Modern Applied Science
www.ccsenet.org/journal.html
Vol. 2, No. 4
July 2008

Analysis of Heat Transfer Enhancement in Spiral Plate Heat Exchanger

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Abstract

In the present study, the heat transfer coefficients of benzene in a spiral plate heat exchanger are investigated. 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 water (Hot fluid) is varying from 0.5 kg sec^{-1} to 0.8 kg sec^{-1} and the mass flow rate of benzene (cold fluid) varies from 0.4 kg sec^{-1} to 0.7 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 which 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

A spiral plate heat exchanger consists of two relatively long strips of sheet metal, normally provided with welded studs for plate spacing, wrapped helically around a split mandrel to form a pair of spiral channels for two fluids. Alternate passage edges are closed. Thus each fluid has a long single passage arranged in a compact package. To complete the exchanger, covers are fitted at each end. It can handle viscous, fouling liquids and slurries more readily because of a single passage. If the passage starts fouling, the localized velocity in the passage increases. The fouling rate then decreases with increased fluid velocity. The fouling rate is very low compared to that of a shell-and-tube unit. It is more amenable to chemical, flush and reversing fluid cleaning techniques because of the single passage. Mechanical cleaning is also possible with removal of the end covers. Thus, maintenance is less than with a shell-and-tube unit. Considerable research is being pursued in spiral and helical heat exchanger in heat transfer and flow areas.

Heat transfer for pulsating flow in a curved pipe was numerically studied by Chung and Hyun (Chung, J. H. and J. M. Hyun., 1994) for strongly curved pipes with substantial pulsation amplitudes. Local Nusselt numbers were developed based on the Womersley number [ratio of transient inertial to viscous forces], which is a function of the pipe radius, the kinematic viscosity, and the frequency of the pulsation. It was found that the strength of the Womersley number affected the distribution of the Nusselt number around the periphery. Dry-out characteristics were studied experimentally by Kaji et al. (Kaji, M., K. Mori, S. Nakanishi, K. Hirabayashi and M. Ohishi., 1995). for a two-phase flow through helical coils. Their work also studied wall temperature fluctuations and compared the results to straight tube experiments. 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. Kubair and Kuloor (Berg, R. R. and C. F. Bonilla., 1950) offered pressure drop and heat transfer for laminar flow of glycerol 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.

Outside-film and inside-film heat transfer coefficients in an agitated vessel were calculated 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. Kalb and Seader (Kalb, C. E. and J. D. Seader., 1972). performed 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. They found that the curvature ratio parameter had insignificant effect on the average Nusselt number for any given Prandtl number. Bai et al (Bai, B., L. Guo, Z. Feng, and X. Chen., 1999) experimentally studied turbulent heat transfer from horizontal helical coils. They concluded that as the Reynolds number is increased, the contribution of secondary flow to the heat transfer diminished and the heat transfer approaches that of a straight tube. This is due to the fact that as the Reynolds number increases the boundary layer becomes smaller. It is the large boundary layer that is shed off into the center of the tube by the secondary flow that increases the heat transfer coefficient, and this effect decreases with increasing Reynolds number Bai et al (Bai, B., L. Guo, Z. Feng, and X. Chen., 1999). The local heat transfer coefficient on the outer wall can be 3 to 4 times that of the inner wall. They developed a correlation of the Nusselt number as a function of the location on the periphery. They also developed a Nusselt number correlation; however it did not contain the Dean number as only one size of coil was used in the experiment. Comparisons for the heat transfer coefficients between straight tubes and helically coiled tubes immersed in a water bath were performed by Prabhanjan et al (Prabhanjan, D. G., G. S. V. Raghavan, and T. J. Rennie., 2002). Findings showed that the heat transfer coefficient were greater in the helically coiled system.

Using a method of fractional steps for a wide range of Dean [10 to 7000] and Prandtl [0.005 to 2000] numbers, laminar flow and heat transfer were studied numerically by Zapryanov et al (Zapryanov, Z., Christov, C. and E. Toshev. ,1980). 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 considered 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. Havas et al (Havas, G., A. Deak, and J. Sawinsky. ,1987) performed the study of the heat transfer to a helical coil in an agitated vessel 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. Acharya et al. (Acharya, N., Sen, M., and H. C. Chang., 1992, Acharya, N., Sen, M., and H. C. Chang., 2001) experimented the heat transfer enhancements due to chaotic particle paths 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.

Guo et al (Guo, L., Chen, X., Feng, Z., and B. Bai., 1998). studied the heat transfer for pulsating flow in a curved pipe 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 correlation for steady turbulent flow for the Reynolds number range of 6000 to 1 80 000.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.

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, however the study of heat transfer and flow characteristics in spiral plate heat exchanger has not received as much of consideration. 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 benzene 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, manometers, 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 benzene 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, 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 Rotometer fitted at the outlet of the corresponding fluids. Table 3.1 shows the experimental conditions.

4. Results and Discussion

4.1 Length from spiral center Vs Heat transfer coefficient

Figure 4.1, 4.2, 4.3 and 4.4. Shows the variation of the length from spiral center and heat transfer coefficient of benzene for different mass flow rates. It is clear that the heat transfer coefficient is varying with mass flow rates. When the mass flow rate is increased the heat transfer coefficient is also increased. On the other hand, the heat transfer coefficient is decreased when the length of spiral plate is increased.

4.2 Length from spiral center Vs Liquid Reynolds number

Figure 4.5 and 4.6 shows the variation of the length from spiral center and Reynolds number of water and benzene for different mass flow rates. It should be noted that the Reynolds number is varying with mass flow rates. When the mass flow rate is increased the Reynolds number is also increased. On the other hand the Reynolds number is decreased when the length of spiral plate is increased.

4.3 Comparison of Nusselt number (Experimental) Vs (predicted)

Figure 4.7 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 ±3%.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 proposed Nusselt number correlation (1) for spiral plate heat exchanger is expressed as follows.

The correlation is obtained by fitting a total of 129 experimental data. (R^2 =0.98)

$$Nu = 0.1868\,\mathrm{Re}^{0.708}\,\mathrm{Pr}^{-0.371} \qquad (1)$$

$6000<Re<11\,000 \qquad 4.7<Pr<5.9$

4.4 Comparison of experimental Nusselt number with Holger Martin correlation

Holger martin correlation (2)

$$Nu = 0.04\,\mathrm{Re}^{0.74}\,\mathrm{Pr}^{0.4} \qquad (2)$$

$4.10^2<Re<3.10^4$

Comparisons of the Nusselt numbers obtained from the present experiment with those calculated from the existing correlation are shown in Figure 4.8. It can be noted that the values obtained from the correlation are slightly consistent with the experimental data and lie within ±14%for the Holger martin correlation.

5. Conclusion

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

6. Acknowdgement

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

7. Nomenclature

Nu- Nusselt number

Re-Reynolds number

Pr-Prandtl number

h- Heat transfer coefficient [wm^{-2} k^{-1}]

U-Over all heat transfer coefficient [wm^{-2} k^{-1}]

Mh-Mass flow rate of hot fluid [Kg Sec^{-1}]

Mc-Mass flow rate of cold fluid [Kg Sec^{-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	0.5-0.8 kg sec^{-1}
4	Mass flow rate of cold Water	0.4-0.7 kg sec^{-1}
5	Reynolds Number of Cold Fluid	6000-11 000

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

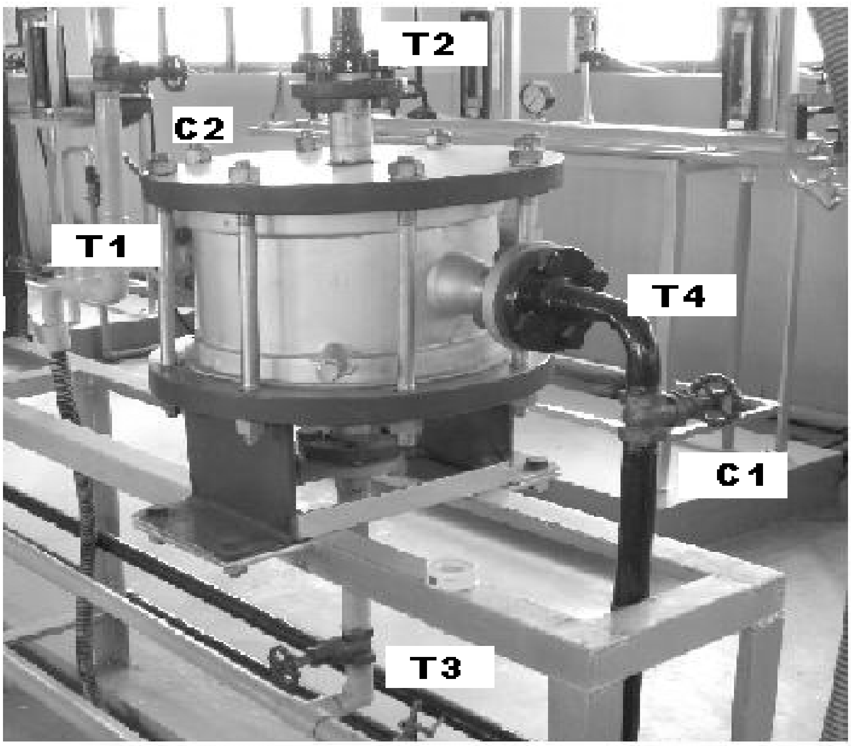

Figure 2.1 Schematic diagram of experimental apparatus C1&C2-Control Valves ,T1,T2,T3&T4-Thermocouples

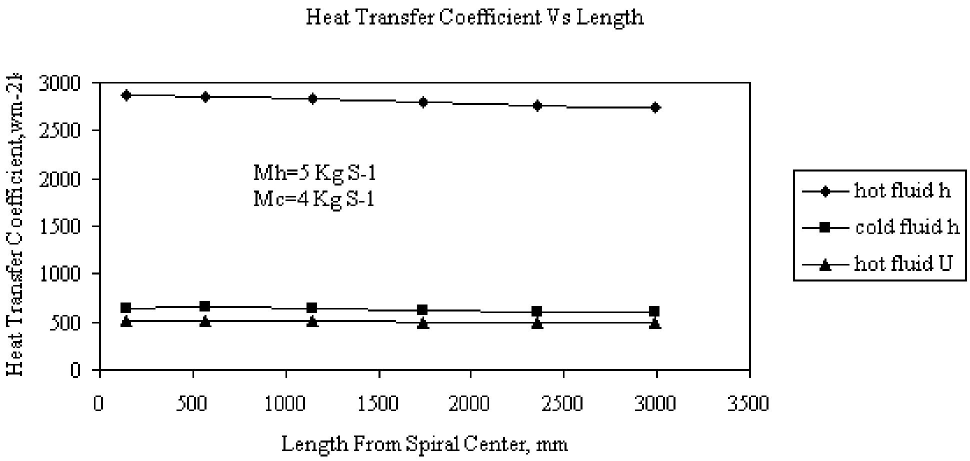

Figure 4.1 Heat transfer coefficient vs length from spiral center

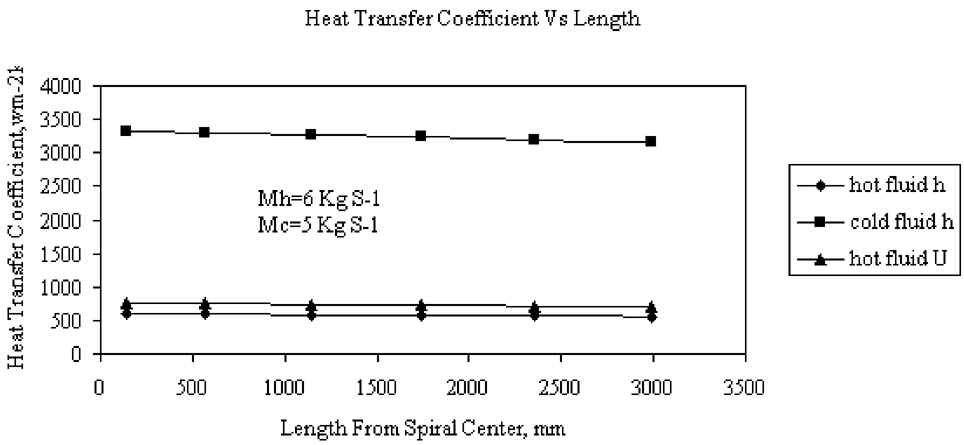

Figure 4.2 Heat transfer coefficient vs length from spiral center

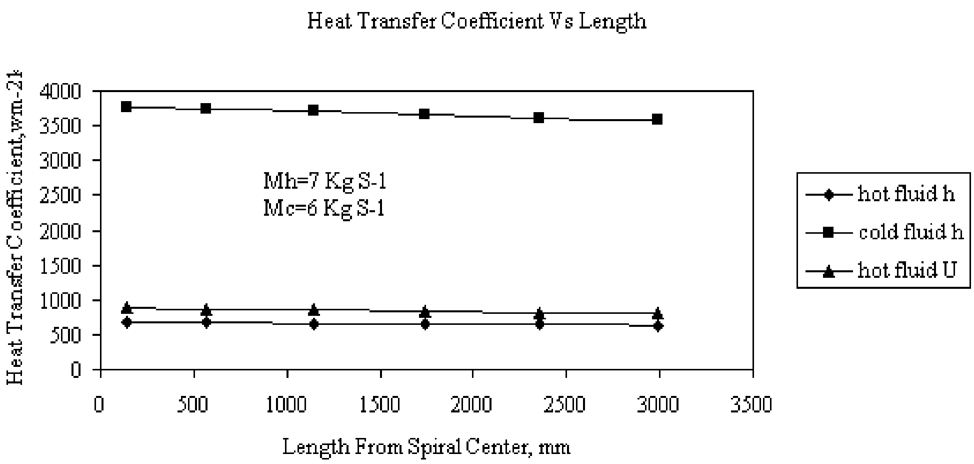

Figure 4.3 Heat transfer coefficient vs length from spiral center

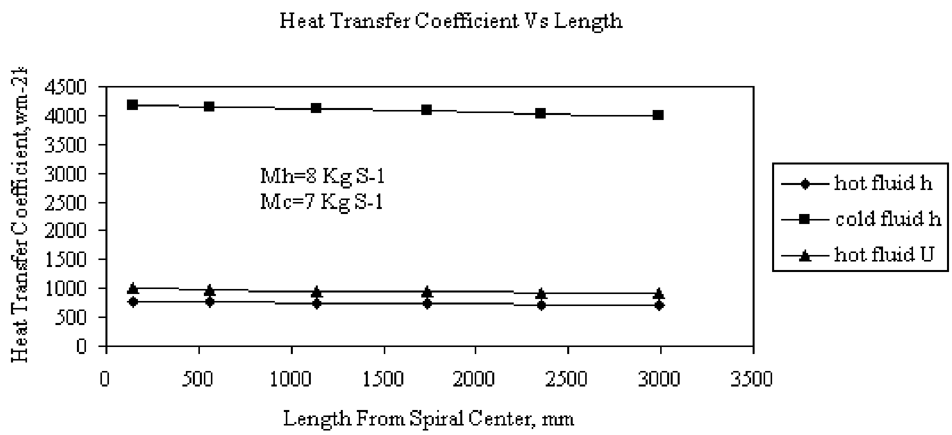

Figure 4.4 Heat transfer coefficient vs length from spiral center

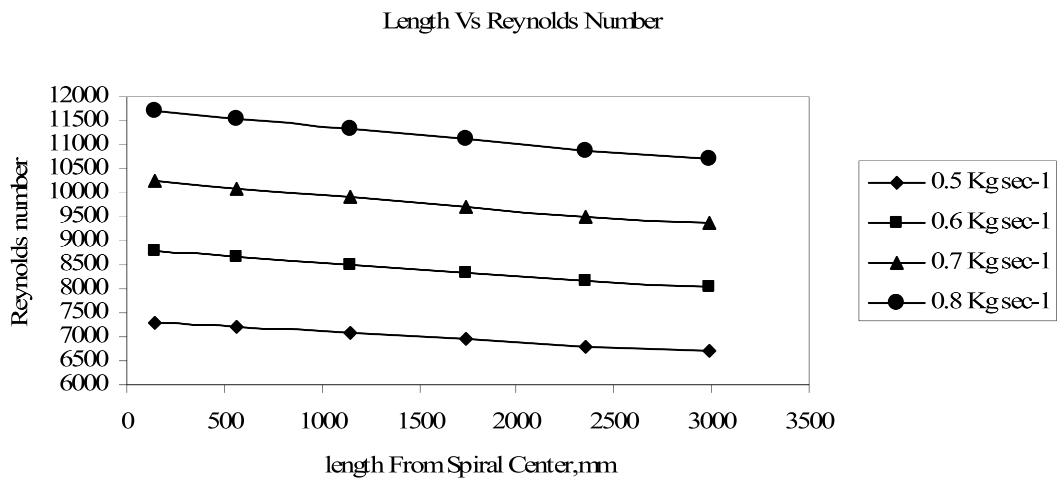

Figure 4.5 Reynolds number (water) vs. length from spiral center

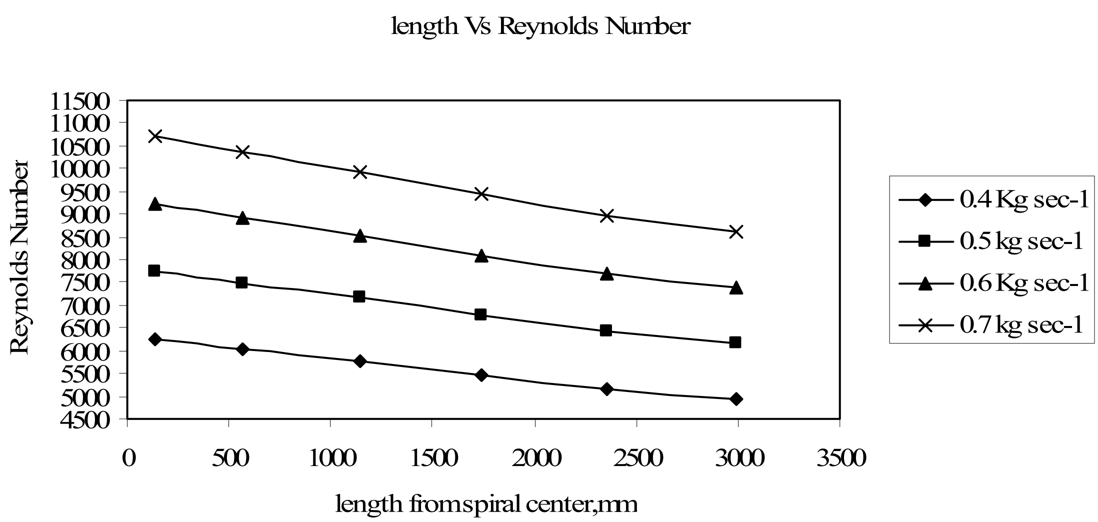

Figure 4.6 Reynolds number (benzene) vs. length from spiral center

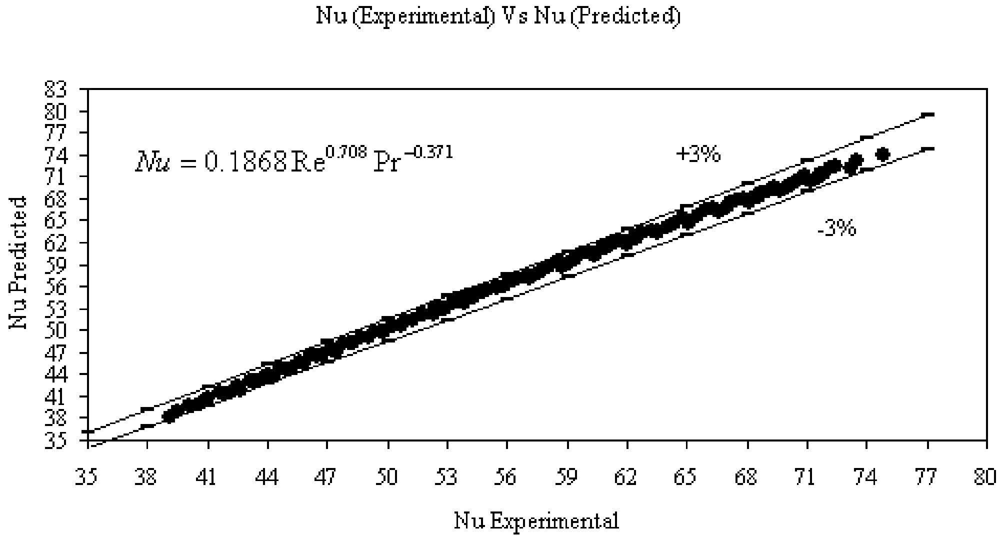

Figure 4.7 Comparison of Nusselt number (Experimental) with Nusselt number (Predicted)

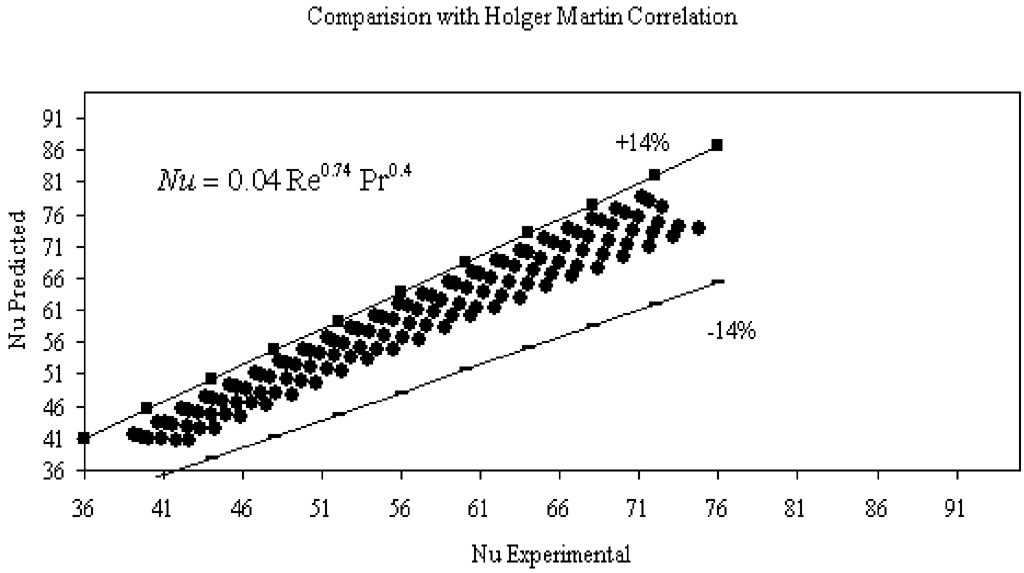

Figure 4.8 Comparison of Experimental data with Holger Martin correlation