus and the inner tube were tested. These were done for both coils, and in parallel flow and counterflow configurations. Furthermore, three replicates were done for every combination of flow rate, coil size and configuration. This resulted in a total of 300 trials. Temperature data was recorded every ten seconds. The data used in the calculations was from after the system had stabilized. Temperature measurements from the 120 s of the stable system were used, with temperature reading fluctuations within ±0.15 °C. Though the type-K thermocouples had limits of error of 2.2 °C, when placed in a common water solution the readings at steady state were all within ±0.1 °C. All the thermocouples were constructed from the same roll of thermocouple wire, and hence the repeatability of temperature readings was high.

### 3.4. Calculation of heat transfer coefficients

The overall heat transfer coefficient, $U_o$, was calculated from the temperature data and the flow rates using the following equation [18]:

$$U_o = \frac{q}{A_o \text{LMTD}}$$  \hspace{1cm} (1)

where $A_o$ is the surface area; $q$ is the heat transfer rate; and LMTD is the log mean temperature difference, based on the inlet temperature difference, $\Delta T_1$, and the outlet temperature difference, $\Delta T_2$, using the following equation [18]:

$$\text{LMTD} = \frac{(\Delta T_2 - \Delta T_1)}{\ln \left( \frac{\Delta T_2}{\Delta T_1} \right)}$$  \hspace{1cm} (2)

Heat transfer coefficients for the annulus side, $h_o$, and for the inner tube side, $h_i$, were calculated using traditional “Wilson plots” as described by Rose [14]. Wilson plots have been used in other heat exchanger studies [2,16]. Wilson plots allow the heat transfer coefficients to be calculated based on the overall temperature difference and the rate of heat transfer, without the requirement of wall temperatures. This method was chosen to avoid the disturbance of flow patterns and heat transfer while attempting to measure wall temperatures. Wilson plots are generated by calculating the overall heat transfer coefficients for a number of trials where one fluid flow is kept constant and the other is varied. In this work, the flow in the inner tube was kept constant and the flow in the annulus was varied for the five different flow rates mentioned above. The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients by the following equation [18]:

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + \frac{A_o}{2\pi kL} + \frac{1}{h_o}$$  \hspace{1cm} (3)
where $D_i$ is the inner diameter of the annulus; $d$ is the diameter of the inner tube; $k$ is the thermal conductivity of the wall; and $L$ is the length of the heat exchanger.

After calculating the overall heat transfer coefficients, the only variables in Eq. (3) that are unknown are the heat transfer coefficients. By keeping the mass flow rate in the inner tube constant, it is then assumed that the inner heat transfer coefficient is constant. The outer heat transfer coefficient is assumed to behave in the following manner with the fluid velocity in the annulus, $v_o$:

$$h_o = Cv_o^n$$  \hspace{1cm} (4)

Eq. (4) was placed into Eq. (3) and the values for the constant $C$ and the exponent $n$ were determined through curve fitting. The inner and outer heat transfer coefficients could then be calculated. This procedure was repeated for each inner flow rate, coil size, configuration, and replicate. This resulted in 60 Wilson plots, and 60 inner heat transfer coefficients. For each Wilson plot, five outer heat transfer coefficients were calculated, one for each of the flow velocities used.

4. Results and discussion

4.1. Overall heat transfer coefficients

Overall heat transfer coefficients for parallel flow are presented in Figs. 3 and 4 for the large coil and the small coil, respectively. The overall heat transfer coefficient is plotted against the inner Dean number for each of the flow rates in the annulus. The trends are typical for a fluid-to-fluid heat exchanger with the overall heat transfer coefficient increasing with both inner and annulus flows. For a given annulus flow rate, increasing the inner flow rate results in an eventual asymptotic overall heat transfer coefficient. It appears that this asymptotic value is reached at a lower Dean number for the large coil than the smaller coil.

The results from the counterflow configuration were similar to the parallel flow, as is expected, as changing the flow direction should have negligible effects on the heat transfer coefficients. Heat transfer rates, however, are much higher in the counterflow configuration, due the increased log mean temperature difference. The counterflow versus the parallel flow overall heat transfer coefficients are plotted in Fig. 5, where the values plotted against each other are from the same experimental parameters. There is a reasonable agreement between the two values.

Fig. 6 shows the overall heat transfer coefficient versus the inner Dean number for the case when the mass flow rates in the inner tube and in the annulus are identical. For this case the overall heat transfer coefficient is higher for the large tube at the same inner Dean number, which is the same result as found in Rennie [13].
4.2. Inner Nusselt numbers

The inner Nusselt numbers are presented in Fig. 7 (with ±2 standard errors). These values are the average inner Nusselt number at each Dean number (average of parallel flow and counterflow values). The data is compared to the correlation of Dravid et al. [3]. Their correlation was developed based on both numerical and experimental data with a constant wall heat flux. The Dean number range for Dravid et al. [3] was from 50 to 2000 and the Prandtl number range was from 5 to 175. In order to calculate these correlations, Prandtl numbers were essential for the flow. These were evaluated using the arithmetic mean temperature of the corresponding fluid (average of inlet and outlet temperatures). Decent agreement between the experimental and literature values indicate that the use of existing correlations for heat transfer in helical coils could be used to estimate inner heat transfer rate in a double-pipe helical heat exchanger. The experimental data fit close to the correlation of Dravid et al. [3]. However the Wilson plots did not work so well with the smaller coil, as there was more variance in the results.

4.3. Annulus Nusselt numbers

The Nusselt numbers in the annulus were calculated and correlated to the following modified Dean number:

\[ \text{De}^* = \frac{\rho v_0}{\mu} \left( \frac{D_o^2 - D_i^2}{D_o + D_i} \right) \left( \frac{D_o - D_i}{R} \right)^{1/2} \]

where \( \rho \) is the fluid density; \( \mu \) is the fluid viscosity; and \( R \) is the radius of curvature of the coil. The correlation developed numerically in Rennie [13] is shown with the experimental data from this study in Fig. 8 (with ±2 standard errors). The data for the large coil fits well with the correlation; though the data for the small tube tends to diverge with increasing Dean number. This may indicate that the Nusselt number correlations need to take into account factors other than just the Dean number, or that the particular Dean number should be modified. In Rennie [13], the correlation of the data to a Dean number that used a different curvature ratio was also performed. However, in that case, separate correlations need to be made for each tube size, as could be done in here. Furthermore, the entrance region between the numerical and experimental heat exchanger were different. In the numerical work the inlet and outlet flows in the annulus were straight, whereas in this work the flow underwent a 90° bend at the entrance to the heat exchanger. It is difficult to predict the effect of the end connections of the Nusselt number. This could cause larger entrance effects which may not be applicable to heat exchangers with more than one loop. Furthermore, the data in this experimental work and the numerical data differ in Prandtl number, which was not considered in the correlation of the numerical data, as the numerical data used a Prandtl number for water at 20 °C, and the average water temperature in the annulus in this

![Fig. 7. Inner Nusselt number versus inner Dean number.](image)

![Fig. 8. Annulus Nusselt number versus annulus Dean number.](image)
work was higher. This should lower the correlation from Rennie [13] bringing it closer to the small coil. Despite the differences between the two coil sizes, the Nusselt numbers for each coil size can be correlated to the modified Dean number linearly, as was done in Rennie [13]. Further research needs to be performed on this entrance region to determine its effect on the heat transfer rates.

The low values for the small coil could also be due to the nature of the Wilson plots. It was noted that small changes in the coefficients used in the Wilson plots had a small effect on the inner Nusselt numbers, but much larger effects on the annulus Nusselt numbers. So the Wilson plots may be part of the reason for the divergence between the experimental and numerical results.

5. Conclusions

An experimental study of a double-pipe helical heat exchanger was performed using two differently sized heat exchangers. The mass flow rates in the inner tube and in the annulus were both varied, as well as both parallel flow and counterflow configurations were tested. There were little differences between the overall heat transfer coefficients for the parallel flow and counterflow configurations. However, heat transfer rates were much higher in the counterflow configuration due to the larger average temperature difference between the two fluids.

The Nusselt number in the inner tube was compared to the literature values. Though the boundary conditions are different between the experimental data and the literature data, there was reasonable agreement between the two. The Nusselt number in the annulus was compared to the numerical work, and though the experimental data fit well with the numerical data for the larger coil, there was a deviation between the data for the small coil. There may be a cause for the difference in the Nusselt number as the entrance region was very different between the numerical and experimental setups. The 90° elbow used in the connection in the experimental setup could increase the Nusselt number significantly in the entrance region, further work needs to be done to quantify this effect.

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