

EXPERIMENTAL AND NUMERICAL STUDIES OF A SPIRAL PLATE HEAT EXCHANGER

by

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An experimental and numerical study of heat transfer and flow characteristics of spiral plate heat exchanger was carried out. The effects of geometrical aspects of the spiral plate heat exchanger and fluid properties on the heat transfer characteristics were also studied. Three spiral plate heat exchangers with different plate spacing (4 mm, 5 mm, and 6 mm) were designed, fabricated and tested. Physical models have been experimented for different process fluids and flow conditions. Water is taken as test fluid. The effect of mass flow rate and Reynolds number on heat transfer coefficient has been studied. Correlation has been developed to predict Nusselt numbers. Numerical models have been simulated using CFD software package FLUENT 6.3.26. The numerical Nusselt number have been calculated and compared with that of experimental Nusselt number.

Key words: *spiral plate heat exchanger, heat transfer, Nusselt number*

Introduction

Heat exchanger is a device in which energy is transferred from one fluid to another across a solid surface. Compact heat exchangers are characterized with its large amount of surface area in a given volume compared to traditional heat exchangers, in particular the shell-and-tube type. The development and investigation of compact heat exchangers, has become an important requirement during the last few years. Compact heat exchangers are of two types, spiral and plate type heat exchangers. Spiral heat exchanger is self cleaning equipment with low fouling tendencies, easily accessible for inspection or mechanical cleaning and with minimum space requirements. Egner and Burmeister [1] numerically studied spiral ducts of rectangular section using computational fluid dynamics techniques and determined the Nusselt number as a function of the Dean number, showing the strong dependence of the heat transfer coefficient upon the spiral radii. Burmeister [2] found more approximate solution to determine the thermal effectiveness. Martin [3] numerically studied the heat transfer and pressure drop characteristics of spiral plate heat exchanger. The apparatus used in the investigations had a cross-section of 5 mm × 300 mm, number of turns $n = 8.5$, core diameter of 250 mm, outer diameter of 495 mm and 5 × 5 cylindrical bolts in a rectangular in line arrangement of 61 mm × 50 mm, and for data in the range of $400 < Re < 30000$. Nusselt number correlation for their particular set up with water as a medium has presented. An average in-tube heat transfer coefficient in a spi-

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rally coiled tube was proposed by Naphon and Wongwises [4]. The test section was a spirally coiled heat exchanger consisting of six layers of concentric spirally coiled tubes. The experiments were performed under cooling and dehumidifying conditions and considered the effects of inlet conditions of both working fluids on the in-tube heat transfer coefficient. The results obtained from experiments were compared with those calculated from other correlations. A new correlation for the in-tube heat transfer coefficient for spirally coiled tube was proposed. In their second and third papers, Naphon and Wongwises [5, 6] developed a mathematical model to determine the performance and heat transfer characteristics of spirally coiled finned tube heat exchangers under wet-surface conditions. In addition, the heat transfer characteristics and performance of a spirally coiled heat exchanger under dry surface conditions were studied theoretically and experimentally. The calculated and measured results were in reasonable agreement.

Naphon and Wongwises [7] experimentally investigated the average tube-side and air-side heat transfer coefficients in a spirally coiled finned tube heat exchanger under dry- and wet-surface conditions. The test section was a spiral-coil heat exchanger, which consists of six layers of concentric spirally coiled tube. The chilled water and the hot air were used as working fluids. The effects of the inlet conditions of both working fluids flowing through the heat exchanger on the heat transfer coefficients were discussed. New correlations based on the data gathered during their work for predicting the tube-side and air-side heat transfer coefficients for the spirally coiled finned tube heat exchanger were proposed. Yang and Chiang [8] studied the effects of the Dean number, Prandtl number, Reynolds number and the curvature ratio on the heat transfer for periodically varying curvature curved-pipe inside a larger diameter straight pipe to form a double-pipe heat exchanger. The results showed that the heat transfer rate increased by up to 100% as compared with a straight pipe. All of the experimental data were regressed to obtain the correlation of the Nusselt number.

The turbulence and heat transfer in two types of square sectioned U-bend duct flows with mild and strong curvature, using recent second moment closures, were predicted by Suga [9, 10]. A two-component limit turbulence model and the wall reflection free model were presented. The results showed that the two-component limit turbulence model was reliable in the case of strong curvature. Huttel and Friedrich [11] used direct numerical simulation for turbulent flow in straight, curved and helically coiled pipes in order to determine the effects of curvature and torsion on the flow patterns. They showed that turbulent fluctuations are reduced in curved pipes compared to the straight pipes. They also demonstrated that the effect of torsion on the axial velocity is much lower than the curvature effect. Egner and Burmeister [1] conducted a numerical study of spiral ducts of rectangular section using computational fluid dynamics techniques and determined the Nusselt number as a function of the Dean number, showing the strong dependence of the heat transfer coefficient upon the spiral radii. They demonstrated that except for the entry regions, the heat transfer coefficient is nearly constant, however, at entry regions, heat transfer coefficients may be even as 50% larger than the fully developed values. An important contribution of their work is the general conclusion for estimating the thermal entry length for laminar Reynolds numbers between 100 and 500.

Experimental set-up

The experimental set-up is shown in fig. 1. The heat exchanger was constructed using 316 stainless steel plates. The spiral plate heat exchanger had a width of 304 mm and a plate thickness of 1 mm. The total heat transfer area is 2.24 m². The end connections are shown in fig. 1. Plate had a radius of curvature of 172.9 mm; the gap between the plate is 4 mm, 5 mm, and 6 mm.

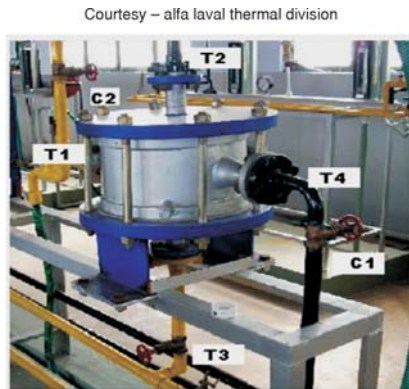


Figure 1. Photographic view of experimental set-up; C1 and C2 – control valves, T1, T2, T3 and T4 – thermocouples

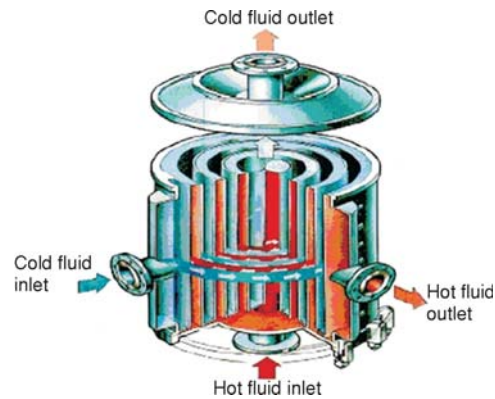


Figure 2. Spiral flow heat exchanger (courtesy - alfa laval thermal division)

A pump was used to provide flow to the cold fluid side. The flow rate was measured by a calibrated flow meter, which is regulated by a control valve. The flow rate was maintained between 0.2 and 1.0 kg/s. The cold fluid inlet pipe is connected to the periphery of the spiral plate heat exchanger and the outlet is taken from the centre of the heat exchanger. The hot fluid is heated between 60 and 80 °C using a steam boiler. The hot fluid is stored in a reservoir, which is pumped to the heat exchanger with the help of a pump. The hot fluid inlet pipe is connected at the center core of the spiral plate heat exchanger and the outlet pipe is taken from periphery of the heat exchanger. The flow rate of hot fluid was maintained by an identical flow meter. Temperature data was recorded using a data acquisition unit connected to a personal computer. To ensure the repeatability of data, all the instruments are calibrated periodically.

Water was used as the hot fluid. The inlet hot fluid flow rate was kept constant and the inlet cold fluid flow rate was varied using a control valve. The flow of cold and hot fluid was varied using control valves, C1 and C2, respectively. Hot and cold fluid flow paths of heat exchanger are shown in fig. 2. Thermocouples T1 and T2 were used to measure outlet temperature of hot and cold fluids, respectively; T3 and T4 were used to measure the inlet temperature of hot and cold fluids, respectively. For different cold fluid flow rate the temperatures at the inlet and outlet of hot and cold fluids were recorded. The same procedure was repeated for different hot fluid flow rates. The data related to cold fluid temperatures and mass flow rates were recorded. Temperature data was recorded in the span of ten seconds.

Results and discussion

The effect of Reynolds number on heat transfer coefficient is shown in fig. 3. Hot fluid flow rate (0.5 kg/s) was kept constant for different plate spacing. From the fig. 3 it can be seen that the heat transfer coefficient increases as Reynolds number increases. Reynolds number is a dimensionless number that gives a measure

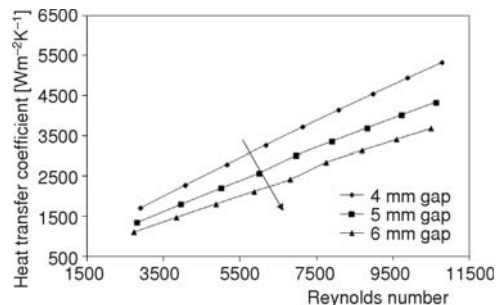


Figure 3. Effect of plate spacing on heat transfer coefficient – water

of the ratio of inertial forces to viscous forces. It can also be seen that the heat transfer coefficient increases with decrease in plate spacing. The plate spacing of 4 mm shows the maximum heat transfer coefficient than that of 6 mm. This may be due to higher turbulence for smaller gap size, which results in higher heat transfer coefficient for smaller gap size than the plate with larger gap. High turbulence in the medium this gives a higher convection, which results in efficient heat transfer between the media. When the plate has a narrow pattern, the pressure drop is greater and the heat transfer coefficient is accordingly somewhat greater. Heat transfer coefficient is a quantitative characteristic of convective heat transfer between a fluid medium and the surface flowed over by the fluid.

Empirical correlation has been developed based on the experimental data. It was found that correlation of the type as in eq. (1):

$$\text{Nu} = a \text{Re}^b \text{Pr}^c \left(\frac{H}{b}\right)^d \quad (1)$$

The correlation (1) was tested for all Reynolds number and Prandtl number. For water-water system the constant a , b , c , and d are shown in eq. (2):

$$\text{Nu} = 0.02846 \text{Re}^{0.828} \text{Pr}^{0.336} \left(\frac{H}{b}\right)^{-0.0871} \quad (2)$$

Regression analysis shows that for 189 data with an error of $\pm 4\%$ was obtained. The eq. (2) is valid for the range :

$$2560 < \text{Re} < 11770, \quad 2.82 < \text{Pr} < 4.39, \quad \text{and} \quad 50.6 < H/b < 76$$

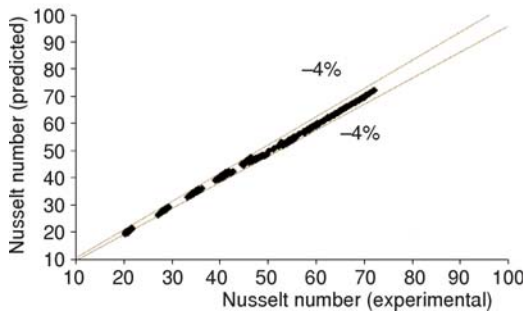


Figure 4. Comparison of Nusselt number with present correlation – water

namely the RNG k - ϵ model and the Reynolds stress transport model (RES). In the case of the finite volume method, two levels of approximation are needed for surface integrals: the integral is approximated in terms of the variables values at one location on the cell face, the midpoint point rule was used in this task; the cell face values are approximated in terms of the nodal values (control volume (CV) centers), the linear interpolation was used in this task. The volume integrals were approximated by a second-order approximation replacing the volume integral by the product of the mean value and the CV volume.

Figure 4 compares the results from the present correlation with the experimental data. The Nusselt number (Nu) is the ratio of convective to conductive heat transfer across the boundary. The majority of the data falls within $\pm 4\%$ of the proposed correlation in eq. (2).

The discrepancy between the experimental data and the predicted results is due to the difference in the configurations of test sections, the difference in the wall boundary conditions and uncertainty of the correlations.

All calculations were performed in a double precision segregated steady-state solver. In the simulations of flows two different models were employed for turbulence modeling,

Simulations were performed using water as hot and cold fluid. Flow rates in the cold fluid and in the hot fluid were varied. The following nine levels were used: 0.2-1.0 kg/s. Different combinations of these flow rates in both the cold fluid and the hot fluid were simulated. These were done for gaps of 4 mm, 5 mm, and 6 mm configurations between the plates.

A predicted contours of static temperature plot of spiral plate heat exchanger for hot fluid mass flow rate of 0.5 kg/s and cold fluid mass flow rate of 0.4 kg/s is shown in figs. 5 to 7 for plate spacing of 4 mm, 5 mm, and 6 mm respectively. Figures 5 to 7 shows the cross-sectional view of heat exchanger. It can be seen that the temperature difference is more when the gap between the plates are less.

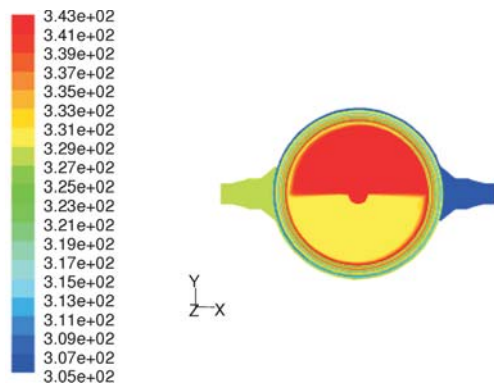


Figure 5. Contours of static temperature at $b = 4 \text{ mm}$ – water

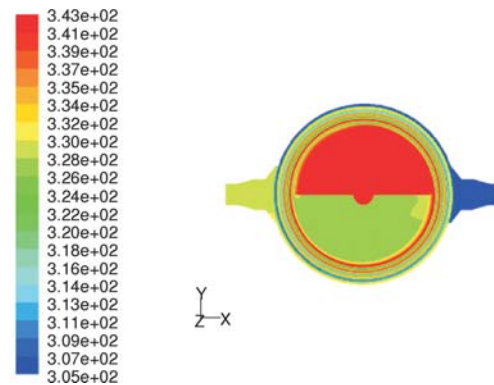


Figure 6. Contours of static temperature at $b = 5 \text{ mm}$ – water

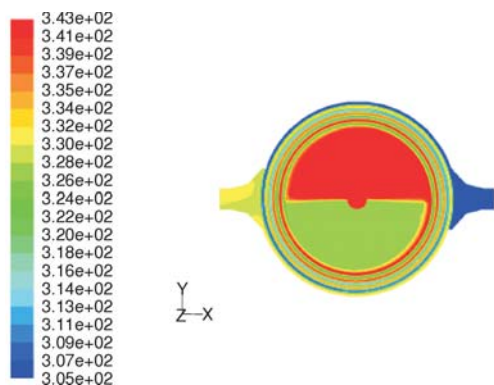


Figure 7. Contours of static temperature at $b = 6 \text{ mm}$ – water

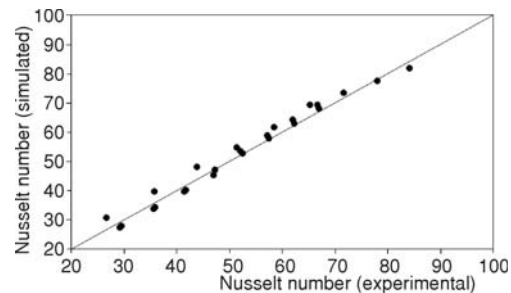


Figure 8. Comparison of experimental and simulated Nusselt number – water

Comparisons between the Nusselt numbers obtained from the experiments with those calculated from the simulations are shown in fig. 8. It can be seen that the majority of the data falls within $\pm 17.8\%$ of the experimental data. The discrepancy between the experimental data

and the simulated results is due to the difference in the configurations of test sections, the difference in the wall boundary conditions, and uncertainty of the correlations.

Conclusions

Heat transfer and flow characteristics of a spiral plate heat exchanger were studied. Three spiral plate heat exchangers were designed, fabricated, and tested. Heat transfer coefficient for water was studied for different mass flow rate and physical models. Experimental Nusselt number was compared to that of literature data for three different configurations. Empirical correlation was obtained to predict Nusselt numbers for water. Heat transfer coefficient was studied for water. The space between the plates was varied from 4 mm to 6 mm. Among these, the plate having space 4 mm shows more heat transfer coefficient. The numerical Nusselt number was compared with the experimental data which shows a reasonable fit.

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