

AN EXPERIMENTAL STUDY OF SPIRAL PLATE HEAT EXCHANGER FOR ELECTROLYTES

R.Rajavel¹, K. Saravanan

¹Department of Mechanical Engineering,
Department of Chemical Engineering,
Kongu Engineering College,
Perundurai, Erode,
Tamilnadu, India-638 052
E-mail:rajavel_7@yahoo.com

Received 07 December 2007
Accepted 12 April 2008

ABSTRACT

This paper presents an experimental investigation of convective heat transfer coefficient for electrolytes using spiral plate heat exchanger. The test section consists of a plate of width 0.315 m, thickness 0.001 m and mean hydraulic diameter of 0.01 m. The mass flow rate of water (hot fluid) is 0.636 kg s⁻¹ and the mass flow rate of electrolytes (cold fluid) varying from 0.483 kg s⁻¹ to 0.704 kg s⁻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 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 which can be used for practical applications is developed.

Keywords: spiral plate heat exchanger, heat transfer coefficient, mass flow rate, electrolytes.

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 most basic compact heat exchangers have a volume less than 50 % of that of a comparable shell-and-tube heat exchanger, for a given duty. The development and investigation of compact heat exchangers has become an important requirement during the last few years. The interest stems from various reasons viz. decreasing raw material and energy resources, the increasing environmental pollution and increasing costs for manufacturing and operation of heat

exchangers. 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. Considerable research is being pursued in spiral and helical heat exchanger in heat transfer and flow areas.

Seban and McLaughlin [1] 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 [2]. 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 [3] 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 [4] 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 [4] match with those of Seban and McLaughlin [1] 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 [5]. 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 [6]. They found that the curvature ratio parameter had insignificant effect on the average Nusselt number for any given Prandtl number. Kalb and Seader [7] 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 show 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 [8]. 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. [9] using a method of fractional steps for a wide range of Dean (10 to 7000) and Prandtl [0.005 to 2000] numbers. 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. [10], 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 studied by Havas et al. [11] and a 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. [12, 13] 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 [14] 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. [15] 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 180000

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

They found that the Reynolds number was increased to very large values (>140000), the heat trans-

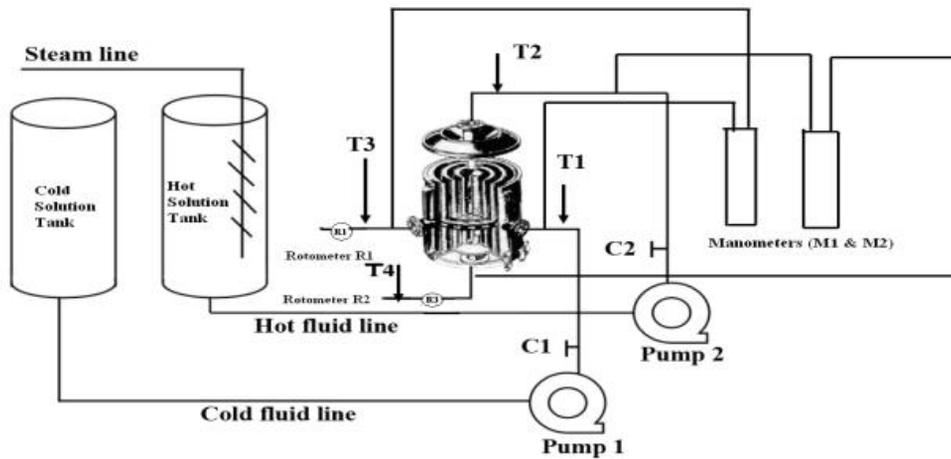


Fig. 1. Schematic diagram of experimental apparatus.

fer 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. [16] studied the outside heat transfer coefficient for helically coiled bundles for Reynolds numbers in the range of 6000 to 22000 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 curved tubes have been studied by a number of researchers. Although some information is currently available to calculate the performance of the spiral plate heat exchanger, there is still room to discuss whether it gives reliable prediction of the performance. This is because the heat transfer and flow characteristics of spiral plate heat exchanger has been studied. In the present study, the heat transfer and flow characteristics of electrolytes for spiral plate heat exchanger have been experimen-

Table 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

tally studied, in addition to the development of a new correlation for nusselt number.

EXPERIMENTAL

The experimental setup

The experimental setup consists of spiral heat exchanger, thermometer, and steam purging coil, manometers, pumps and tanks as shown in Fig. 1. The parameters of heat exchanger are shown in the Table 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 0.5 hp pump.

Table 2. Experimental Conditions.

S.No	Variables	Range
1	Hot water temperature	38-75 °C
2	Electrolyte temperature	25-53 °C
3	Mass flow rate of hot water	0.636 kg s ⁻¹
4	Mass flow rate of Electrolyte	0.483-0.704 kg s ⁻¹

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 0.5 hp pump.

Experimental procedure

The heat transfer and flow characteristic of electrolytes is tested using an Alfa Laval; Model P5-VRB, Spiral plate heat exchanger as shown in Fig. 2. Electrolytes are used as the cold fluids and the water as the hot fluid. 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, C1 and C2, respectively. Thermometers T1 and T2 are used to measure inlet temperature of cold and hot fluids, respectively; T3 and T4 are used to measure the outlet temperature 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 determined by using the Rotameter fitted at the outlet of the corresponding fluids.

The performance of spiral plate heat exchangers mainly depend on mass flow rate of fluid, flow area, pressure drop and logarithmic temperature difference between entering and leaving fluids.

CALCULATION METHODOLOGY

The heat load (Q, Watts) is calculated using the expression

$$Q = m_h C_{p_h} (\Delta T)_h$$

where m_h - Mass flow rate of hot fluid $kg s^{-1}$
 C_{p_h} - Specific heat of hot fluid $jk g^{-1} k^{-1}$
 $(\Delta T)_h$ - Temperature drop of hot fluid, k

The overall heat transfer coefficient is obtained from the relation

$$U = \frac{Q}{A(\Delta T)_{lm}}$$

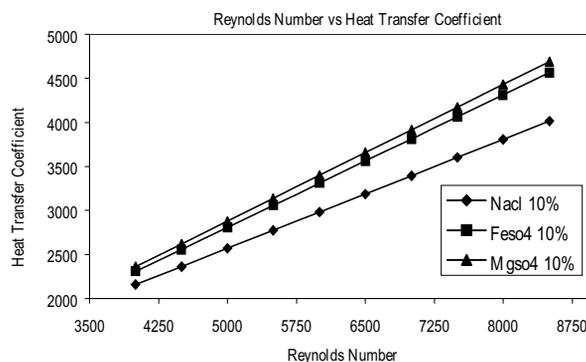


Fig. 2. Variation of Re with heat transfer coefficient for 10 % concentration of electrolytes.

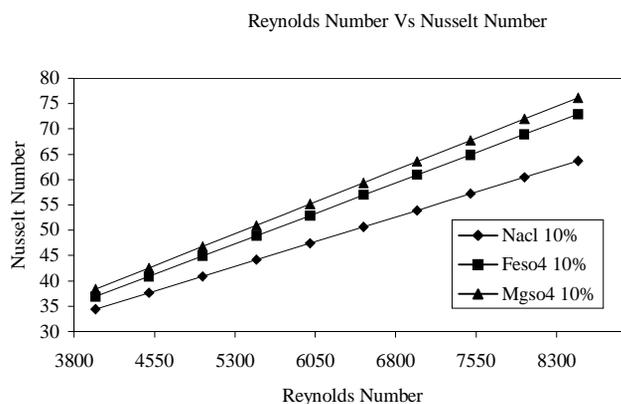


Fig. 3. Variation of Re with Nu for electrolytes.

U-Overall heat transfer coefficient $wm^{-2}k^{-1}$
 A-Heat transfer area, m^2
 $(\Delta T)_{lm}$ -Logarithmic mean temperature difference, k
 The hot fluid side heat transfer coefficient (h_h) is estimated using the following relation

$$Nu = 0.04 Re^{0.74} Pr^{0.4}$$

$$Nu = \frac{h_h d_e}{k_h}$$

where d_e -equivalent diameter of the flow channel, m
 k_h -thermal conductivity of hot fluid, $wm^{-1}k^{-1}$

The cold side heat transfer coefficient is calculated using the expression

$$\frac{1}{U} = \frac{1}{h_h} + \frac{t}{k_{ss}} + \frac{1}{h_c}$$

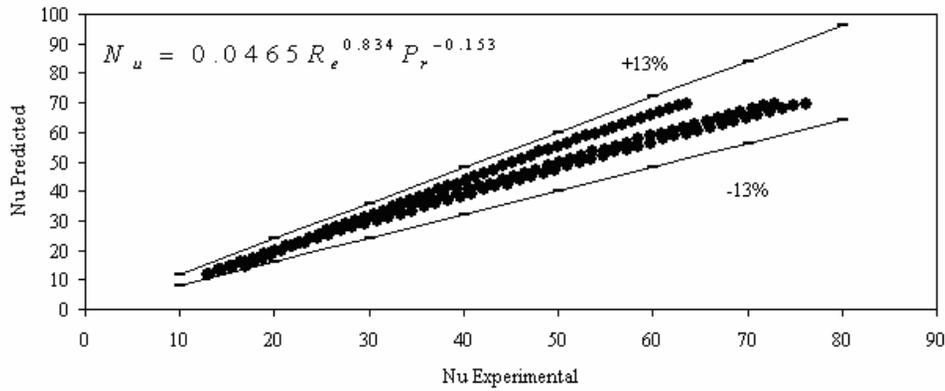


Fig. 4. Variation of Nu (Experimental) with Nu (Predicted).

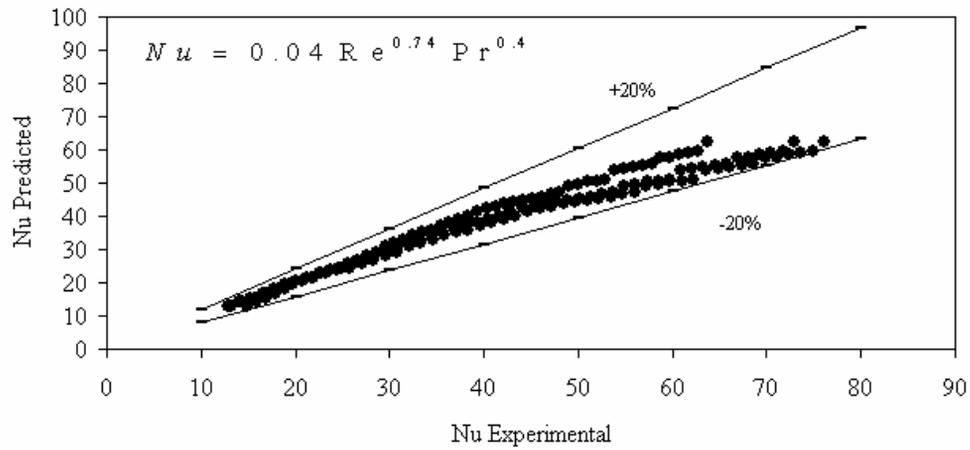


Fig. 5. Holger Martin correlation.

where: t -wall thickness of the spiral plate, m
 k_{ss} -thermal conductivity of the wall.

RESULTS AND DISCUSSION

Fig. 2 shows the variation of Re with heat transfer coefficient of three electrolytes for different inlet temperatures. From the experimental results it is shown that the heat transfer coefficient increases with increasing Re of electrolytes. Solvation of ion with water is the reason to account for maximum heat transfer rate.

Fig. 3 shows the variation of the Reynolds number with Nusselt number of electrolytes for different inlet temperatures. It is clear that the Nusselt number increases with increasing Reynolds number of electrolytes.

Reynolds number vs Nusselt number

Fig. 4 shows the comparisons of the Nusselt numbers obtained from the experiment conducted with those

calculated from theoretically. It can be noted that the experimental and predicted Nusselt numbers fall within ± 13 %. 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 (eq. 3) for spiral plate heat exchanger is expressed as follows. The correlation is obtained by fitting a total of 130 experimental data, ($R^2 = 0.94$).

$$Nu = 0.0465 Re^{0.834} Pr^{-0.153} \tag{3}$$

$$3750 > Re < 8750 \quad 4 < Pr < 6$$

Comparison of experimental Nusselt number with Holger Martin correlation

$$Nu = 0.04 Re^{0.74} Pr^{0.4} \tag{4}$$

$$4.10^2 < Re < 3.10^4$$

Comparisons of the Nusselt numbers obtained from the present experiment with those calculated from the existing correlation are shown in Fig.5. It can be noted that the values obtained from the correlation are slightly consistent with the experimental data and lie within $\pm 20\%$ for the Holger Martin correlation.

CONCLUSIONS

This paper presents new experimental data from the measurement of the heat transfer coefficient of electrolytes flowing 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.

Acknowledgements

The authors are grateful to the Management and the Principal of Kongu Engineering College, Erode, Tamilnadu, India, for granting permission to carry out the research work.

REFERENCES

1. R.A. Seban, E. F. McLaughlin, Heat transfer in tube coils with laminar and turbulent flow, *International Journal of Heat and Mass Transfer*, **6**, 1963, 387-395.
2. G.F.C. Rogers, Y.R. Mayhew, Heat transfer and pressure loss in helically coiled tubes with turbulent flow, *International Journal of Heat and Mass Transfer*, **7**, 1964, 1207-1216.
3. Y. Mori, W. Nakayama, Study on forced convective heat transfer in curved pipes (1st report, laminar region), *International Journal of Heat and Mass Transfer*, **8**, 1965, 67-82.
4. R.R. Berg, C.F. Bonilla, *Trans. N. Y. Acad. Sci.*, **13**, 12, 1950, (as cited in Kubair and Kuloor, 1966).
5. R.K. Jha, M.R. Rao, Heat transfer through coiled tubes in agitated vessels, *International Journal of Heat and Mass Transfer*, **10**, 1967, 395-397.
6. C.E. Kalb, J.D. Seader, Heat and mass transfer phenomena for viscous flow in curved circular tubes, *International Journal of Heat and Mass Transfer*, **15**, 1972, 801-817.
7. C.E. Kalb, J.D. Seader, Fully developed viscous-flow heat transfer in curved circular tubes with uniform wall temperature, *AIChE Journal*, **20**, 2, 1974, 340-346.
8. L.-S. Yao, S.A. Berger, 1978. Flow in heated curved pipes, *Journal of Fluid Mechanics*, **88**, 2, 1978, 339-354.
9. Z. Zapryanov, C. Christov, E. Toshev, Fully developed laminar flow and heat transfer in curved tubes, *International Journal of Heat and Mass Transfer*, **23**, 1980, 873-880.
10. J.B. Lee, H. A. Simon, J. C. F. Chow, Buoyancy in developed laminar curved tube flows. *International Journal of Heat and Mass Transfer*, **28**, 2, 1985, 631- 640.
11. G. Havas, A. Deak, J. Sawinsky, Heat transfer to helical coils in agitated vessels, *The Chemical Engineering Journal*, **35**, 1987, 61-64.
12. N. Acharya, M. Sen, H.C. Chang, Heat transfer enhancement in coiled tubes by chaotic mixing, *International Journal of Heat and Mass Transfer*, **35**, 10, 1992, 2475-2489.
13. N. Acharya, M. Sen, H.C. Chang, Analysis of heat transfer enhancement in coiled-tube heat exchangers, *International Journal of Heat and Mass Transfer*, **44**, 2001, 3189-3199
14. T. Lemenand, H. Peerhossaini, 2002. A thermal model for prediction of the Nusselt number in a pipe with chaotic flow, *Applied Thermal Engineering*, **22**, 2002, 1717-1730.
15. L. Guo, X. Chen, Z. Feng, B. Bai, Transient convective heat transfer in a helical coiled tube with pulsating fully developed turbulent flow, *International Journal of Heat and Mass Transfer*, **41**, 1998, 2867-2875.
16. Y. Inagaki, H. Koiso, H. Takumi, I. Ioka, Y. Miyamoto, Thermal hydraulic study on a high-temperature gas-gas heat exchanger with helically coiled tube bundles, *Nuclear Engineering and Design*, **185**, 1998, 141-151.
17. Holger Martin, *Heat exchangers*, Hemisphere publishing corporation, London, 1992.