The Effect of Pitch and Fins on Enhancement of Heat Transfer in Double Pipe Helical Heat Exchanger

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Abstract: Laminar heat transfer and flow investigated experimentally in the annulus of double pipe helical heat exchanger. Experimental study included designing and manufacturing two types of heat exchanger one with inner plain tube and the other with finned tube with three pitches coil 75mm, 50mm and 35mm. The combined effects of finned tube, pitches with different Dean Number range from 394 to 723 were studied and discussed. The combinations gives higher heat transfer enhancement with the smaller pitch in finned tube. This enhancement in finned tube up to 16 % at Dean Number 723.The correlation equation have been developed for predicting for Nusselt number and the maximum deviation between this equation and experimental results is about 10 %.

Keywords Helical coil, Double pipe, Finned tube, Heat exchanger, Dean Number.

Introduction

Improve the heat transfer is one of the most important tools in saving energy in multiple processes. The performance improvements in heat transfer to increase the heat transfer coefficient lead in turn to reduce the size and cost of the heat exchanger. The flow tube through coil spiral be more effective in heat transfer compared with flow in straight tube due to centrifugal force. Centrifugal forces generated secondary flow and this leads to increased friction factor as well as the heat transfer rate and this causes an increase in heat transfer rate between the wall and the flowing fluid. The helical pipes provide greater heat transfer area within a small space which causes largest heat transfer coefficient [1-2]. Timothy J et al. [3] performed experimental study of a double-pipe helical heat exchanger by using two differently sized heat exchanger and varied flow rate in the inner tube and annulus. Wilson plots were used to calculate the inner tube and the annulus heat transfer coefficient. The experimental data fit well with the numerical for the large heat exchanger but with some differences for smaller coil. M.A.Akhavan-Behadadi et al [4], investigated experimentally the heat transfer enhancement of a nanofluid flow inside vertically coiled tubes in thermal entrance region with constant wall temperature at 95C. The results show, utilizing helical coild tubes instead of straight ones enhances the heat transfer rate remarkably. Farzan Akbaridoust et al [5] have studied pressure drop and heat transfer behavior of nanofluid in helically coil tubes. More enhanced heat transfer and pressure drop was observed for tubes with greater curvature ratio. Zan Wu et al [6],investigated experimentally the heat transfer characteristics of tap water and alumina nanofluid for both laminar and turbulent flow inside a double-pipe helically coiled heat exchanger. The tap water and the nanofluid showed similar pressure drop and heat transfer performance, which indicates that the net effect of nanoparticles on the heat transfer performance in helically coiled tubes is probably insignificant. The present work studies the effect of coil pitches and fins on the outer surface of inner tube on enhancement of heat transfer in the annulus of double pipe helical heat exchanger.

Experimental Setup and Procedure

Figure (1) and (2) show scheme and photo of the experimental system. The experimental apparatus consists of two loops, for the hot and cold fluids. The hot water runs in the inner tube, while cold water is forced in the annulus. The rig composed of (1) Test section (double-pipe helical coil heat exchanger) (2) hot water tank (3) Cold fluid tank (4) Heat exchanger (5) pumps (6) rotameters (7) thermocouples and Selector switch (8) control valves (9) measuring apparatus

The double-pipe helical coil considered was constructed from copper tubing and standard copper connections. The inner tube has an inner diameter of 13mm with thickness 2 mm. The outer tube has an inner diameter of 26mm with thickness 1 mm. The outer surface of the inner tube was enhanced by finned with height of (1)mm. The number of turns of the helical coils is 4.5; the total length of the tested helical heat exchanger is 3.6 m. The double-pipe helical heat exchanger with different coil pitches 75 mm, 50 mm, 35 mm. Ball valves were used to control and measure the flow with rang from (1-4.5) l/min step (0.5) l/min in the annulus and (5) l/min in the inner tube. The water was heated in a tank by thermostatic electric heater set at 60C^o. The hot water was pumped from the hot tank to the inner tube and the cold water was pumped at 25 C^o from the tank to the annulus.

Data analysis

The Darcy friction factor was calculated from [7]:

$$f = 2.\frac{D_h}{L}\frac{\Delta P}{\rho u^2} \qquad (1)$$

Where

$$D_h = \frac{4A_c}{W_p} \qquad (2)$$

The averaged heat flux q between the heat transferred by the inner hot fluid q_h and the heat absorbed by the annulus cold water q_c is,

$$q = \left(\frac{q_h + q_c}{2}\right) = \left[\frac{C_{ph} m_h (T_{hi} - T_{ho}) + C_{pc} m_c (T_{co} - T_{ci})}{2}\right] \qquad (3)$$

The logarithmic mean temperature difference (LMTD) was determined as follow:

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln[(T_{hi} - T_{co})/(T_{ho} - T_{ci})]} \qquad (4)$$

And

$$q = A_O U_O LMTD \qquad (5)$$

Heat transfer coefficients for the annulus ho, were calculated using traditional 'Wilson plots'' as described by Rose [3,8]. Wilson plots are generated by calculating the overall heat transfer coefficients (U_O) for a number of trials where one fluid flow is kept constant and the other is varied. In this work, the flow in the inner tube side was kept constant and the flow in the annulus was varied. The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficients by the following equation [9]:

$$\frac{1}{U_O} = \frac{A_o}{h_i A_i} + \frac{A_O \ln\frac{A_o}{A_i}}{2\pi K L} + \frac{1}{h_o} \qquad (6)$$

By keeping the mass flow rate in the inner tube side constant, it is then assumed that the inner tube side heat transfer coefficient is constant and the tube wall could be considered constant. Therefore the change in the overall heat transfer coefficient would be mainly due to the variation of the annulus convection coefficient. The first two terms on the right-hand side of Eq. (6) are kept constant then,

$$\frac{1}{U_o} = B + \frac{1}{h_o} \qquad (7)$$

The annulus heat transfer coefficient with fluid velocity in the annulus u_o is assumed to obey the relation [3]

$$h_o = Cu_o^n \quad (8)$$

The equation (7) will be

$$\frac{1}{U_o} = B + \frac{1}{Cu_o^n} \quad (9)$$

The Nussult number is,

$$Nu = \frac{h_o D_h}{k} \quad (10)$$

and

Dean number in annulus for laminar flow is calculated from [3]:

$$D_e = \frac{\rho u_o}{\mu}\left(\frac{D_o^2 - D_i^2}{D_o + D_i}\right)\left(\frac{D_o - D_i}{R}\right)^{1/2} \quad (11)$$

Thermal performance factor η is a function of the heat transfer and pressure drop simultaneously, the Nusselt number ratio to the friction factor ratio at the same pumping power [10], or

$$\eta = \frac{Nu_{finned\ tube}/Nu_{plain}}{(f_{finned}/f_{plain})^{1/3}} \quad (12)$$

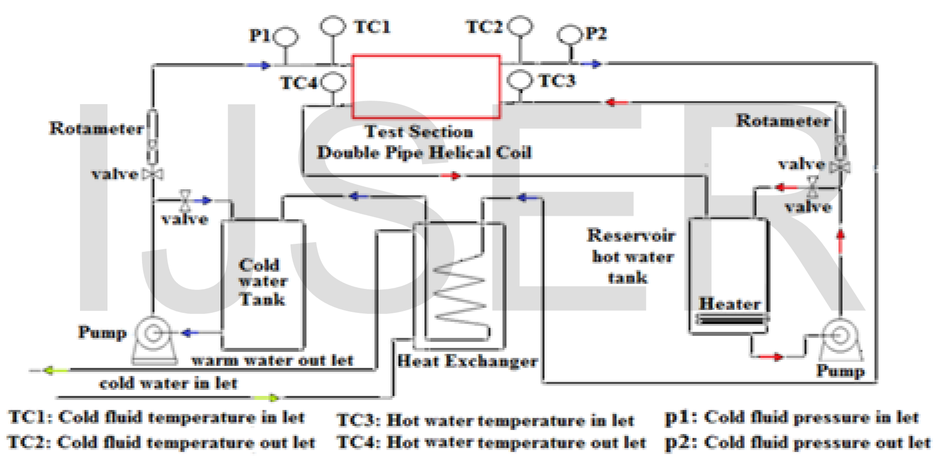

Fig.(1) Schematic of the experimental setup

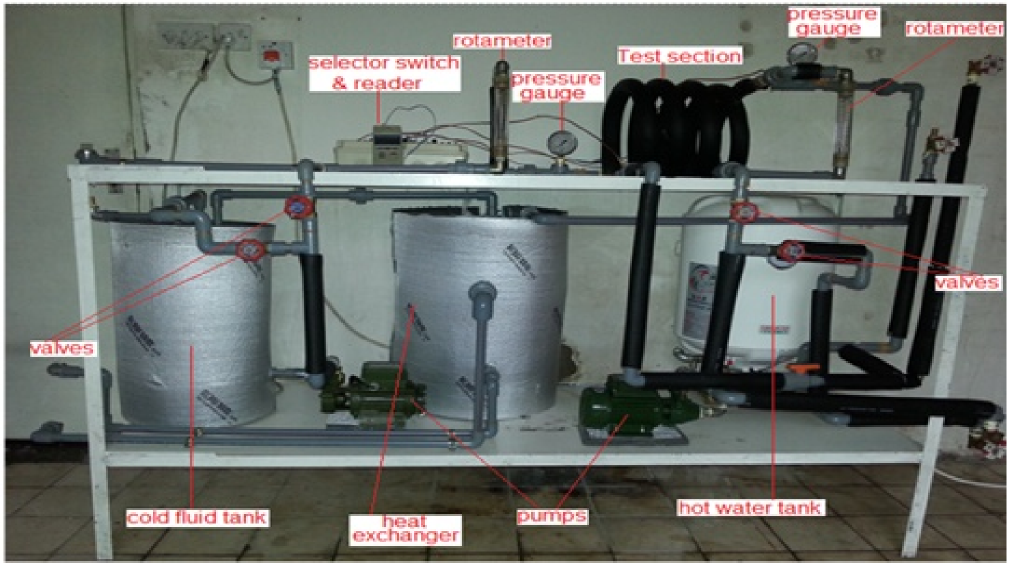

Fig. (2) photo of the experimental setup

Results and Discussion

Fig (3) and (4) show comparison between present work and correlation of xin and Ebadian [11] for Nusselt number and Seban and Mclauchlin [12] for friction factor verses Dean number for laminar flow in the annulus of double pipe helical heat exchanger for plain inner tube using distilled water, the results of the present work are in good agreement with those correlations.

Figures (5) shows the effect of using finned tube of helical Coil on Nusselt number within the range of Dean number from 394-723 with three pitch coil 75 mm, 50 mm and 35 mm. It is found that the tube with finned gives higher heat transfer rate than plain tube. The increase is almost 16 % in pitch 35mm at the same Dean number. Figures show the effect of finned tube is more than the effect of pitch of helical coil and the value of Nusselt number at smaller pitch on plain tube is 44.5 whereas the value of Nusselt number at larger pitch on finned tube is 45.36 and 48.1for smaller pitch. This is due to the effect of finned tube is more than the effect of Pitch of helical coil because of that the annulus heat transfer coefficient for finned tube is larger than that for plain tube due to the intensive secondary flow and swirl induced by the finned on the outer surface of the inner tube also increasing in the surface area by finned.

Figure (6) shows the friction factor experimental results with rang of Dean Number in laminar flow for all cases, friction factors considerably decrease with increasing Dean Number. Using smaller pitch leads to higher friction loss than that of larger pitch. The friction factor obtained from finned tube is significantly higher than that plain tube due to friction generated by outer surface of the inner finned tube as well as larger constant surface area and the intensive secondary flow and swirl induced by the finned on the outer surface of the inner tube. Figure (7) shows the relative Nusselt number ($\mathrm{Nu_{finned}}/\mathrm{Nu_{plain}}$) versus the Dean number, the relative Nusselt number slightly increases with increasing The Dean number and with smaller pitch. Figure (8) shows the evolutions of relative friction factor ($\mathrm{f_{finned}}/\mathrm{f_{plain}}$) versus Dean Number. The friction factor ratio increases with decreasing the Dean number.

Figure (9) shows the variation of performance ratio with Dean Number, from this figure it can be seen that use of helical double pipe finned tube result in further increase in Nusselt number with increase Dean number in laminar flow. The helical finned tube should enhance the heat transfer due to increasing the surface area and swirl induced by the finned on the outer surface of the inner tube and the values of this

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ratio for all cases are greater than one. Figure (10) shows the maximum deviation between the equation of correlation and experimental results about 10 %.

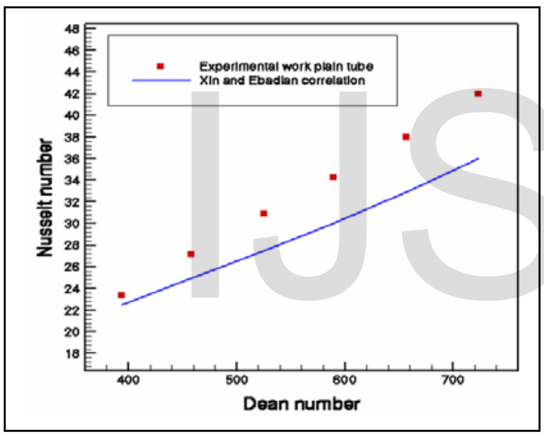
$$\mathrm{Nu} = 0.055\mathrm{De}^{0.91}\mathrm{Pr}^{0.32}(\mathrm{H/p})^{0.12} \qquad (13)$$

For the conditions

Dean number $394 \leq \mathrm{De} < 723$, Prandtl number $5.4 \leq \mathrm{Pr} < 4.8$, coil heigh to pitch ratio, $1.87 \leq (\mathrm{H/p}) < 4$

Conclusions

1. The Nusselt number increase with increase Dean Number and decrease of coil pitch, and the friction factor increase with decrease pitch of coil.
2. The combination of using finned tube with effect of pitch gives higher heat transfer enhancement or smaller coil pitch augment higher heat transfer.
3. The performance factor is more than unity. For this purpose the enhancement heat transfer is competent from the viewpoint of energy saving.
4. The correlation equations for Nusselt number have been developed and the maximum deviation with experimental results is about 10 %.



Fig(3) Nusselt number verses Dean Number

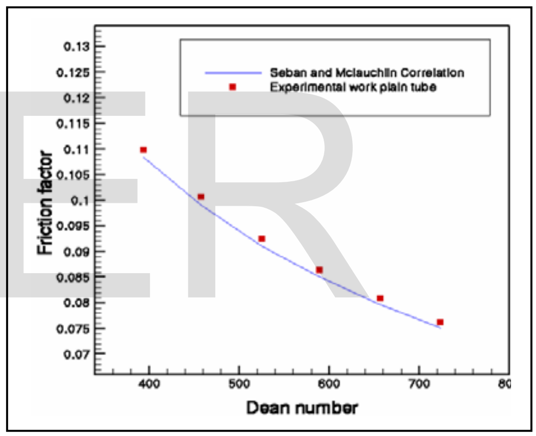


Fig (4) Friction factor verses Dean Number

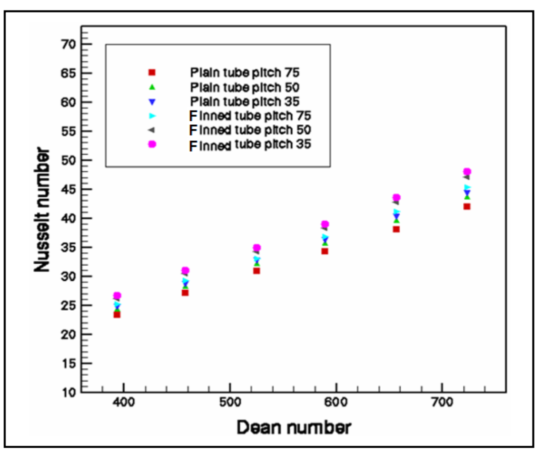


Fig (5) Nusselt number verses Dean Number for plain and finned tube

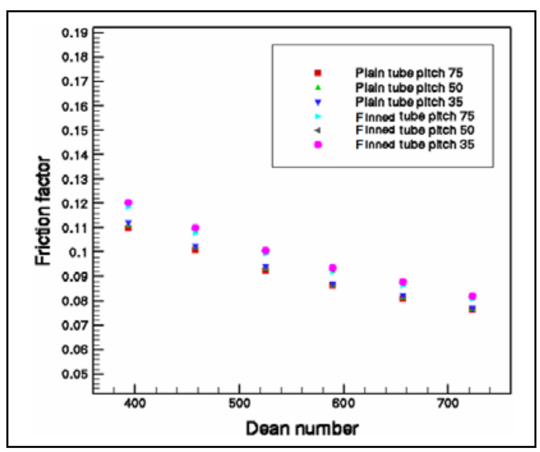


Fig (6) Friction factor verses Dean Number for plain and finned tube

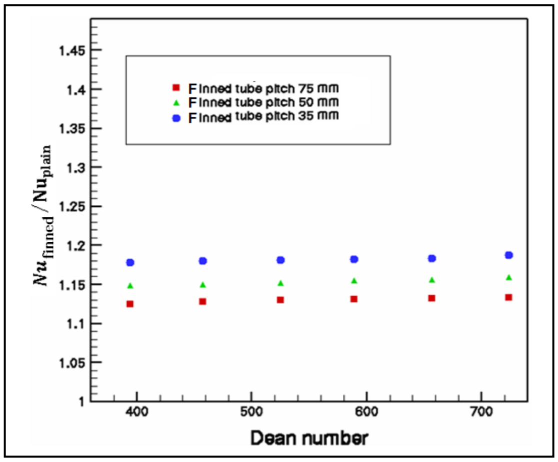


Fig (7) Relative Nusselt number versus Dean Number

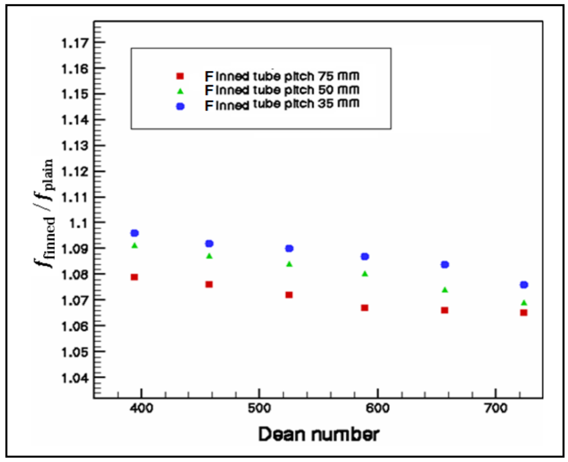


Fig (8) Relative friction factor versus Dean Number

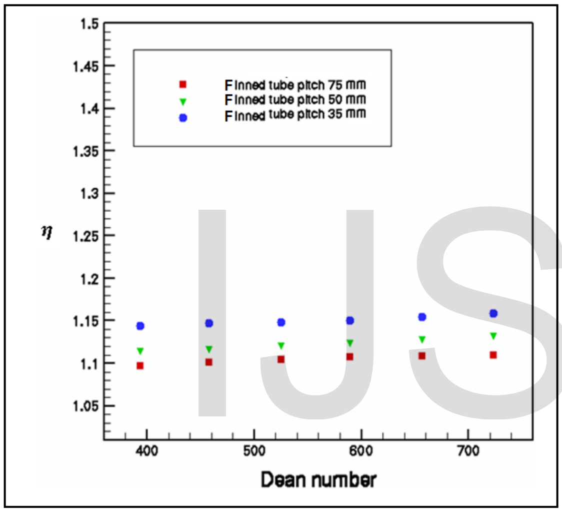


Fig (9) performance ratio with Dean Number

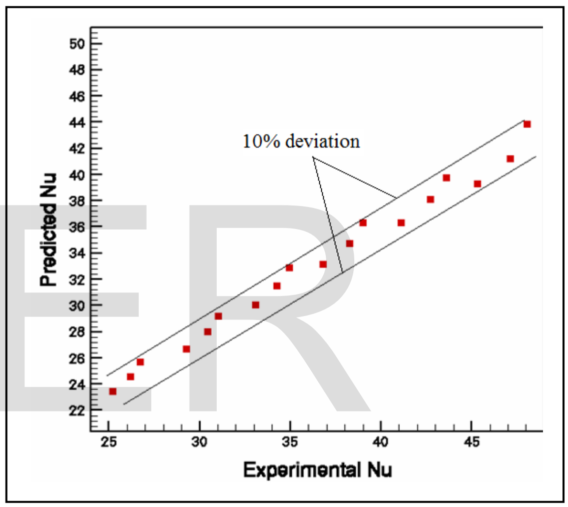


Fig (10) predicted versus experimental results

Nomenclature

Ac	cross section area	(m^2)
Ai	inner surface area of the inner tube	(m^2)
Ao	outer surface area of the inner tube	(m^2)
B	Constant in Eq. (8)	
C	Constant in Eq. (7)	
Cp	Specific heat at constant pressure	$(Jkg^{-1}K^{-1})$
D	diameter of annulus	(m)
Dh	hydraulic diameter	(m)
f	friction factor	
H	high of coil at pitch equal zero	(m)
h	heat transfer coefficient	$(Wm^{-2}K^{-1})$
ho	annulus heat transfer coefficient	$(Wm^{-2}K^{-1})$
k	thermal conductivity	$(Wm^{-1}K^{-1})$
L	length of the helical heat exchanger	(m)
m	mass flow rate	$(kg\ s^{-1})$
Nu	Nusselt number	
n	exponent in Eq. (7)	
p	pitch of helical coil	(m)
Pr	Prandtl number,	$(Cp\mu/k)$
R	radius of curvature of the coil	(m)
T	temperature	(K)
u	velocity	$(m\ s^{-1})$
Wp	Wet perimeter	(m)
Δp	pressure drop	(Pa)
μ	dynamic viscosity	(Pa s)
ρ	density	$(kg\ m^{-3})$

Subscripts

c	cold side
ci	cold side inlet
co	cold side outlet
h	hot side
hi	hot side inlet
ho	hot side outlet
i	inside / inner
o	outside / outer

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