



An Experimental Investigation of Heat Transfer Coefficients for Spiral Plate Heat Exchanger

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Abstract

Spiral plate heat exchangers play a vital role in cooling high density and high viscous fluids. This paper presents an experimental investigation of convective heat transfer co-efficient for electrolytes using spiral plate heat exchanger. The test section consists of a Plate of width 0.3150 m, thickness 0.001 m and mean hydraulic diameter of 0.01 m. The mass flow rate of hot fluid is varying from 0.4 kg sec⁻¹ to 0.8 kg sec⁻¹ and the mass flow rate of cold fluid varies from 0.3 kg sec⁻¹ to 0.8 kg sec⁻¹. Experiments have been conducted by varying the mass flow rate, temperature and pressure of cold fluid, keeping the mass flow rate of hot fluid constant. The effects of relevant parameters on spiral plate heat exchanger are investigated. The data obtained from the experimental study are compared with the theoretical data. Besides, a new correlation for the nusselt number that can be used for practical applications is proposed.

Keywords: Spiral plate heat exchanger, Reynolds number, Nusselt number, Heat transfer coefficient, Mass flow rate

1. Introduction

Heat exchanger is a device in which energy is transferred from one fluid to another across a solid surface. Spiral plate heat exchangers are good heat transfer devices in process industries that handle slurries and viscous fluids. A great number of applications are there for the spiral plate heat exchanger due to its compactness, high heat transfer coefficients, resistance to fouling and economy in operation. It is more satisfying to chemical, flush, and reversing fluid cleaning techniques because of the single passage. No insulation is used outside the exchanger because of the cold fluid flowing in the outermost passage, resulting in negligible heat loss, if any, due to its temperature closer to atmospheric temperature. It is used in the cellulose industry for cleaning relief vapors in sulfate and sulfite mills. Considerable research is being pursued in spiral heat exchanger in heat transfer and flow areas.

Seban and McLaughlin (Seban, R. A. and E. F. McLaughlin., 1963) calculated heat transfer in coiled tubes for both laminar and turbulent flows. Plot of Nusselt versus Graetz numbers were presented for coils with curvature ratios of 17 and 104 with Reynolds numbers ranging from 12 to 5600 for the laminar flow region. Prandtl numbers ranged from 100 to 657. Heat transfer and pressure loss in steam heated helically coiled tubes were studied by Rogers and Mayhew (Rogers, G. F. C. and Y. R. Mayhew., 1964). They observed that even for a steam heated apparatus, uniform wall temperature was not obtained, mainly due to the distribution of the steam condensate over the coil surface. Mori and Nakayama (Mori, Y. and W. Nakayama., 1965) studied the fully developed flow in a curved pipe with a uniform heat flux for large Dean Numbers. Flow and temperature fields were studied both theoretically and experimentally. They assumed that the flow was divided into two sections, a small boundary layer near the pipe wall, and a large core region making up the remaining flow. Pressure drop and heat transfer for laminar flow of glycerol was presented by Kubair and Kuloor (Berg, R. R. and C. F. Bonilla., 1950) for different types of coiled pipes, including helical and spiral

configurations. Reynolds numbers were in the range of 80 to 6000 with curvature ratios in the range of 10.3 to 27. The number of turns varies from 7 to 12. The results of Kubair and Kuloor (Berg, R. R. and C. F. Bonilla., 1950) match with those of Seban and McLaughlin (Seban, R. A. and E. F. McLaughlin., 1963) at low Graetz numbers, but deviated at higher Graetz numbers.

Outside-film and inside-film heat transfer coefficients in an agitated vessel were studied by Jha and Rao (Jha, R. K. and M. R. Rao., 1967). Five different coils were studied, along with different speeds and locations of the agitator. They derived an equation to predict the Nusselt number based on the geometry of the helical coil and the location of the agitator. Numerical studies for uniform wall heat flux with peripherally uniform wall temperature for Dean numbers in the range of 1-1200, Prandtl numbers of 0.005-1600, and curvature ratios of 10 to 100 for fully developed velocity and temperature fields were performed by Kalb and Seader (Kalb, C. E. and J. D. Seader., 1972). They found that the curvature ratio parameter had insignificant effect on the average Nusselt number for any given Prandtl number. Kalb and Seader (Kalb, C. E. and J. D. Seader., 1974) furthered this work by applying the method to the case of a uniform wall-temperature boundary condition with Dean numbers up to 1200, Prandtl numbers and curvature ratios in the ranges of 0.05 to 1600 and 10 to 100, respectively. Their results illustrate that there is a slight effect of curvature on the peripheral variation of the Nusselt number. However, it did not affect the average Nusselt number. The effects of buoyancy forces on fully developed laminar flow with constant heat flux were studied analytically by Yao and Berger (Yao, L.-S., and S. A. Berger., 1978). Their studies were based on the Boussinesq approximation for the buoyancy forces and analyzed for both horizontally and vertically orientated curved pipes. Nusselt number relationships based on the Reynolds number, Raleigh number and Dean Number were presented for both orientations.

Laminar flow and heat transfer were studied numerically by Zapryanov et al (Zapryanov, Z., Christov, C. and E. Toshev., 1980) using a method of fractional steps for a wide range of Dean numbers from 10 to 7000 and Prandtl numbers from 0.005 to 2000. Their work focused on the case of constant wall temperature and showed that the Nusselt number increased with increasing Prandtl numbers, even for cases at the same Dean number. They also presented a series of isotherms and streamlines for different Dean and Prandtl numbers. The effect of buoyancy on the flow field and heat transfer was studied numerically by Lee et al (Lee, J. B., H. A. Simon, and J. C. F. Chow., 1985) for the case of fully developed laminar flow and axially steady heat flux with a peripherally constant wall temperature. They found that buoyancy effects resulted in an increase in the average Nusselt number, as well as modifying of the local Nusselt number allocation. It was also found that the buoyancy forces result in a rotation of the orientation of the secondary flow patterns. The heat transfer to a helical coil in an agitated vessel was studied by Havas et al. (Havas, G., A. Deak, and J. Sawinsky., 1987). Correlation was developed for the outer Nusselt number based on a modified Reynolds number, Prandtl number, viscosity ratio, and the ratio of the diameter of the tube to the diameter of the vessel. Heat transfer enhancements due to chaotic particle paths were studied by Acharya et al. (Acharya, N., Sen, M., and H. C. Chang., 1992, Acharya, N., Sen, M., and H. C. Chang., 2001) for coiled tubes and alternating axis coils. They developed two correlations of the Nusselt number, for Prandtl numbers less than and greater than one, respectively. Lemenand and Peerhossaini (Lemenand, T. and H. Peerhossaini., 2002) developed a Nusselt number correlation based on the Reynolds number, Prandtl number and the number of bends in the pipe. For the same Reynolds and Prandtl numbers, their work showed that the Nusselt number slightly drops off with increasing number of bends.

Heat transfer for pulsating flow in a curved pipe was numerically studied by Guo et al (Guo, L., Chen, X., Feng, Z., and B. Bai., 1998) for fully developed turbulent flow in a helical coiled tube. In their work they examined both the pulsating flow and the steady state flow. They developed the following correlation (1) for steady turbulent flow for the Reynolds number range of 6000 to 1 80 000

$$Nu = 0.328 Re^{0.58} Pr^{0.4} \quad (1)$$

They found that the Reynolds number was increased to very large values [$>1\ 40\ 000$], the heat transfer coefficient for coils began to match the heat transfer coefficient for straight tubes. They also presented correlations of the peripheral local heat transfer coefficients as a function of the average heat transfer coefficients, Reynolds number, Prandtl number, and the location on the tube wall. Inagaki et al. (Inagaki, Y., Koiso, H., Takumi, H., Ioka, I., and Y. Miyamoto, 1998) studied the outside heat transfer coefficient for helically coiled bundles for Reynolds numbers in the range of 6000 to 22 000 and determined that the outside Nusselt number described by the following relationship (2) for their particular setup.

$$Nu = 0.78 Re^{0.51} Pr^{0.3} \quad (2)$$

Heat transfer and flow characteristics in the spiral and helical coils have been studied by a number of researchers. Although some information is currently available to calculate the performance of the spiral plate heat exchanger, the study of heat transfer and flow characteristics in spiral plate heat exchanger has received relatively less attention. This is because the heat transfer and flow characteristics of spiral plate heat exchanger have been studied. In the present study, the heat transfer and flow characteristics of mixture of water and Mgso₄, Feso₄ and Nacl for spiral plate heat exchanger have been experimentally studied, in addition to the development of a new correlation for Nusselt number.

2. Experimental Setup

The experimental setup consists of spiral plate heat exchanger, thermocouple, pumps and tanks as shown in Figure 2.1. The parameters of heat exchanger are shown in the Table 2.1. The hot fluid inlet pipe is connected at the center core of the spiral heat exchanger and the outlet pipe is taken from periphery of the heat exchanger. The hot fluid is heated by pumping the steam from the boiler to a temperature of about 60-70° C and connected to hot fluid tank having a capacity of 1000 liters then the hot solution is pumped to heat exchanger using a pump. Thus the counter flow of the fluid is achieved. The cold fluid inlet pipe is connected to the periphery of the exchanger and the outlet is taken from the centre of the heat exchanger. The cold fluid is supplied at room temperature from cold solution tank and is pumped to the heat exchanger using a pump.

3. Experimental Procedure

The heat transfer and flow characteristic of mixture of water and Mgso4, Feso4 and Nacl is tested using an Alfa Laval; Model P5-VRB, Spiral plate heat exchanger as shown in Figure. 2.1. The inlet hot fluid flow rate is kept constant and the inlet cold fluid flow rate is varied using a control valve. The flow of hot and cold fluid is varied using control valves, C2 and C1 respectively. Thermocouple T1 and T2 are used to measure inlet temperature of cold and hot fluids respectively; T3 and T4 are used to measure the outlet temperatures of cold and hot fluids respectively. For different cold fluid flow rate the temperatures at the inlet and outlet of hot and cold fluids are recorded, after achieving the steady state. The same procedure is repeated for different hot fluid flow rates and the data related to temperatures, the corresponding temperatures and mass flow rates are recorded. The mass flow rate is noted by using the Rotometer R1 and R2, fitted at the outlet of the corresponding fluids. Experimental conditions are shown in Table 3.1.

4. Results and Discussion

4.1 Reynolds Number Vs Heat Transfer Coefficient

In this study electrolytes such as Feso4, Nacl and Mgso4 have been used. The concentration of electrolytes has been varied from 5% to 15 % (volume %). The effect of heat transfer rate on Reynolds number for different electrolytes is shown in Figure 4.1. It is observed that the heat transfer rate increases with increasing Reynolds number of mixture of water and electrolytes. Solvation of ion with water is the reason enough to account for maximum heat transfer rate.

4.2 Reynolds Number Vs Nusselt Number

Figure 4.2 shows the variation of the Nusselt number with Reynolds number of electrolytes for different inlet temperatures. From the experimental results, it is shown that the Nusselt number increases with increasing of Reynolds number of mixture of water and electrolytes.

4.3 Nusselt number (Experimental) Vs (Predicted)

Figure 4.3 shows the comparisons of the Nusselt numbers obtained from the experiment conducted with those calculated theoretically. It can be noted that the experimental and predicted Nusselt numbers fall within $\pm 20\%$. The major discrepancy between the measured data and calculated results may be due to the difference in the configuration of test sections and uncertainty of the correlation.

The Nusselt number correlation (3) for spiral plate heat exchanger is expressed as follows.

The correlation is obtained by fitting a total of 153 experimental data. ($R^2=0.97$)

$$N_u = 0.0476 R_e^{0.832} P_r^{-0.149} \quad (3)$$

$$3800 > R_e < 8500$$

4.4 Comparison of Experimental number with Holger Martin Correlation

Holger martin correlation (4)

$$N u = 0.04 R e^{0.74} P r^{0.4} \quad (4)$$

$$4.10^2 < R_e < 3.10^4$$

Comparisons of the Nusselt numbers obtained from the present experiment with those calculated from the existing correlation are shown in Figure 4.4. It can be noted that the values obtained from the correlation are slightly consistent with the experimental data and lie within $\pm 21\%$ for the Holger martin correlation.

5. Conclusion

This paper presents new experimental data from the measurement of the heat transfer coefficient of mixture of water and electrolyte flows in a spiral plate heat exchanger. The effects of relevant parameters are investigated. The data obtained from the present study are compared with the theoretical data. In addition, a new correlation based on the experimental data is given for practical applications.

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7. Nomenclature

Nu- Nusselt number

Re-Reynolds number

Pr-Prandtl number

h- Heat transfer coefficient [$\text{W m}^{-2} \text{K}^{-1}$]

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Table 2.1 Dimensions of the Spiral plate Heat Exchanger

Parameters	Dimensions (P5-VRB plate)
Plate width, m	0.3150
Plate thickness, m	0.0010
Mean channel spacing, m	0.0050
Mean hydraulic diameter, m	0.0100
Heat transfer area, m ²	2.2400

The above table gives the dimensions and the parameters of the Spiral Plate Heat Exchanger.

Table 3.1 Experimental Conditions

Sl.No	Variables	Range
1	Hot water temperature	65-50 °C
2	Cold Water temperature	30-50 °C
3	Mass flow rate of hot water	0.4-0.8 kg sec ⁻¹
4	Mass flow rate of cold Water	0.3-0.8 kg sec ⁻¹

The table above provides the experimental conditions of the hot and cold water temperature, mass flow rate of hot and cold water.

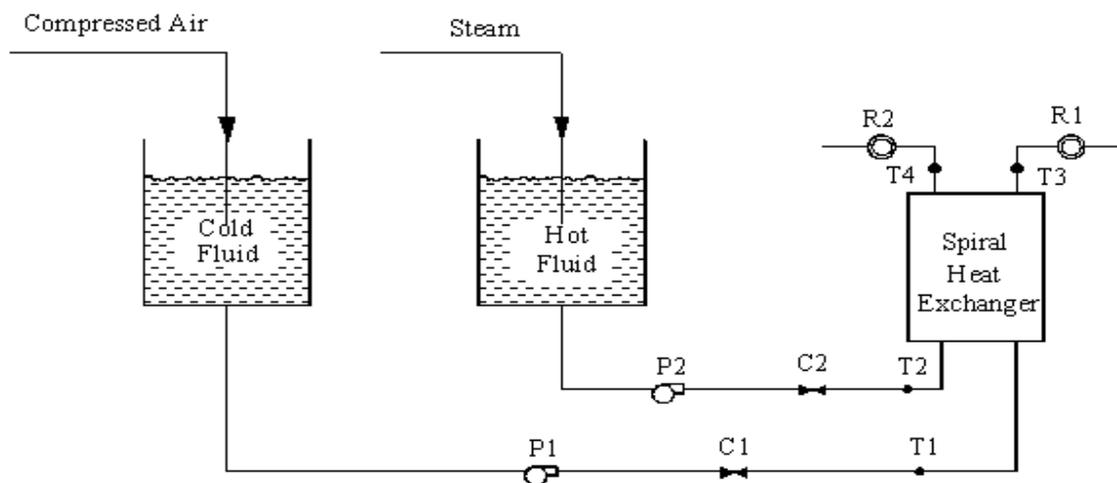


Figure 2.1 Schematic diagram of experimental apparatus

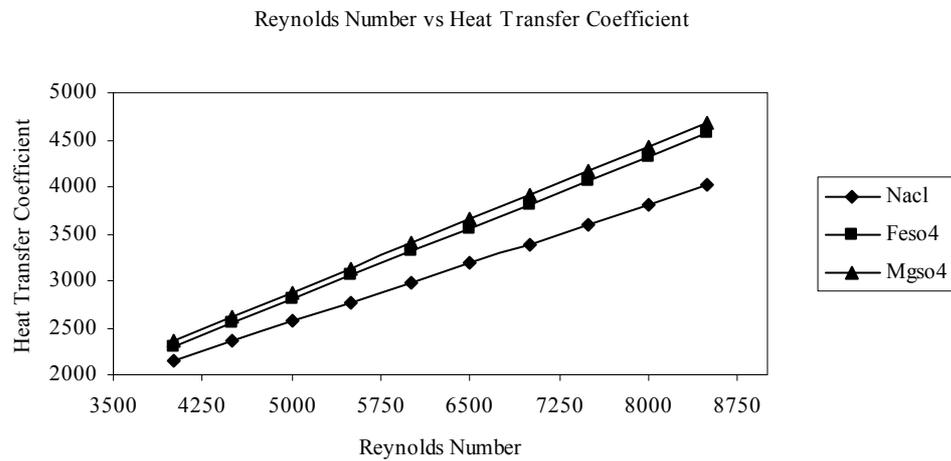


Figure 4.1 Variation of Reynolds number with Heat transfer rate

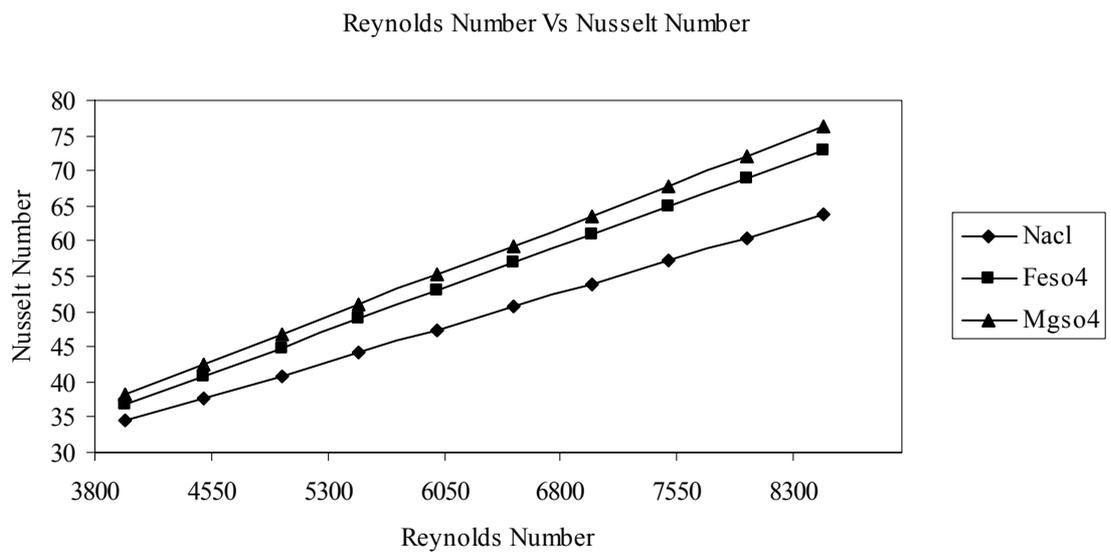


Figure 4.2 Variation of Reynolds number with Nusselt number

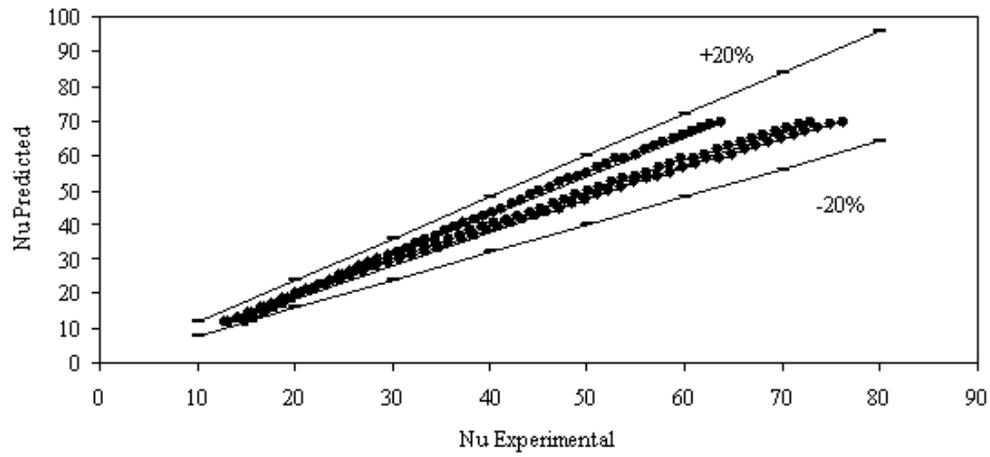


Figure 4.3 Variation of Nusselt number (Experimental) with Nusselt number (Predicted)

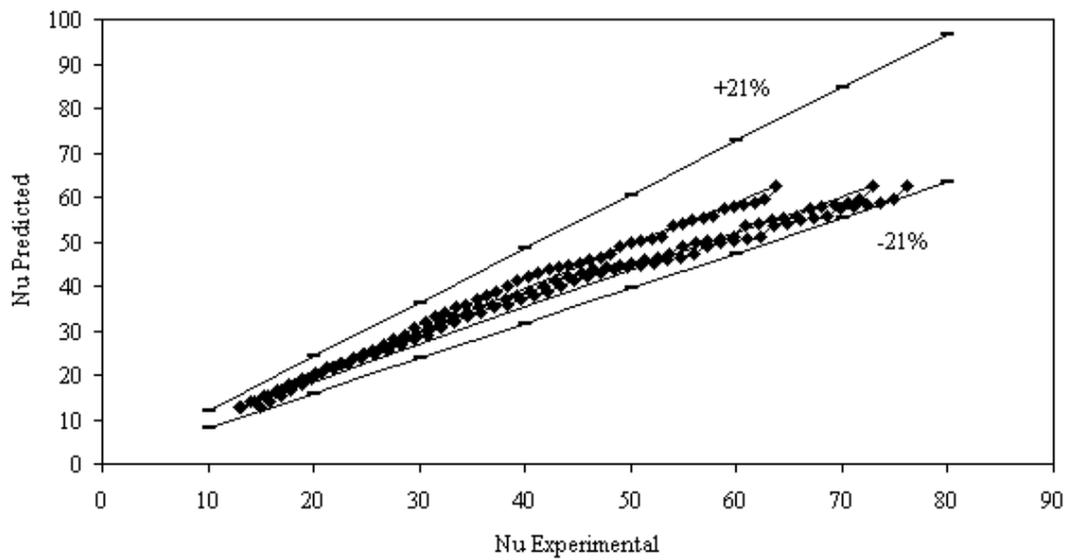


Figure 4.4 Comparison of Experimental data with Holger Martin correlation