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Enhancement of Heat Transfer in The Tube-Sid of A Double Pipe Heat Exchanger by Wire Coils

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Abstract

Heat transfer enhancement by wire coils is adopted in order to increase the thermal performance of a double pipe heat exchanger. The heat exchanger adopted is 1245 mm effective length, 28 mm outer diameter and changeable inner diameter (11 or 14 mm). Wire coils of e = 1 mm and p = 10, 20, 30 and 40 mm are used as turbulence promoters to augment heart transfer inside the inner tube at a Reynolds number range of 5000 to 40000. Water is used as the working fluid in the two sides. Variation in the experimental conditions is attained by changing the mass flowrates of unenhanced side and changing the inlet temperature of hot fluid. These conditions are followed in order to have as large amount of data points as possible in addition to observe the effect of changing these conditions. Heat transfer is increased inside the inner tube by 2.43 folds, as compared to empty tube at the same Reynolds number accompanied by friction factor increase of 4.75 folds. New correlations of Nusselt number and friction factor for the enhanced tubes are proposed as functions of Reynolds number, Prandtl number and the geometrical characteristics of inserts and tube sizes.

Introduction

Wire coils inserts are currently used in applications as oil cooling devices, preheaters or fire boilers. They show several advantages with respect to other enhancement techniques for its low cost, easy installation and removal, preservation of original plain tube mechanical strength and possibility of installation in an existing smooth tube heat exchanger. Kumar et al., [1] examined the influence of wire coils inserted in a tube on heat transfer and the pressure drop. Water is used as the test fluid. The pitch (p/d_i=1.05-5.5) and the wire size (e/d_i=0.1-0.15) were employed. They had maximum increase of heat transfer of 280% with finding a large increase of pressure drop. Zhang et al., [2] investigated heat transfer and friction factor of hot air, regarding the influence of pitches and wire diameter of the helical coils in tubes. Viedma, et al., [3] experimentally studied wire coils inserted in a tube to observe their thermodynamic behavior in laminar, transition and turbulent flows. They used water and propylene glycol mixtures at different concentrations, for a range of Reynolds number of 100 to 90,000 and Prandtl number from 2.8 to 200. Their results showed that the wire coil increased pressure drop up to 9 times and heat transfer up to 4 times compared to the empty smooth tube. Eiamsa-ard et al., [4] studied experimentally heat transfer, friction factor and thermal performance behaviors in a tube equipped with the combined devices between the twisted tape and constant and periodically varying wire coil pitch ratio. Their experiments were conducted in a turbulent flow regime with Reynolds numbers ranging from 4600 to 20000 using air as the test fluid. They found that heat transfer rate was further augmented by the compound devices by 3.65 tames compared to plane tube, 1.39 times compared to wire coil insert and 2.34 times compared to tube inserted with twisted tape with friction factor augmentation about 28.8, 2.24 and 8.37 respectively.

Experimental Work

An experimental rig is designed and assembled to carry out the experiments that require particular fluid temperatures, particular fluid flowrates, and for each run, temperatures of four points and pressure drop in specific sections which represent the heart of the present work, must be measured in acceptable accuracy (± 0.6%), as shown in fig. 1. Water is used as the working fluid in the two sides of the double pipe heat exchanger for its availability and being the most used fluid in heat exchangers in the industrial applications.

The experimental rig consists of: (1) Cold fluid unit designed to supply water at 20 ± 0.5, where tap water is used for one pass without circulating. A heating tank and a water cooler are the most important parts used to maintain the tap water to the wanted temperature. (2) Hot fluid unit is used to supply water at 60 and 70 ± 0.5 °C, circulated throughout the system. (3) Fluid flow measurements consisting of 0.12-1.6 m^3/hr range rotameter for cold water and an orifice plate of 0.05-0.8 m^3/hr range for the hot water. (4) Temperature measurement device type DORIC with four thermocouples type J having an accuracy of ± 0.6%.

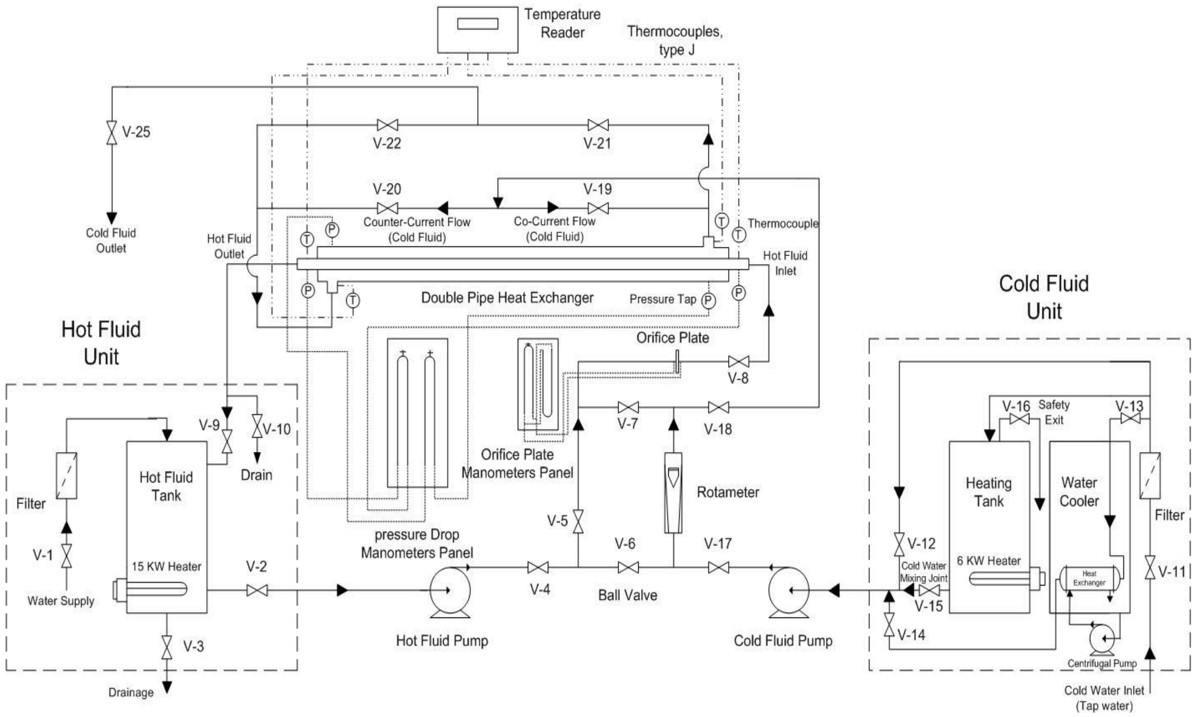

Fig. 1: A schematic diagram of the experimental rig

(5) Manometers panel including a water and mercury manometers, and (6) A double pipe heat exchanger used as the test section as shown in fig. 2. The latter consists of a stainless steel pipe of ID = 28 mm and OD = 32 mm and a changeable inner tube made of copper of ID = 11 mm and OD = 12.5 mm or ID = 14 mm and OD = 15.5 mm. Cold water is flowing in the annulus of the heat exchanger while hot water is flowing in the inner tube. The two fluids are flowing counter-currently. The four thermocouples as well as the pressure taps are fixed at the inlet and outlet of the hot and cold water in a manner obvious in fig. 2.

The wire coils, employed, had one wire diameter (e = 1 mm) and four coiling pitches (p = 10, 20, 30 and 40), (i.e., the present study is concerned with the range of e/d_i = 0.0714 to 0.0909 and p/d_i = 0.7143 to 3.6364), as illustrated in fig. 3, for the two tube sizes, i.e., having eight samples to be

studied. In addition, the experimental conditions are varied by adopting two mass flowrates for the cold water flowing in the annulus (0.1 and 0.15 kg/s) for the range of hot water flowing in the inner tube (Re = 5000 to 40000). Furthermore, the hot water inlet temperature is 60 and 70 ± 0.5 °C for all changes above with keeping the cold water inlet temperature fixed at 20 ± 0.5.

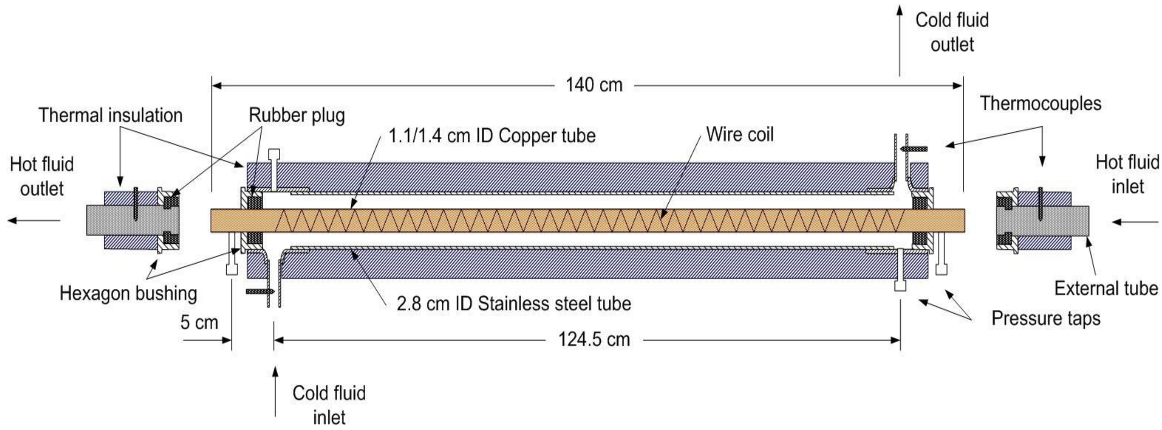

Fig. 2: A sketch of the double pipe heat exchanger

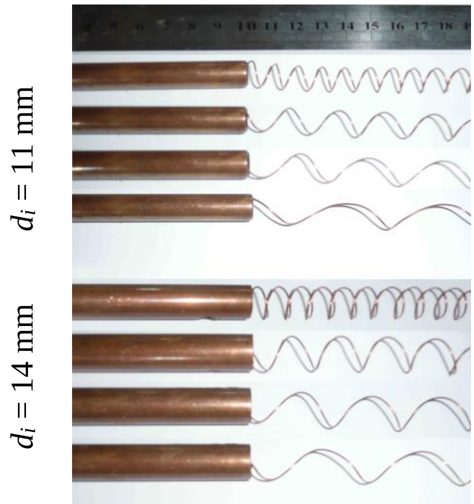

Fig. 3: Wire coils employed in the experiments

Calculations

For both the enhanced and unenhanced sides of the heat exchanger, Darcy friction factor is calculated directly by using Darcy-Weisbach equation:

$$\Delta p = f \frac{L}{d_i} \rho \frac{v^2}{2} \qquad (1)$$

adopting the pressure drop measured by the manometers. The friction factor calculated for the inner tube side is one of the characteristics under study while that for the annulus side is used in Gnielinski equation [5]:

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{0.5}(\mathrm{Pr}^{2/3}-1)}\left[1+\left(\frac{d_i}{L}\right)^{2/3}\right] \qquad (2)$$

which is adopted to calculate Nusselt number in the annulus corrected by the correction factor adopted by Petukhov and Roizen [6] to enable eq. (2) to be suitable for use in case of the annulus:

$$\frac{Nu_{annulus}}{Nu_{tube}} = 0.86\left(\frac{D_i}{D_o}\right)^{-0.16} \qquad (3)$$

Heat transfer rate is calculated by considering the enthalpy change in the inner tube and the annulus side and the average value is adopted:

$$q = \dot{m}Cp\left(T_{out} - T_{in}\right) \quad (4)$$

Using the equation:

$$q = UA\Delta T_m \quad (5)$$

Where

$$\Delta T_m = LMTD \equiv \frac{\left(T_{h2} - T_{c2}\right) - \left(T_{h1} - T_{c1}\right)}{\ln\left[\left(T_{h2} - T_{c2}\right)/\left(T_{h1} - T_{c1}\right)\right]} \quad (6)$$

with eq. (4) gives the overall heat transfer coefficient. The thermal resistances in the tube side the tube wall and the annulus side are combined to calculate the heat transfer coefficient (and then Nusselt number) in the inner tube as in the relationship [7]:

$$U_o = \frac{1}{\frac{A_o}{A_i}\frac{1}{h_i} + \frac{A_o \ln\left(r_o/r_i\right)}{2\pi kL} + \frac{1}{h_o}} \quad (7)$$

Results And Discussion

Nusselt number as well as friction factor are the two important parameters deciding the efficacy of the heat transfer enhancement process. Nusselt number values for different coiling pitches, different tube sizes, and experimental conditions are calculated using the procedure explained above. In fig. 4, Nusselt number vs. Reynolds number relationship for the inner tube of 14 mm diameter at the experimental conditions (T_{h1} = 60 °C and m_c = 0.1 kg/s) are plotted

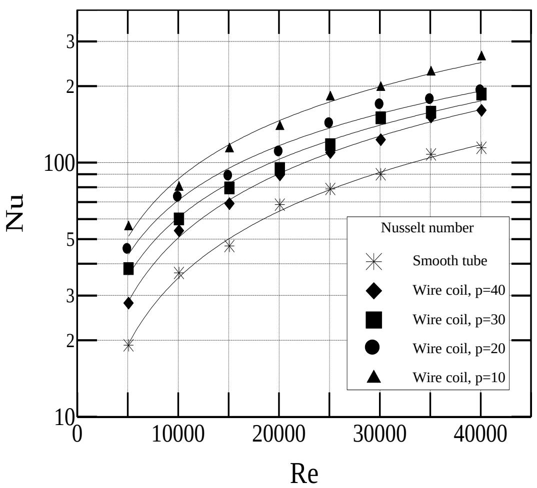

Fig. 4: Nusselt number vs. Reynolds number.

Similar plots are obtained for the other three conditions for the two tube sizes giving 256 Nusselt number-Reynolds number points. These points are used to produce a new correlation of Nusselt number as function of Reynolds number, (e/d_i), (p/d_i) and Prandtl number as in eq. (8):

$$Nu_a = 0.0668\,\text{Re}^{0.7938}\,\text{Pr}^{0.2741}\left(\frac{e}{d_i}\right)^{0.2049}\left(\frac{p}{d_i}\right)^{-0.3532} \quad (8)$$

Eq. (8) is valid for the range of Reynolds number of 5000 to 40000, e/d_i = 0.0714 to 0.0909 and p/d_i = 0.7143 to 3.6364. In addition, this equation is valid for water only in the range of temperatures employed in the hot fluid stream of the present work which is about 50 to 70 °C.

Fig. 4 reveals that the dependency of Nusselt number on Reynolds number is very close to that in smooth tube. Furthermore, Nusselt number increases with decreasing the coiling pitches or the ratio (p/d_i).

The other characteristic that is important to be studied in the field of heat transfer enhancement is the friction factor being the penalty one must pay for enhancing heat transfer. Friction factor in the inner tube is calculated by adopting the pressure drop in the inner tube and calculated by eq. (1). Friction factor values of smooth tube and with wire coils for the 14 mm diameter tube size for the four experimental conditions are plotted together in fig.5.

Adopting the two inner tube sizes and the four experimental conditions has led to 256 friction

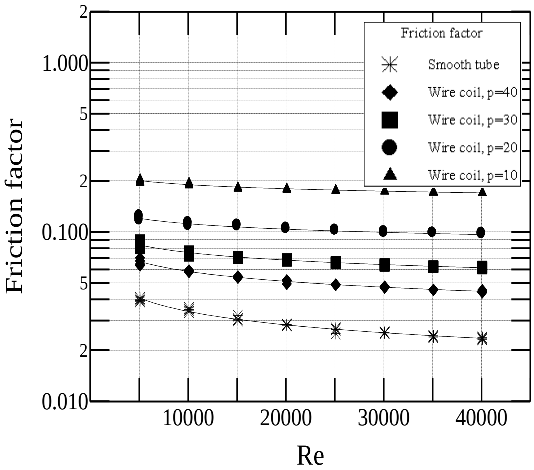

Fig. 5: Friction factor vs. Reynolds number.

factor-Reynolds number points. These points are used to produce a new correlation of friction factor in enhanced tubes by wire coils as a function of Reynolds number and the geometrical characteristics of the wire coils in the rages mentioned above. The obtained correlation is:

$$f_a = 3.6346\,\mathrm{Re}^{-0.0964}\left(\frac{e}{d_i}\right)^{0.8912}\left(\frac{p}{d_i}\right)^{-0.7856} \qquad \textbf{(9)}$$

Fig. 5 as well as eq. (9) reveals low dependency of friction on Reynolds number especially in case of small coiling pitches. This fact might be considered as a good agreement with the relationship of the friction factor, Reynolds number and the relative roughness where the dependency of friction factor decreases with increasing the relative roughness [8]. Here the wire coil is considered as the roughness.

Considering the accuracy in calculations, actual correlations for Nusselt number and friction factor in smooth tubes are obtained by correlating the values of Nusselt number friction factor that are fixed in fig. 4 and 5 and the similar values producing the following relationships:

$$Nu_s = 0.013\,\mathrm{Re}^{0.833}\,\mathrm{Pr}^{0.265} \qquad (10)$$

$$f_s = 0.4185\,\mathrm{Re}^{-0.2708} \qquad (11)$$

Eq. (10) deviates by 8.6 % from Gnielinski equation (eq. (2)), while eq. (11) deviates by 8.1 % from Petukhov equation [10] which is concerned with the friction factor in smooth tube as fixed here

$$f = \left(0.79\ln\mathrm{Re} - 1.64\right)^{-2} \qquad (12)$$

Dividing eq. (8) by eq. (10) gives the Nusselt number augmentation as plotted in fig. 6 for the 14 mm diameter tube.

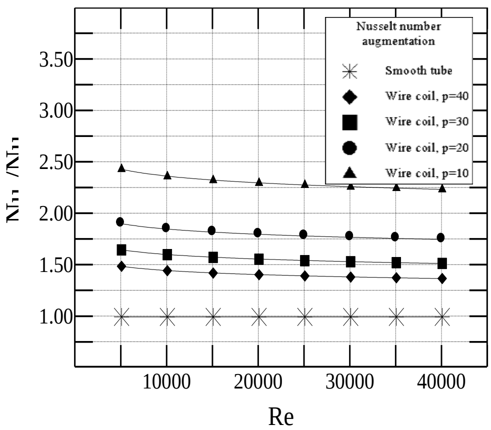

Fig. 6: Nusselt number augmentation.

On the other hand, the friction factor augmentation is obtained by dividing eq. (9) by eq. (11). Fig. 7 represents the friction factor augmentation in case of the 14 mm diameter tube

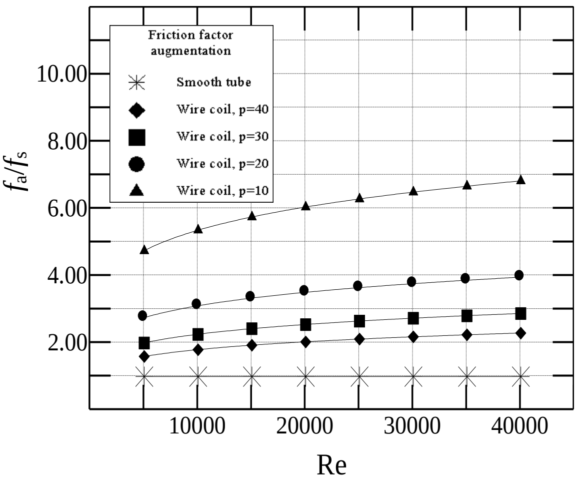

Fig. 7: Friction factor augmentation.

In fig. 6 and 7, the maximum value of Nusselt number augmentation obtained is 2.43 at Reynolds number of 5000 and p = 10, corresponding to a maximum in friction factor augmentation of 4.75. For the same geometries, for high Reynolds number (40000), the maximum Nusselt number is 2.24 with corresponding friction factor augmentation value of 6.83. That means that the maximum heat transfer augmentation is obtained for the wire coil of geometries of (e/d_i = 0.0714 and p/d_i = 0.7143).

Conclusions

In the present study the following can be concluded:

1. Heat transfer as well as friction factor increases with increasing the intensity of coils or decreasing the coiling pitches.
2. Heat transfer enhancement by wire coils is more effective at low values of Reynolds number than high values.
3. For the range of Reynolds number of 5000 to 40000 and e/d_i = 0.0714 to 0.0909 and p/d_i = 0.7143 to 3.6364, a maximum increase in Nusselt number of 2.43 folds is obtained at Re = 5000, e/d_i = 0.0714 and p/d_i = 0.7143, corresponding to an increase in friction factor of 4.75 folds.

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Notations

Symbols	Description
A	Heat exchange surface area [m^2]
Cp	Heat capacity [J/kg.°C]
d	Tube diameter [m]
e	Wire or rib diameter [m]
f	Darcy friction factor [—]
h	Convective heat transfer coefficient [$W/m^2.°C$]
k	Thermal conductivity [W/m.°C]
L	Length [m]
$\dot{m}$	Mass flowrate [kg/s]
Nu	Nusselt number (hd/k) [—]
p	Coiling or ribbing pitch [m]
Pr	Prandtl number (Cpμ/k) [—]
q	Heat transfer rate [W]
r	Radius [m]
Re	Reynolds number (ρ d ν/μ) [—]
T	Temperature [°C]
U	Overall heat transfer coefficient
Δp	Pressure drop [N/m^2]

Greek Symbols

N	Fluid Velocity [m/s]
P	Density [kg/m^3]

Subscripts

a	Augmented
c	Cold
h	Hot
i	Inner
m	Mean
o	Outer
s	Smooth

Abbreviations

ID	Inner diameter of tube
OD	Outer diameter of tube
LMTD	Logarithmic Mean Temperature Difference

تحسين انتقال الحرارة في أنابيب سيد من انبوب باستخدام محلزنات سلكية

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الخلاصة

تم استخدام محلزنات سلكية بقطر ١ ملم وفواصل لف بمقدار ١٠، ٢٠، ٣٠ و٤٠ ملم كمسببات شدة الاضطراب داخل الانبوب الداخلي لمبادل حراري من نوع الانابيب المتمركزة بطول ١٢٤٥ ملم وبقطر خارجي ٢٨ ملم وانبوب داخلي قابل للتغيير بقطرين ١١ و ١٤ ملم وذلك من اجل زيادة كفائتة الحرارية في مدى رقم رينولدز بين ٥٠٠٠ و٤٠٠٠٠. أستخدام الماء في جانبي المبادل الحراري كما تم تنويع الظروف التجريبية وذلك بتغيير معدل الجريان الكتلي للجانب الغير المراد رفع كفائته وكذلك بتغيير درجة حرارة الدخول للمائع الساخن. الهدف من تغيير الظروف التجريبية هو الحصول على اكبر ما يمكن من النقاط التجريبية للحصول على معادلات تجريبية بأدق مايمكن بالاضافة الى معرفة مدى تأثير تغيير تلك الظروف. تم زيادة انتقال الحرارة في داخل الانبوب الداخلي بمقدار ٢,٤٣ ضعف ماهو عليه في حالة استخدام انبوب املس بنفس رقم رينولدز وكان ذلك مصحوبا بازدياد معامل الاحتكاك بمقدار ٤,٧٥ اضعاف. تم الحصول على معادلات تجريبية جديدة لرقم نسلت ومعامل الاحتكال للانبوب الداخلي للمبادل الحراري وذلك كدوال لرقمي رينولدز وبرانتل بالاضافة للابعاد الهندسية للمضافات والانابيب.