International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:4, No:11, 2010

An Experimental Study of the Effect of Coil Step on Heat Transfer Coefficient in Shell-Side of Shell-and-Coil Heat Exchanger

Mofid Gorji Bandpy, Hasan Sajjadi

Abstract**—**In this study the mixed convection heat transfer in a coil-in-shell heat exchanger for various Reynolds numbers and various dimensionless coil pitch was experimentally investigated. The experiments were conducted for both laminar and turbulent flow inside coil and the effects of coil pitch on shell-side heat transfer coefficient of the heat exchanger were studied. The particular difference in this study in comparison with the other similar studies was the boundary conditions for the helical coils. The results indicate that with the increase of coil pitch, shell-side heat transfer coefficient is increased.

Keywords**—**Coil pitch, Shell-and-Coil heat exchanger, Mixed convection, Experimental investigation.

I. INTRODUCTION

A mixed convection heat exchanger is a heat exchanger in which the dominant mode of heat transfer in shell-side flow is mixed convection, as opposed to the traditional heat exchangers with forced convection or natural convection domination on shell-side.

Coiled pipes are used as compact heat exchangers, condensers and evaporators in the food, pharmaceutical, modern energy conversion and power utility systems, heating ventilating and air-conditioning (HVAC) engineering, and chemical industries, and show high heat transfer performance in these applications. The main application of coil-in-shell heat exchanger is in solar domestic hot water (SDHW) systems. The main characteristics of coiled pipes are the compactness and the high heat transfer performance.

Taherian and Allen [1] investigated the natural convection heat transfer on shell-and-coil heat exchanger. In their experiments, the effects of tube diameter, coil diameter, coil surface and shell diameter on the shell-side heat transfer coefficient of shell-and-coil natural convection heat exchanger were studied. They concluded that the Nusselt number was correlated with the Rayleigh number based on the hydraulic diameter of the heat exchanger and the heat-flux on the shell-side. They suggested the following correlation in the range of $6\times10^4 \le Ra_{q,Dhx} \le 2\times10^{10}$

$$Nu_{Dhx} = 0.139\, Ra_{q,Dhx}^{0.293} \tag{1}$$

They also gave a correlation based on modified Rayleigh number by multiplying Rayleigh number by H/L. The following equation shows the correlation in the range of $2\times10^3 < Ra^*_{Dhx} < 2\times10^7$

$$Nu_{Dhx} = 0.182\, {Ra^*_{Dhx}}^{0.394} \tag{2}$$

Rogres and Mayhew [2] concentrated their attention on heat transfer and pressure loss in helically coiled tubes with turbulent flow. Three coils having mean diameters of 10.2, 12.5 and 190 mm, were made of 9.45 mm ID copper tubes. The coils were heated by steam at slightly above atmospheric pressure. The heat transfer data resulted in the empirical equation (3), for $10^4 \le \mathrm{Re} \le 10^5$ which was considered turbulent.

$$Nu = 0.023\,\mathrm{Re}^{0.85}\left(d/D_{c,m} \right)^{0.1} \mathrm{Pr}^{0.4} \tag{3}$$

Ali [3] studied natural convection heat transfer from helical coils immersed in a large water tank. Two different tube diameters, 8 mm and 12 mm OD, with five coil diameters and up to five different pitches of 1.5 to 4 with 5 or 10 turns were tested. Equations (4) and (5) were presented for the Nusselt number data based on the coil length, for 12 and 8 mm OD tubes respectively.

$$Nu_L = 0.685 (Ra_L)^{0.295} \qquad (d_o = 12\text{mm}) \tag{4}$$

$$Nu_L = 0.00044 (Ra_L)^{0.516} \qquad (d_o = 8\text{mm}) \tag{5}$$

Based on the exponents of the Rayleigh numbers in Equations. (4) and (5), the heat transfer coefficient for the 12 mm OD tube decreases with length enhancement, while for the 8 mm OD tube, an increase in length of the coil will increase the heat transfer coefficient considerably. No explanations were provided for this unexpected behavior.

Prabhanjan et al. [4] presented results of an experimental investigation of natural convection heat transfer from helical coiled tubes in water. They used different characteristic

F. A. Department of Mechanical Engineering, Babol Noshirvani University of Technology, Babol, Iran (corresponding author to provide phone: 00981113224048; fax: 00981112296582; e-mail: gorji@nit.ac.ir).
S. B. Department of Mechanical Engineering, Babol Noshirvani University of Technology, Babol, Iran (e-mail: hasansajadi@gmail.com).

lengths to correlate the outside Reynolds numbers to the Rayleigh number. They considered the coil height as the best representation for a vertical coil and their prediction procedure shows a promise as a method of predicting the outlet temperature from a coil given the inlet temperature, both temperature and coil dimensions.

Manlapaz and Churchill [5] also worked on fully developed laminar convection in helical coil. By reviewing and employing previously published work of other authors, a new correlation was developed. For the case of axially uniform heat flux with peripherally uniform wall temperature, the following equation was proposed.

$$Nu = \left[\left(\frac{48}{11}+\frac{51/11}{1+\frac{1342}{\Pr He^2}}\right)^3 + 1.816\left(\frac{He}{1+\frac{1.15}{\Pr}}\right)^{3/2}\right]^{1/3} \quad (6)$$

That equation should not be used for $He > 2000$.

In this study the flow inside the coiled tube was both laminar and turbulent for different mass flow rates. For the laminar and turbulent flow equations (6) and (7) were used respectively.

$$Nu_i = 1+3.6\left(1-\frac{D_t}{D_c}\right)\left(\frac{D_t}{D_c}\right)^{0.8}\left[0.0023\mathrm{Re}_i^{0.8}\Pr_i^{0.4}\right] \quad (7)$$

However there are many works in tube-side of Shell-and-Coil heat exchanger correlated to heat transfer coefficient and on natural convection on shell-side, but there are not many investigations on forced and mixed convection.

Xin and Ebadian [6] studied shell-and-coil natural convection heat exchangers experimentally. They proposed several correlations for Nusselt numbers versus Rayliegh numbers based on different characteristic length.

Ajele [7] studied shell and coil natural convection heat exchangers experimentally. Combinations of up to four coils, as well as, single coil were tested in a 100 mm inner diameter shell. A correlation was proposed for multiple coi1 tests of shell-and-coil natural convection heat exchangers.

Numerical investigations were conducted to understand forced laminar fluid flow over coiled pipes with circular cross-section by Conte et al. [8]. The focus was addressed on exploring the convective heat transfer from conical and helical coils with comparative studies.

The same numerical investigation method was applied to the two differentially coiled pipes (helical and conical) and for different Reynolds numbers corresponding to five cases of exterior flow arrangement. The results show better heat transfer performance for cases of conical coils where much flow turbulence was observed due to an effective flow arrangement.

II. EXPERIMENTALAPPARATUS AND TEST SECTION

Figure (1) shows the apparatus arranged for heat exchanger experiments. Water was used as the hot and cold fluid whereas hot water was pumped to the tank and coil passing through six electric heaters. A valve is installed at the inlet of the heat exchanger to control the flow rate and cold water in the shell side was taken from urban water. The flow rate was measured by using a calibrated measuring cylinder and a stopwatch positioned at the outlet of heat exchanger. The temperature of the inlet water of coiled tube to the heat exchanger was controlled by thermostat. Four constant temperatures (50, 60, 70 and 80 degrees Celsius) were considered for inlet mass flow rate of coil and the temperature of shell side inlet was also temperature of the tap water. These temperatures are in accordance with the outlet temperature of a flat plate solar collector.

Fig. 1. Apparatus for heat exchanger experiments

The coil was formed carefully by using 9.52 and 12.5 mm OD straight copper tubing. Care was taken to locate the coil at the middle of the two shells. Temperatures were measured by four K-type thermocouples placed at equally distanced locations in order to measure the coil surface and the fluid temperature. Four other thermocouples were located at inlets and outlets of heat exchanger to measure the temperatures of the hot and cold fluids.

A data acquisition device made by Advantech model USB 4718 having a capacity of 8 analog input channels in connection with a PC was used to record all temperature measurements. All tests were performed under steady state conditions. A Visual Basic code was written to retrieve and store temperature data and to perform calculations. The data acquisition system stored data every 5 seconds. The measured values were averaged over a period of 4 minutes.

III. HEAT TRANSFER

For determining the heat flux rate, we assumed that the thermal resistance of the copper tube wall was negligible. A temperature of coil surfaces was taken as equal to the water temperature inside the coil at the same location in order to calculate local heat flux. The values of heat transfer rate and h0, were calculated by using equations. (8) to (11). All these properties were averaged over inlet and outlet of the fluid in each side. The values of Reynolds, Rayleigh and Nusselt numbers will be obtained from equations. (12), (13) and (14) respectively.

$$Q_h = \dot{m}_c Cp_c \left(T_{h,i} - T_{h,o}\right) \tag{8}$$

$$LMTD = \frac{\left(T_{h,i} - T_{c,o}\right) - \left(T_{h,o} - T_{c,i}\right)}{\ln\left(\dfrac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right)} \tag{9}$$

$$UA = \dot{m}_c Cp_c \left(Th_i - Th_o\right) / \text{LMTD} \tag{10}$$

$$\frac{1}{h_0} = \frac{1}{U} + \frac{1}{h_i} \tag{11}$$

$$Re_s = V_s L / v \tag{12}$$

$$Ra_s = g\beta\Delta TL^3 / \alpha v \tag{13}$$

$$Nu = h_o d / k \tag{14}$$

IV. RESULTS

In this research the influence of the coil step, as a geometric parameter of heat transfer coefficient of the shell is investigated. In Figure 2 shell side heat transfer coefficient in relation to heat flux rate for a fixed pipe diameter is drawn in two different steps.

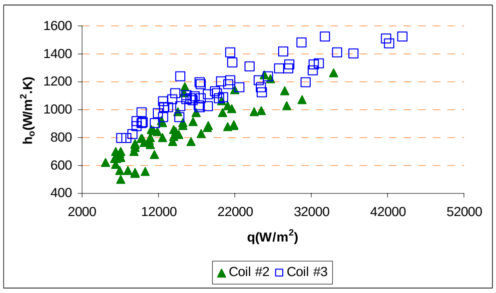

Fig. 2. Heat transfer coefficient in relation to heat flux rate for pipe diameter 12.5 mm

According to the figure.2, it is clear that with the increase in coil step and fixed tube diameter, the increase in h0 is resulted and it is concluded that in a particular heat flux rate and fixed tube diameter, with the increase in coil step, a higher heat transfer coefficient can be obtained. For more investigation of the effect of coil step, shell-side heat transfer coefficient, is plotted in figures 3 to 18, in relation to different flows and temperatures.

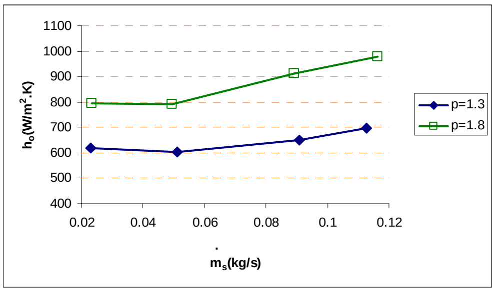

Fig. 3. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 50^\circ C, \dot{m}_c = 0.024\, kg/s$

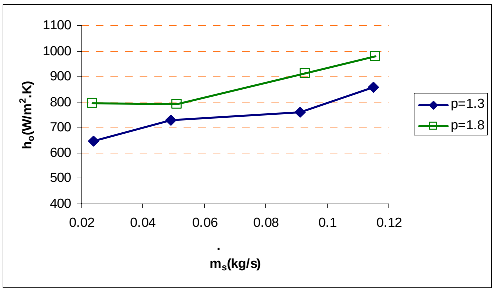

Fig. 4. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 50^\circ C, \dot{m}_c = 0.05\, kg/s$

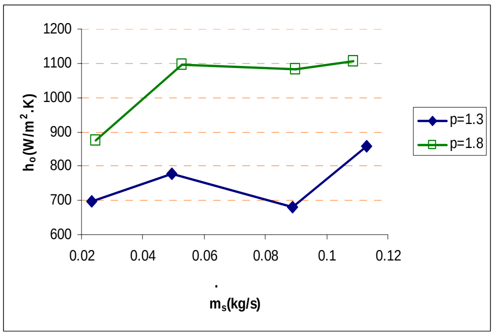

Fig. 5. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 50^\circ C, \dot{m}_c = 0.09\, kg/s$

International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:4, No:11, 2010

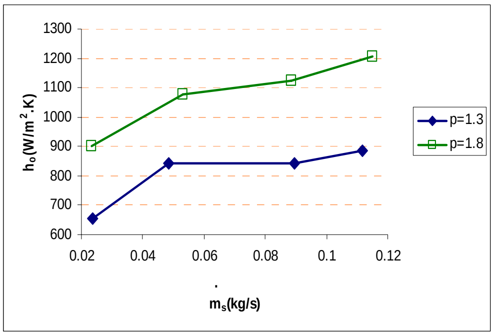

Fig. 6. Shell side heat transfer coefficient in relation to the crust

Flow $T_{h,i} = 50^\circ C, \dot{m}_c = 0.113 \, {}^{kg}\!/_{s}$

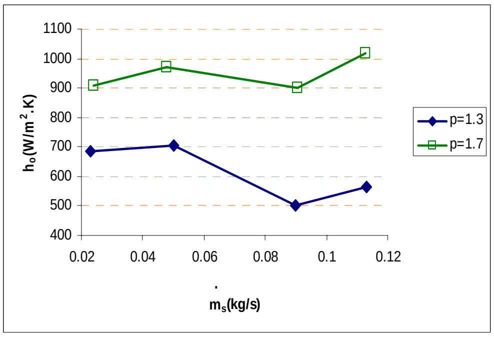

Fig. 7. Shell side heat transfer coefficient in relation to the crust

Flow $T_{h,i} = 60^\circ C, \dot{m}_c = 0.024 \, {}^{kg}\!/_{s}$

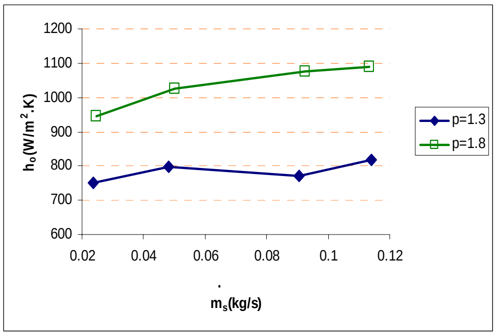

Fig. 8. Shell side heat transfer coefficient in relation to the crust

Flow $T_{h,i} = 60^\circ C, \dot{m}_c = 0.05 \, {}^{kg}\!/_{s}$

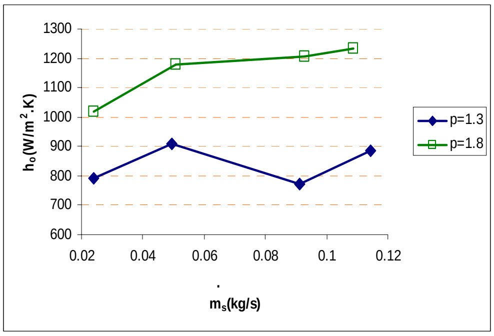

Fig. 9. Shell side heat transfer coefficient in relation to the crust

Flow $T_{h,i} = 60^\circ C, \dot{m}_c = 0.09 \, {}^{kg}\!/_{s}$

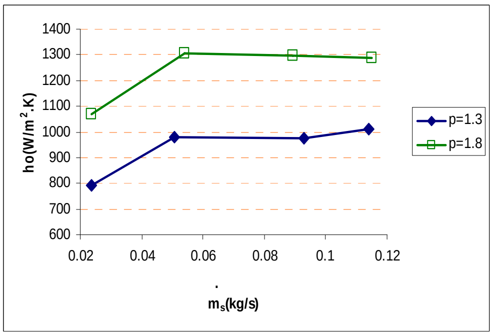

Fig. 10. Shell side heat transfer coefficient in relation to the crust

Flow $T_{h,i} = 60^\circ C, \dot{m}_c = 0.113 \, {}^{kg}\!/_{s}$

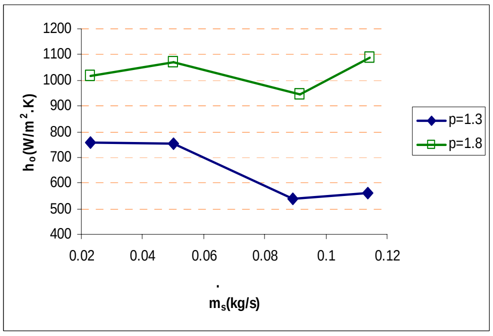

Fig. 11. Shell side heat transfer coefficient in relation to the crust

Flow $T_{h,i} = 70^\circ C, \dot{m}_c = 0.024 \, {}^{kg}\!/_{s}$

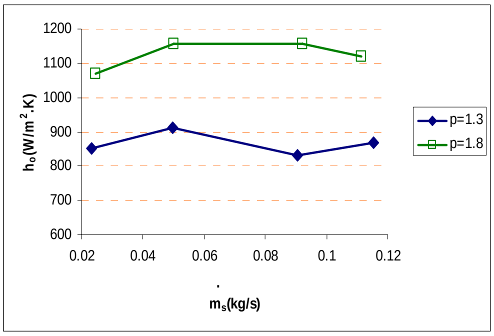

Fig. 12. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 70^\circ C, \dot{m}_c = 0.05 \, {}^{kg}\!/_{s}$

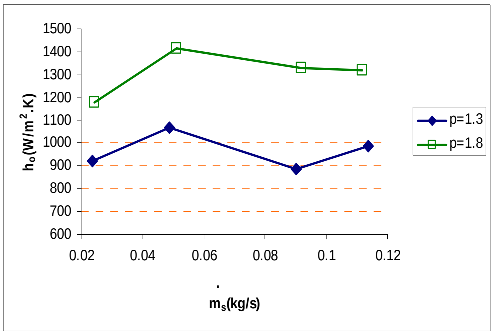

Fig. 13. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 70^\circ C, \dot{m}_c = 0.09 \, {}^{kg}\!/_{s}$

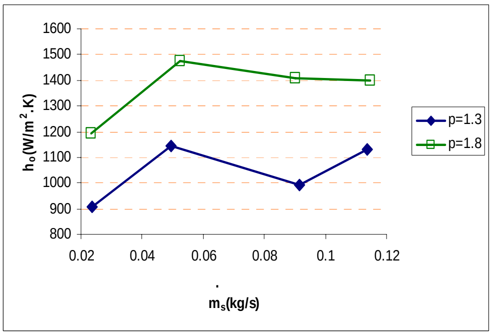

Fig. 14. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 70^\circ C, \dot{m}_c = 0.113 \, {}^{kg}\!/_{s}$

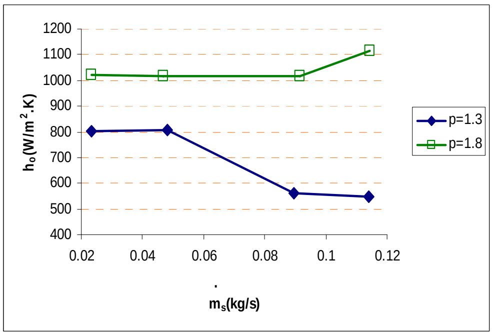

Fig. 15. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 80^\circ C, \dot{m}_c = 0.024 \, {}^{kg}\!/_{s}$

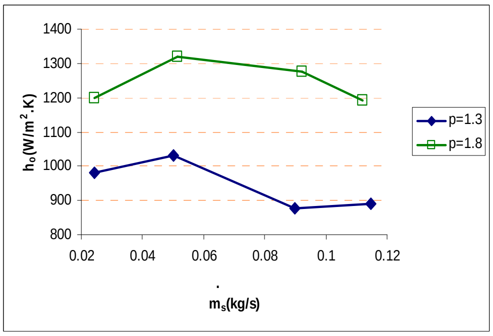

Fig. 16. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 80^\circ C, \dot{m}_c = 0.05 \, {}^{kg}\!/_{s}$

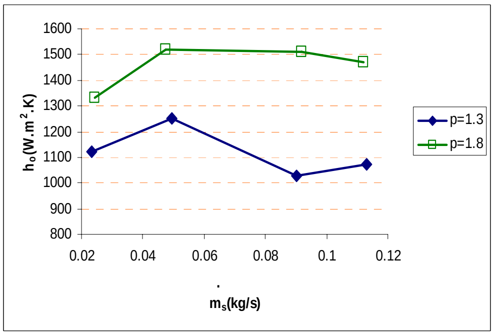

Fig. 17. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 80^\circ C, \dot{m}_c = 0.09 \, {}^{kg}\!/_{s}$

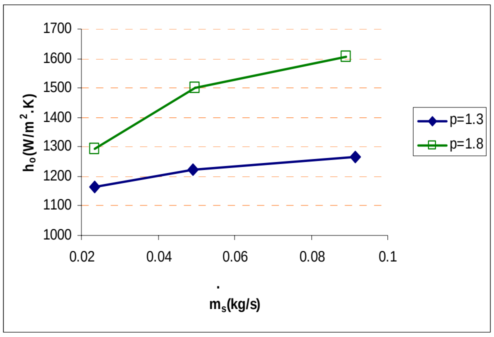

Fig. 18. Shell side heat transfer coefficient in relation to the crust Flow $T_{h,i} = 80^{\circ} C, \dot{m}_c = 0.113 \, {}^{kg}\!/_{s}$

As it is observed in figures 3 to 18, heat transfer coefficient for coil inlet temperature, increases with the enhancement of coil step from 1.3 to 1.8 inch in a fixed tube diameter. All these figures, approve of the results of figure 2.

V. Conclusion

In this paper, an experimental study on the effect of coil step changes, in shell spiral coil heat exchanger for heat transfer coefficient, on the shell-side was performed. Forced convection heat transfer from spiral coil, was done in different Reynolds numbers and various dimensionless steps. Four various flow amounts for shell and coil and also four different inlet temperatures were considered and the effect of coil step on shell-side heat transfer in different mass flow rates was investigated. Results represent that with the enhancement of coil step in a fixed tube diameter, shell-side heat transfer coefficient increases.

References

[1] H. Taherian, Peter L. Allen, Experimental study of natural convection shell-and-coil heat exchanger, The American Society of Mechanical Engineers, HTD-Vol. 357-2, ASME Proceeding of the 7th . AIAA/ASME, 1998.
[2] G. F. C. Rogrers, Mayhew, Y. R., Heat transfer and pressure loss in helically coiled tubes with turbulent flow, International Journal of Heat and Mass Transfer, Vol. 7, pp.1207-1216, 1964.
[3] M. E. Ali, Experimental Investigations of Natural Convection from Vertical Helical Coiled Tubes, Int. J. Heat Mass Transfer, Vol. 37, pp. 665–671, 1994.
[4] D. G. Prabhanjan, T. J. Rennie, and G. S. V. Raghavan, Natural Convection Heat Transfer from Helical Coiled Tubes, Int. J. Thermal Sciences, Vol. 43, pp. 359– 365, 2004.
[5] R. L. Manlapaz, Churchill, S. W. , Fully developed laminar convection from a helically coil, Chemical Engineering Communication, Vol. 9, pp. 185-200, 1981.
[6] R. C. Xin, M. A. Ebadian, Natural convection Heat Transfer from Helicoidal pipes, Journal of Thermophysics and Heat Transfer, Vol. 10, No. 2, pp. 297- 302. 1996.
[7] O. J. Ajele, "Natural convection heat transfer from enclosed helical coils, Ph.D. Thesis, Technical university of Nova Scotia, 1995.
[8] I. Conté, X. F. Peng, B. X. Wang, Numerical Investigation of Forced Fluid Flow and Heat Transfer from Conically Coiled Pipes, Numerical Heat Transfer, Part A: Applications, 53:9, 945 – 965, 2008.