Al-Khwarizmi Engineering Journal, Vol. 14, No. 1, March, (2018)
P.P. 10-18

Al-Khwarizmi Engineering Journal

Experimental Investigation of the Effect of Curvature Ratio on Heat Transfer in Double Pipe Helical Heat Exchanger

Hisham Assi Hoshi
Department of Electromechanical Engineering/ University of Technology/ Baghdad/ Iraq.
Email: hah_62@yahoo.com

(Received 27 April 2017; accepted 3 August 2017)
https://doi.org/10.22153/kej.2018.08.001

Abstract

Different parameters of double pipe helical coil were investigation experimentally. Four coils were used; three with a curvature ratio (0.037, 0.031, and 0.028) and 11mm diameter of the inner tube while the fourth with 0.033 curvature ratio and 13 mm diameter of the inner tube. The hot water flow in the inner tube whereas the cold water flows in the annulus. The inlet temperatures of hot and cold water are 50 ^{0}C and 18 ^{0}C respectively. The inner mass flow rate ranges from 0.0167 to 0.0583 kg/s. The results show the Nusselt number increase with increase curvature ratio. The Nusselt number of the coil with 0.037 curvature ratio increases by approximately 12.3 % as compare with 0.028 curvature ratio. The results also reveal that the Nusselt number of the coil with curvature ratio 0.033 increases by approximately 11.6 % as compare with 0.028 curvature ratio for 400 mm coil diameter.

Keywords: *Curvature ratio, dean number, heat transfer, helical coil.*

1. Introduction

One of most favorable passive techniques is helical coiled. The secondary flow motion induced due to centrifugal force resulting from the curvature of the tube makes the heat transfer rate greater than in a straight tube [1, 2, and 3]. The heat transfer and flow characteristic in curved tubes such as spiral and helical coiled tubes have been investigated. Jamal-Abad et al [4] have studied experimental investigation on heat transfer performance and pressure drop characteristics in spiral coil. They found that the geometry of spiral coil is much effective in enhancing the thermal observed. Also annulus Nusselt number values increases with increase in the annulus flow rate. The heat transfer characteristic in helical coiled tubes with different design parameter such as tube diameter, coil diameter and coil pitch have rarely been investigated experimentally and numerically [10, 11, 12 and13]. Mohammed et al [14] performed a numerical study to investigate the performance. Kubir and Kuloor [5, 6] found that friction factor in spiral coil greater than that in straight tubes. Rennie [7] and Rennie and Raghavan [8] experimentally studied the heat transfer in double-pipe helical heat exchanger comprised of one loop. They showed that the overall heat transfer coefficient increase with increase in tube Dean number or annulus Dean numbers. Kumer et al [9] numerically and experimentally investigated the heat transfer in double-pipe helical heat exchanger comprised of four turns of coils. They found that the numerical results of friction factor agree fairly well with the experimental data and an error of 5-8% was heat transfer and fluid flow characteristics in a helical coiled tube heat exchanger using finite volume method. The study was carried out with different geometrical parameter. The results show that the geometrical parameters such as the helix radius and inner tube diameter have significant effect on the heat transfer under laminar flow conditions. Akhavan- Behabadi et al [15] investigated experimentally the heat transfer characteristics of helically coiled tubes. The experiments were carried out for different geometrical parameter. It was concluded that the coils with greater curvature ratio had higher performance. Pawar et al [16] study experimentally the heat transfer characteristics in a helically coiled heat exchanger for Newtonian as well as non-Newtonian fluids. The experiments were performed for different curvature ratio. They found that overall heat transfer coefficient is higher for smaller helix diameter as compared to larger helix diameter.

In the present work four coils were considered; three coils with a curvature ratio (0.037, 0.031, and 0.028) and 11mm diameter of the inner tube while the fourth with 0.033 curvature ratio and 13 mm diameter of the inner tube and all with same coil length. The purpose of this work is to study the effect of curvature ratios and inner tube diameter on heat transfer and pressure drop characteristics in a helical double pipe heat exchanger and compare the result with previous work.

2. Experimental Set Up and Procedure

2.1. Material

The test section is a double pipe helical coil it was made of copper. Four coils with different coil diameter (D_c) were considered, each coil having 1260 mm long but the coil diameter were changed as shown in the figure (1). The inner diameter of outer pipe (D_o) and inner diameter of inner pipe (d_i) are 22 mm and 11 mm respectively for three coils and the fourth with 13 mm inner diameter of inner pipe (d_i) and all with 1 mm thickness. The geometry dimensions were putting in table (1).

2.2. Experimental Apparatus

The experimental set up mainly consist of a test section, hot water closed loop, cold water open loop as shown in the figure (2) and figure (3). The water is heated in hot tank by 3000 W thermostatic electrical heater then pumped to the inner helically coiled tube of test section for circulation, then returns to the hot tank. The cold water was driven by pump from the cold water tank to flow through annulus as counter flow configuration. Two flow meter with rang (1-10 lit/min) with 0.2 % accuracy were used to measure the flow rate of hot and cold fluid .The flow rates were controlled by ball valve. The pressure drop through the inner tube was measured with mercury U-tube manometer. Four T-type thermocouples with ±0.1 °C accuracy were inserted at inlet and outlet of the test section and connected to a digital temperature recorder to measure the cold and hot fluid temperature. The experimental were conducted by changing for six different flow rates in the inner tube from (1-3.5 lit/min) with annulus cold water flow rate was kept constant at 2 lit/min. The inlet hot water temperature was set to (50 ± 1 °C) whereas the inlet cold water was set to (18 ± 0.5 °C). The test section is isolated by rubber tube to minimize the heat losses with surrounding.

2.3. Experimental Procedure

In the beginning, the heater is switch on to warm the water inside the hot tank and then the two pumps are turned on to allow hot and cold water to pass through the test section. The flow is adjusted through a control ball valve until flows reach an accurate value. Then, until the system reaches a state of stability, we begin to take the readings of the inlet and outlet temperature of the water in the inner and annulus of test section. The measurement readings are repeated to make sure they are correct. Six values of flow rate were considered from 1 lit/m to 3.5 lit/m and the experiments were repeated.

2.4. Data Reduction

The heat transfer rate is averaged between the heat supplied by inner hot water and the heat absorbed by the annulus cold water

$$Q\,avg = \left[\frac{\dot{m}_h Cp_h(T_{h,i}-T_{h,o})+\dot{m}_c Cp_c(T_{c,i}-T_{c,o})}{2}\right] \quad ...(1)$$

The overall heat transfer coefficient (U) is calculated as follow [17]:

$$U = \frac{q}{A\,LMTD} \quad ...(2)$$

where LMTD is

$$\Delta T_{LMTD} = \frac{(T_{h,i}-T_{c,o})-(T_{h,o}-T_{c,i})}{\ln\frac{(T_{h,i}-T_{c,o})}{(T_{h,o}-T_{c,i})}} \quad ...(3)$$

The inner side heat transfer coefficient (hi) is determined by neglecting the conduction thermal resistance of tube wall through.

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \quad ...(4)$$

Critical Reynolds number is given as [18]

$$Re_{critical} = 20000\left(\frac{d_i}{D_c}\right)^{0.32} \quad ...(5)$$

The outside heat transfer coefficient (ho) was estimated by correlation of Rennie for laminar flow [7].

$$Nu_o = \frac{h_o D_h}{k} = 0.075De + 5.36 \quad ...(6)$$

Where the annulus Dean number De is

$$D_e = \frac{\rho V_o}{\mu}\left(\frac{D_o^2 - D_i^2}{D_o + D_i}\right)\left(\frac{D_o - D_i}{R}\right)^{1/2} \quad \ldots(7)$$

where D_o is the inner diameter of outer pipe, D_i is the outer diameter of inner pipe and R is the radius of coil.

The hydraulic diameter D_h is

$$D_h = D_o + D_i \quad \ldots(8)$$

Thus, $Nu_i = \frac{D_i h_i}{k}$ ...(9)

Where k is calculated by using mean bulk fluid temperature (T_b).

$$T_b = \frac{T_i + T_o}{2} \quad \ldots(10)$$

Table1,
Geometrical dimensions of double pipe helical coils.

Helical coil	D_o(mm)	d_i (mm)	L (mm)	D_c (mm)	$\delta=\frac{d_i}{D_c}$	N	pitch (mm)
Coil- I	22	11	1260	300	0.037	1.336	40
Coil- II	22	11	1260	350	0.031	1.146	40
Coil- III	22	11	1260	400	0.028	1.003	40
Coil- v	22	13	1260	400	0.033	1.003	40

Fig. 1. Helical coils with different coil diameters.

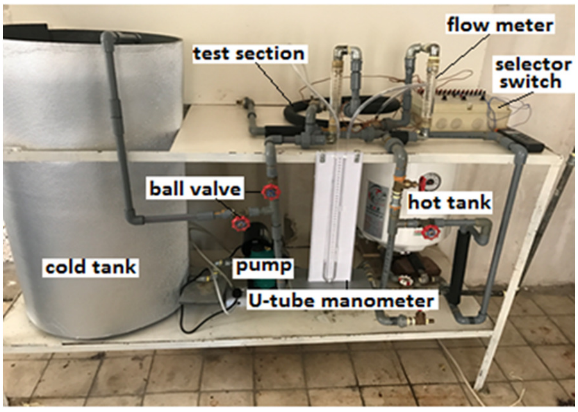

Fig. 2. Experimental Rig.

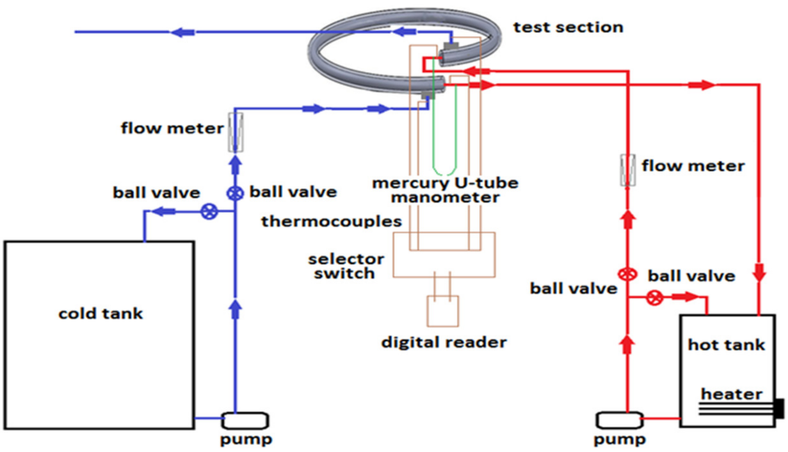

Fig. 3. Schematic of experimental Rig.

3. Results and Discussion

Fig (4) and Fig (5) show the variation of pressure drop with inner mass flow rate and with different curvature ratio. The pressure drop increase with increasing mass flow rate because of increasing flow rate led to increase the turbulent between the fluid elements. Also when the flow increases, the flow becomes more turbulent, resulting in the loss of a large amount of energy due to the conversion of the energy of the pressure into kinetic energy. Also pressure drop increasing with increase curvature ratio because secondary flow intensifies in higher curvature ratio due to centrifugal force that led to increase the shear stress means that pressure drop will increase. The figure (5) shows at bigger diameter pressure drop reduces because pressure drops inversely proportion to the diameter tube and directly proportion to the velocity of flow which is responsible for lower pressure drop.

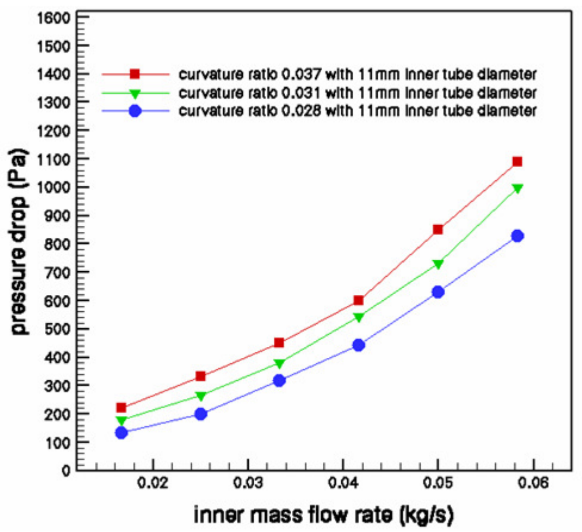

Fig. 4. Variation of pressure drop with inner mass flow rate.

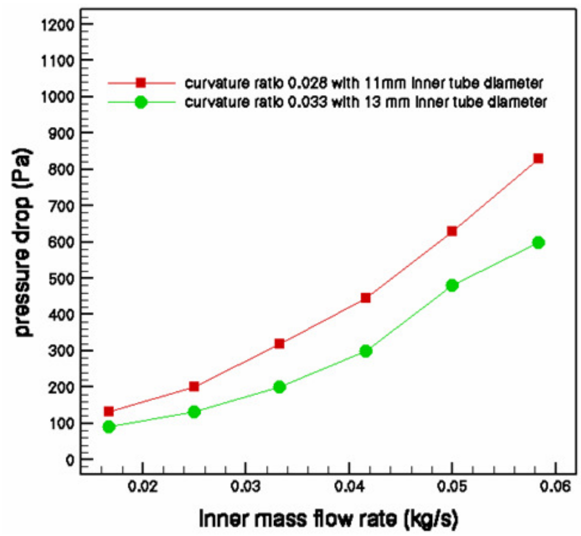

Fig. 5. Variation of pressure drop with inner mass flow rate for 400 mm coil diameter

Fig (6) and Fig (7) show the variation of overall heat transfer coefficient with mass flow rate and curvature ratio. The result show that increase overall heat transfer coefficient by increasing mass flow rate this can be attributed increasing the velocity of fluid flow that makes increasing of heat transfer coefficient. Also overall heat transfer coefficient increase with increase curvature ratio because the secondary flow is more effective due to centrifugal force at higher curvature ratio. But when increase the coil diameter the turbulence due to the effect of curvature ratio disappear and the fluid flow through coiled tube changed to become such as the flow through straight tube which in turn decrease overall heat transfer.

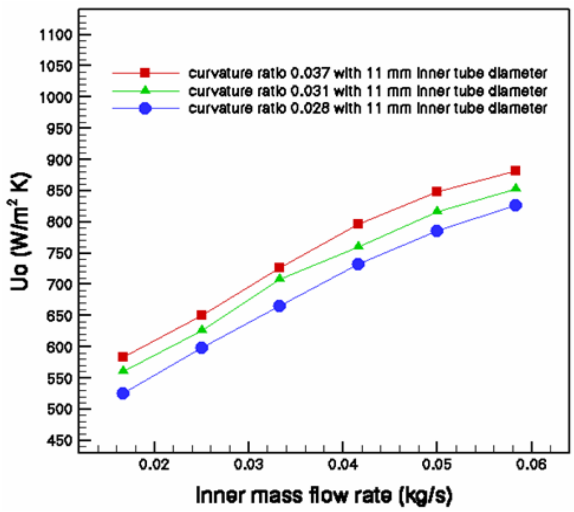

Fig. 6. Variation overall heat transfer with inner mass flow rate.

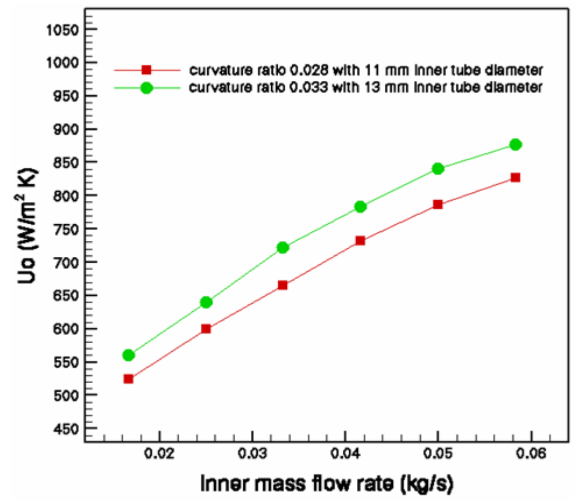

Fig. 7. Variation of overall heat transfer coefficient with inner mass flow rate for 400 mm coil diameter.

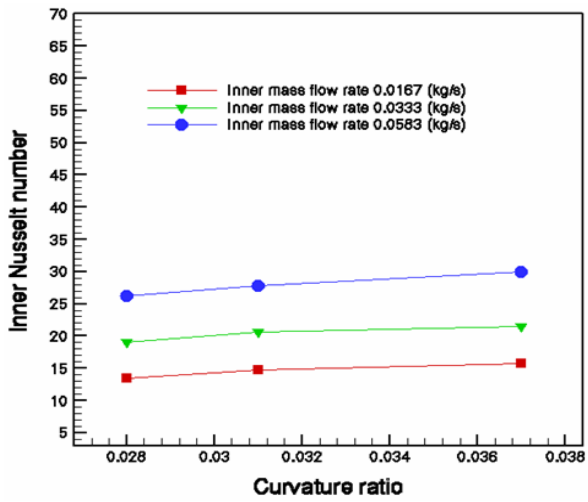

Fig. 9. Variation of Nusselt number with curvature ratio.

Fig (8) and Fig (9) depicted the variation of Nussult number with mass flow rate and with different curvature ratio. The results show the Nussult number increasing with increase of mass flow rate due to increase the turbulent between the fluid elements which will enhance the rate of heat transfer. Also show Nussult number increase with increase of curvature ratio because at the high curvature ratio increase the centrifugal force, which in turn led to increase the effect of secondary flow that increase the rate of heat transfer and as a result increase the Nussult number. Also Secondary flow is more effective due to centrifugal force, which increases at higher mass flow rate. The Nusselt number of the coil with curvature ratio 0.037 increases by approximately 12.3 % as compare with 0.028 curvature ratio.

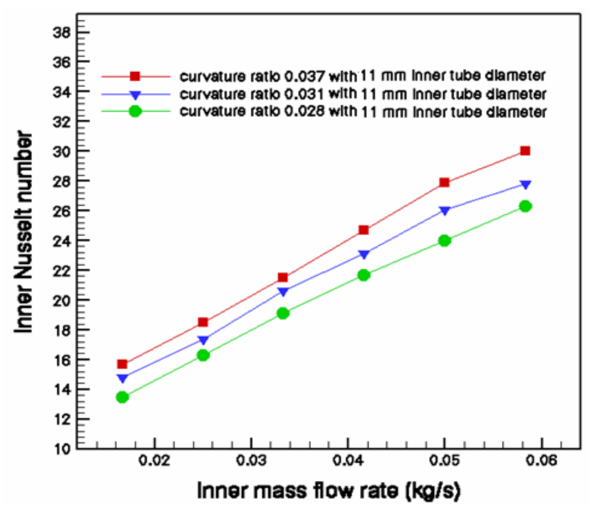

Fig. 8. Variation of Nusselt number with inner mass flow rate

From the figure (10), it was found that Nussult number increase with increase diameter of tube, as known that the increasing diameter of tube led to increase of curvature ratio. This leads to further improved heat transfer and then increases the Nusselt number because when the diameter of the tube is increased, the secondary flows will be stronger, and then more confused. The Nusselt number of the coil with curvature ratio 0.033 increases by approximately 11.6 % as compare with 0.028 curvature ratio for 400 mm coil diameter, or in other words as a comparison between inner tube diameter of 13 mm and 11 mm for the same coil diameter.

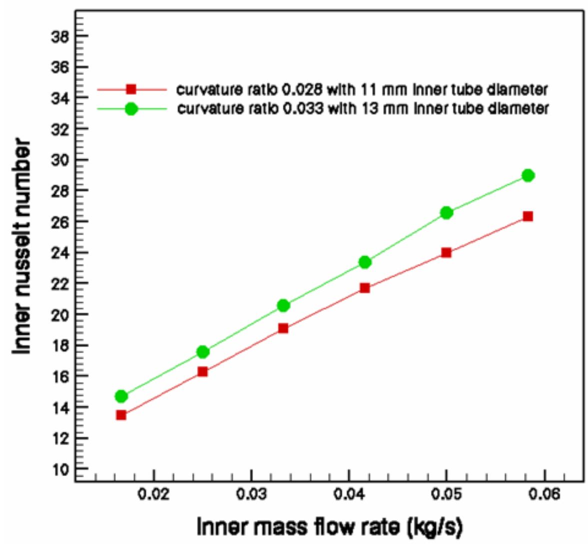

Fig. 10. Variation of Nusselt number with inner mass flow rate for 400 mm coil diameter.

Fig (11) shows the comparison of the experimental work with the results of Dravid et al correlation for laminar flow [19]. The Dravid et el correlation of the Nusselt number for helical coil can be expressed as follows

$$Nu = \left[0.76 + 0.65\sqrt{De}\right]Pr^{0.175}$$

for $50 < De < 2000$ where $D_e = Re\left(\frac{d_i}{D_c}\right)^{0.5}$

In this work a critical Reynolds number approximate about 6943 and the maximum Reynolds number is about 6765 at high mass flow rate so the flow is laminar. From the figure Dean number slightly decreases with decreasing curvature ratio. The curvature ratio and the velocity of the flow are two factors to determine the Dean number. This can be explained with higher velocity and higher curvature ratio which increases mixing effect due to intensify secondary flow which in turn enhances heat transfer. The deviation between Nusselt number obtained from above correlation and experimental results for coil with 0.037 curvature ratio is about 5.7 %.

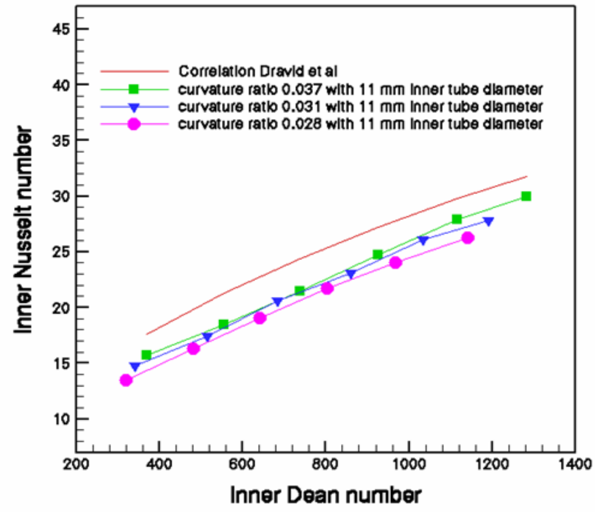

Fig. 11. Variation of Nusselt number with Dean number.

The experimental results of Nusselt number are related with Dean number and Prandtl number through the following correlation:

$$Nu = 0.382De^{0.541}Pr^{0.3}$$

Where $D_e = Re\left(\frac{d_i}{D_c}\right)^{0.5}$ for the conditions;

$320 \leq De < 1295$ and $4.2 \leq Pr < 4.8$,

curvature ratio $0.028 \leq \left(\frac{d_i}{D_c}\right) < 0.037$

and $1930 \leq Re < 6765$

The results of the present correlation are depicted in figures (12) with a deviation of about ± 4 %.

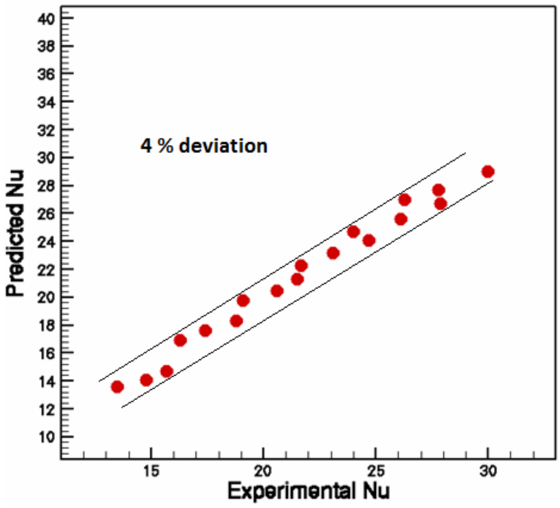

Fig . 12. Comparison between experimental data and predictions present correlation.

4. Conclusion

1. pressure drop increase with increase curvature ratio
2. pressure drop decrease with increase inner tube diameter
3. over all heat transfer coefficient increase with increase curvature ratio
4. The Nusselt number of the coil with curvature ratio 0.037 increases by approximately 12.3 % as compare with 0.028 curvature ratio in case of 11mm inner tube diameter.
5. The Nusselt number of the coil with curvature ratio 0.033 increases by approximately 11.6 % as compare with 0.028 curvature ratio for 400 mm coil diameter.
6. The Nusselt number obtained from Dravid et el correlation and experimental results for coil with 0.037 curvature ratio were close to with deviation is about 5.7 %.

Notation

Q_{avg}	Average heat transfer (W)
m_h	Hot air mass flow rate. (kg/s)
m_c	Cold water mass flow rate. (kg/s)
CP_c	Specific heat of water. (kJ/kg K)
CP_h	Specific heat of air. (kJ/kg K)
L	Tube length, (m)
k	Thermal conductivity, (W/m K)
ΔP	Pressure Drop, (N/m^2)
T	Temperature (K)
Pr	Prandtl number $Pr = \frac{\mu\, cp}{k}$
Re	Reynolds number $Re = \frac{\rho\, v\, d}{\mu}$

v Mean velocity (m/s)
U Overall heat transfer coefficient(W/ m^2 K)
h Convective heat transfer coefficient(W/ m^2 K)
Nu Nusselt Number
A Surface area (m^2)

Greek symbols

ρ Density, (kg/m^3)
μ Dynamic viscosity, (kg/m s)

Subscripts

h hot
h, i hot inlet
h, o hot outlet
c cold
c, i cold inlet
c, o cold outlet
o outside
i inside

5. References

[1] S.A. Berger, L. Talbot, L.S. Yao, Flow in curved pipes Annual Review of Fluid Mechanics 15 (1983) 461–512.

[2] D.G. Prabhanjan, G.S.V. Raghavan, T.J. Rennie, comparison of heat transfer rates between a straight tube heat exchanger and a helically coiled heat exchanger International Communications in, Heat and mass Transfer 29 (2) (2002) 185-191,

[3] Y.I.Kim, S.H.Kim, Y.D.Hwang, J.H. Park, Numerical investigation on the similarity of developing laminar flows in helical pipes,Nucl.Eng.Des.241(12) (2011) 5211-5224.

[4] M.T. Jamal-Abad, A. Zamzamian, M. Dehghan, Experimental studies on the heat transfer and pressure drop characteristics of Cu-water and Al-water nanfluid in a spiral coil, Experimantal Thermal and Fluid Science 47 (2013) 206- 212.

[5] V.Kubair,N.R. Kuloor, Flow of Newtonian fluids in Archimedean spiral tube coil correlation of the laminar,transition and turbulent flows, Indian J. Technol.4 (1969) 3-8.

[6] V.Kubair, N.R. Kuloor, Heat transfer to Newtonian fluid in coiled pipe in laminar flow, Int. J. Heat Mass Transfer 9 (1966) 63-75.

[7] T.J.Rennie, Numerical and experimental studies of a double pipe helical heat exchanger, Ph.D. dissertation, McGill University, Montreal, Canada, (2004)

[8] T.J.Rennie, V.G.S.Raghavan, Experimental studies of a double-pipe helical heat exchanger, Experimental Thermal and Fluid Science 29 (2005) 919-924.

[9] V.Kumar, S.Saini, M.Sharma, K.D.P.Nigam, Pressure drop and heat transfer study in tube-in-tube helical heat exchanger, Chemical EngineeringScience61 (2006) 4403-4416.

[10] N.Ghorbani, H.Taherian, M.Gorji, H.Mirgolbabaei, An experimental study of thermal performance of shell-and-coil heat exchangers, Int. Comm in Heat and Mass Transfer 37 (2010) 775–781.

[11] M. Moawed, Experimental study of forced convection from helical coiled tubes with different parameters, Energy Conversion and Management 52 (2011)1150–1156

[12] J.S.Jayakumar, S.M. Mahajani, J.C.Mandal, K.N.Iyer, P.K.Vijayan, CFDanalysis of single-phase flows inside helically coiled tubes, Computers and Chemical Eng. 34 (2010)430–446.

[13] Y.M.Ferng, W.C.Lin, C.C.Chieng, Numerically investigated effects of different Dean number and pitch size on flow and heat transfer characteristics in a helically coil-tube heat exchanger, Applied Thermal Engineering 36 (2012) 378-385.

[14] H.A. Mohammed, K. Narrein, Thermal and hydraulic characteristics of nanofluid flow in a helically coiled tube heat exchanger, International Communications in Heat and Mass Transfer 39 (2012) 1375–1383

[15] M.A. Akhavan-Behabadi, M.F. Pakdaman, M. Ghazvini, Experimental investigation on the convection heat transfer of nanofluid flow inside vertical helically coild tubes under uniform wall temperature condition, International Communications in Heat and Mass Transfer 39 (2012) 556-564.

[16] S.S. Pawar , V. K. Sunnapwar, Experimental studies on heat transfer to Newtonian and non-Newtonian fluids in helical coils with laminar and turbulent flow Experimental Thermal and Fluid Science 44 (2013) 792–804

[17] F.M. White, Heat Transfer, Addison Wesley Publishing Company Inc., New York, NY, 1984.

[18] Seban, R.A., Mclauchlin E.F. (1962), "Heat transfer in tube coils with laminar and turbulent flow", Heat mass transfer 6, 387-395

[19] A.N. Dravid, K.A. Smith, E.W. Merrill, P.L.T. Brian, "Effect of secondary fluid motion on laminar flow heat transfer in helically coiled tubes, American Institute of Chemical Engineers Journal 17 (5) 1971 1114-1112.

تحقيق عملي من تأثير نسبة الأنحناء على انتقال الحرارة في مبادل حراري حلزوني مزدوج الأنبوب

هشام عاصي حوشي
قسم الهندسة الكهروميكانيكية/ الجامعة التكنولوجية
البريد الالكتروني: hah_62@hahoo.com

الخلاصة

عدة عوامل متغيرة للفائف حلزونية مزدوجة الأنبوب وتم استخدام أربع لفائف .ثلاث مع نسبة انحناء (0.028, 0.031, 0.037) و 11 ملم قطر الأنبوب الداخلي في حين أن الرابع مع 0.033 نسبة انحناء و 13 ملم قطر الأنبوب الداخلي. يتدفق الماء الساخن في الأنبوب الداخلي في حين يتدفق الماء البارد في الحلقة. درجة حرارة مدخل الماء الساخن والماء البارد هي $50\ ^0C$ $18\ ^0C$ على التوالي. ويتراوح معدل التدفق الكتلي الداخلي من 0.0167 إلى 0.0583 كجم / ثانية . أظهرت النتائج زيادة عدد نسلت مع زيادة نسبة الانحناء. أن عدد نسلت من الملف نسبة انحناءه 0.037 يزيد بحوالي % 12.3 مقارنة مع الملف نسبة انحناءه 0.028 . كما كشفت النتائج أن عدد نسلت من الملف نسبة انحناءه 0.033 يزيد بحوالي % 11.6 مقارنة مع الملف نسبة انحنائه 0.028 لملف قطره 400 ملم.