EXPERIMENTAL INVESTIGATION OF NATURAL CONVECTION HEAT TRANSFER IN A HELICALED COILED PIPE

Saba Y. Ahmed
Saba_ya@yahoo.com

Zainab Hussain
zainabalammar945@Gmail.com

ABSTRACT :-

The research presents an experimental investigation on the steady state natural convection heat transfer for helical coiled pipes with a vertical orientation. Water is used as the working fluid for both the target fluid and the bath medium. The heat transfer characteristics have been studied for three helical coils that having three curvature ratio (0.056, 0.0625, and 0.115) and three coil pitches (12.5, 25, and 37.5)mm with a volume flow rate of the target fluid ranges (10-25)L/min and two constant bath water temperatures (70 and 90)^{o}C. All the experiments were performed for Rayleigh number range ($1.76*10^{9}$ – $1.15*10^{12}$). The experimental data have been used in SPSS software to predicted the empirical correlation of outside Nusselt number as a function of Rayleigh number by using height of the coil as a characteristic length then using this empirical relation in a computer program that have written in Fortran 90 to predict the outlet temperature and compared with the measured one. The results show that the inner and outer heat transfer coefficients increase with increasing of curvature ratio (d/D) and the mass flow rate. Also the results show that the difference between the measured and predicted outlet temperature equal to 1.5^{o}C.

Keywords: helical coiled pipe, natural convection, heat exchanger, curvature ratio, heat transfer coefficient .

دراسة عملية لانتقال الحرارة الطبيعي في الانبوب الملتف حلزونيا

صبا يعسوب احمد زينب حسين
جامعة بابل /كلية الهندسة/قسم الهندسة الميكانيكية

الخلاصة :-

تضمن البحث دراسة تجريبية لانتقال الحرارة بالحمل الحر المستقر ولانبوب ملتف حلزونيا وموضوع بشكل عمودي. المائع المستخدم هو الماء ولكل من الانبوب الحلزوني والحوض المائي تم دراسة خواص انتقال الحرارة لثلاث انواع من الانابيب الحلزونية وبثلاث نسب انحناء (0.115,0.0625,0.056) ولثلاث خطوات (12.5، 25، 37.5) ملم مع معدل تدفق حجمي يتراوح (25-10) لتر/دقيقة، ودرجات حرارة ثابته للحوض بمقدار (70 ^{o}C ، 90^{o}C) . تم اجراء الدراسة عند مدى رقم رايلي يتراوح (1.76×10^{9}- 1.15×10^{12}) تم استخدام النتائج التجريبية في برنامج احصائي من اجل ايجاد علاقة تربط بين رقم نسلت الخارجي و رقم رايلي باعتماد ارتفاع الملف كطول مميز في رقم رايلي ومن ثم استخدام تلك العلاقة في برنامج حاسوبي مكتوب بلغة فورتران 90 من اجل تنبؤ درجة حرارة الملف الخارجة . بينت النتائج ان معامل انتقال الحرارة الداخلي والخارجي يزداد مع زيادة نسبة الانحناء (d/D) وزيادة معدل التدفق واوضحت ايضا ان الفرق بين درجة حرارة الملف الخارجة المتنبئة والمقاسة هي (1.5 ^{o}C).

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NOMENCLATURE :-

Latin symbols

A surface area, m^2
Cp specific heat, kJ/kg ^oC
d tube diameter, m
D helix coil diameter, m
h heat transfer coefficient, $W/m^2.\,^oC$
H helix height, m
k thermal conductivity, W/m. ^oC
L coil length, m
m mass flow rate, kg/s
N number of turns
P coil pitch, m
Q heat transfer rate, W
r tube radius, m
R helix coil radius, m
T temperature, oC
u velocity, m/s
U overall heat transfer coefficient, $W/m^2\,^oC$

Dimensionless numbers

De Dean numbe, $De = Re\,(d/D)^{0.5}$
Gr Grashof number, $Gr= g\beta(T_F - T_b)H^3/\nu^2$
Nu Nusselt number, $Nu=hH/k$
Pr Prandtl number, $Pr = c p\mu/k$
Ra Rayleigh number, $Ra= Gr*Pr$
Re Reynolds number, $\rho\, d\, u/\mu$

Greek Symbols

Δ difference
μ Fluid viscosity, kg/m.s
ν Kinematic viscosity m^2/s
ρ density, kg/m^3

Subscripts

b Bath
F Film
i Inside
in Inlet
o outside
out outlet
w wall

INTRODUCTION :-

Today becomes need to increase the heat transfer performance of heat exchangers in order to reduce the energy and material consumption, as well as the associated impact on environmental degradation, has led to the development and usage of many heat transfer enhancement techniques. Enhancement can be achieved by increasing the convection heat transfer coefficient and/or the convection surface area. Different methods have been developed, and they are characterized as either passive or active technique. In active technique the heat transfer rate is improved by supplying an extra energy to the system such as fluid vibration, electric field and surface vibration. While for passive technique generally consists of geometric or material modification of the primary heat transfer and examples include finned surfaces, swirl flow that producing by inserting (turbolater) and coiled tubes, among others. Coiled tubes are used in heat exchangers, condensers and evaporators in food processing, chemical engineering, refrigeration, air conditioner and nuclear reactors due to several advantages of coils such as:

1- A coil provides a large surface area in relatively small reactor volume.
2- The entire surface area of the bended pipe is exposed to the moving fluid, which abolition the dead zones which are a typical disadvantage of the shell and tube heat exchanger.
3- Give a better heat transfer performance, since they have a lower wall resistance and high process side coefficient.
4- Eliminate a thermal shock and expansion problem that helps for high pressure operations.

5- Fouling factor is less in helical coil type than that of shell and tube type due the turbulence created inside it.

The governing parabolic differential equations three dimensional flow as solved by the finite difference marching technique of **PATANKAR (1974)** which applied to developing and fully developed flow appears as the solution for the sections which are remote from the entrance. The effects of the dean number on the friction factor and heat transfer are present. An experimental and numerical studies were made by **(Janssen 1978)** on convective heat transfer in coiled pipes. The experimental carried out for tube to coil diameter ratios (d/D) range (0.01 - 0.1), Prandtl number range (10 - 500) and Reynolds number range (20 - 4000), the heat transfer process was studied for both uniform peripherally averaged heat flux and constant wall temperature. A comparison of the overall heat transfer coefficient was made between a constant wall temperature and constant average heat flux and showed that the effect of the boundary condition is small. Steady state natural convection heat transfer from vertical helical coiled tubes was studied experimentally by **(Ali 1994)**. Turbulent natural convection heat transfer to water was obtained for Rayleigh number up to 10^{15} and four pipe to coil diameter ratios (d/D), with number of turns five and ten respectively, and five pitch to outer pipe diameter ratios (p/d). The data were correlated with the Rayleigh number for two different coil sets. The first set had d_o = 0.012 m and three D/d_o of (20.792, 13.923, and 9.941) And the second set had d_o = 0.008 m and two D/d_o of (19.957 and 9.941). The characteristic length used in the Rayleigh number was the pipe length. The results show that the heat transfer coefficient decrease in the turbulent region. Also, the results show for the first set that the heat transfer coefficient decreases with the increase number of turns while decreases for the second set.

Prabhanjan et al. (2000) designed and built a water bath thermal processor to study the influence of helical coil characteristic on heat transfer to Newtonian fluids such as water and base oil that have three different viscosities. The system consisted of a thermally insulated water bath, an electric heater, pump to re-circulated water in the bath and for pumping the processing fluid through the coil, copper helical coils and a storage tank for the processing fluid. The results show that the outer and total heat transfer coefficients were significantly lower in natural than in forced convection water bath. While, inner heat transfer coefficient was not significantly affected. Also, the flow rate inside the coil had slight effect on Nusselt number due to change in the temperature gradient along the length of the coil while for the three base oils showed significant different in viscosity after heating the oil for several turns. **Shokouhm et al. (2008) and Salimpour (2009)** investigated experimentally the heat transfer coefficients of shell and helically coiled tube heat exchangers that studied the effect of the coil pitch and the curvature ratio on the heat transfer coefficient. the result show that the shell-side heat transfer coefficients of the coils with a higher value of pitch is larger of small value of pitch also the shell-side Nusselt numbers of counter-flow configuration were slightly more than the ones of parallel-flow configuration. A computational fluid dynamics (CFD) methodology was used by **Ferng, et al. (2012)** to investigate the effect of different Dean number (De) and pitch size on the thermal hydraulic characteristics in a helically coil-tube heat exchanger. From the experimental data obtained correlation for the Nusslet number as a function of Dean number was developed, and the value of Nu number increases as the pitch size increases**.**

Patil1 and Patil2 (2014) showed experimentally the effect of the number of turns and the flow rate on the amount of heat transfer in helical coiled tube heat exchanger. For different Helical coil used number of turns N=3.25, N=4 and N=5, it has been found that the water outlet temperature decreases with increase in the water flow rates, also as the number of turns of the helical coils increases from N=3.25 to N=5 the water outlet temperature increases at the same water flow rate.

In the present the influence of helical coil characteristic and the bath temperature on the overall heat transfer convection have been studied. Also, the outer and inner heat transfer coefficients in natural convection water bath to predict better correlations between the outer Nusselt and Rayleigh numbers have been investigated for the height of the coil using as the characteristic length .

EXPERIMENTAL SETUP :-

The test rig is designed and manufactured to fulfill the requirements of the test system for a free convection heat transfer. Figure (1) illustrates the schematic diagram of the experimental apparatus. A water tank has dimensions (60*60*120) cm, which manufactured from a galvanized steel sheet vessel and insulated by glass wool to reduce the heat losses. This tank is worked as a heating medium that filled with water and heated by three heaters each one with (6000 W) that located in the bottom of tank, two of them works on all time and the last one was used as needed to maintain a constant water bath at (70 and 90)^{0}C. The helicoidal pipes used in the test sections are made from copper pipes have a thickness(1mm),length (6m),outer diameters (12.5)mm and three pitches, the pipe length is measured before bending and the inner and outer pipe diameter is measured by a Vernier. The physical dimensions of this coils are given in table (1) and shown schematically in figure (2). The test section is putting vertically in the center of the rectangular tank and its two ends extend straight to exist from the side of the tank. Two thermocouples type (K) are used to measure the water flow inlet and outlet temperature and four thermocouples are used for controlling the water bath temperature, three are placed on the center of the coil and the other is placed outside the coil. The target fluid is pumped by using a centrifugal pump connected to a rot meter to obtain the desired mass flow. The volume flow rates used are (10, 15, 20, 25) l/min which correspond to Reynolds Number from 20930 to 53723. This Reynolds number ensures that the flow inside the coil is turbulent when compared to the critical Reynolds number of flow in helices as obtain in the following equation(**Ito 1959**) :

$$Re)_{critical}=2*10^4(d/D)^{0.32} \tag{1}$$

UNCERTAINTY :-

In this research, the estimation of the experimental uncertainty is based on the approach presented in **(Holman, 1988, Beckwith, 2006)**. The individual experimental uncertainties of the measured variables with the calculated are given in table (2). The analysis is dependent on the maximum percentage error for the parameters.

THEORETICAL ANALYSIS :-

The total amount of heat transferred from the water bath to the coiling pipe is:

$$Q=m.cp.(T_{out}-T_{in}) \tag{2}$$

If the total amount of heat transfer is calculate, the overall heat transfer coefficient could be calculated from:

$$U_o=q/(A_o.\ \Delta T) \tag{3}$$

where,

$$\Delta T = \frac{T_{bath} + \left(\frac{T_{in} + T_{out}}{2}\right)}{2}$$

The overall heat transfer coefficient can be described in terms of the thermal resistances for a cylindrical tube as shown in the relationship below:

$$U_o = \frac{1}{\left[\frac{1}{h_o} + \frac{r_o \ln\left(\frac{r_o}{r_i}\right)}{k} + \frac{r_o}{(r_i h_i)}\right]} \tag{4}$$

In above equation (4) the inside heat transfer coefficient can be calculated from the inside Nusselt number which based on the correlation below **(Rogers and Mayhew (1969)):**

$$Nu_i = 0.021\, Re^{0.85} Pr^{0.4} (r_i/R)^{0.1} \tag{5}$$

Therefore an iterative approach was used to determine the wall temperature and the inner Nusselt number simultaneously. All the physical fluid properties used in Reynolds, Rayleigh and Prandtl numbers are based on the mean film temperature and generated as a polynomial function of temperature as shown below:

$$\rho=a_o+a_1T^1+a_2T^2+a_3T^3+a_4T^4+a_5T^5 \quad \ldots(6)$$

$$\mu=b_o+b_1T^1+b_2T^2+b_3T^3+b_4T^4+b_5T^5 \quad \ldots.(7)$$

$$K=c_o+c_1T^1+c_2T^2+c_3T^3+c_4T^4+c_5T^5 \quad \ldots(8)$$

$$Cp=d_o+d_1T^1+d_2T^2+d_3T^3+d_4T^4+d_5T^5 \quad \ldots(9)$$

Where all the coefficients are given in table (3).
The inside heat transfer coefficient, hi, is then determined:

$$h_i = \frac{Nu_i . K}{2\, r_i} \tag{10}$$

The wall temperature, T_w, is then determined from:

$$T_w = \frac{q}{h_i A_i} + T_{bath} \tag{11}$$

An iteration procedure is repeated by using a computer program written in Fortran 90 until a convergence is obtained. The outside heat transfer coefficient is then calculated from the thermal resistance equation .

RESULTS AND DISCUSSION :-

The overall heat transfer coefficient is effected by the coil geometric and the boundary conditions. As the difference between the bath and target fluid temperature increase the amount of heat transfer (q) increase so that the overall heat transfer coefficient for the bath fluid temperature (90 ^oC) is higher than for the bath fluid temperature (70 ^oC) as shown in figs. (3) and (4). These figures show that the overall heat transfer coefficient increase with the increasing of the curvature ratio (d/D) and the coil pitch. In all figures it can be shown that the increase of the mass flow rate increases the overall heat transfer coefficient due to the decrease in the thermal resistance of the coil tube and then increase the amount of heat transfer to the cold fluid. As the coil diameter (D) increases, the effect of coil curvature on the flow decreases at a constant pipe diameter and the coil pitch. Hence, the centrifugal forces play a less important role in the flow characteristics, this mean transfer of thermal energy to the inner flow decreases as the coil diameter increases. The inner Nusselt number for three coil diameter (D=108mm, D=200mm, D=250mm) are shown in figures (5), The centrifugal force has also effected on the outer Nusselt number and the outer heat transfer coefficient as shown in figure (6). Figures from (7) to (10) show the influence of the coil pitch on the inner Nusselt number and outer heat transfer coefficient, as the coil pitch increase, the torsional force increase which increase the internal energy. According to the experimental data the outer Nusselt number has been correlated as a function of Rayleigh number by using the SPSS software with non- linear regression Analysis as shown below with the characteristic length is the coil height and (R^2=0.977).

$$Nu_o=0.009(Ra)^{0.419} \quad 1.76*10^9 < Ra < 1.15*10^{12} \qquad (12)$$

CONCLUSIONS :-

The conclusions resulted from pervious experimental data can be written as follows:

1- The inlet and outlet heat transfer coefficient and Nusselt number have directly proportional with each of the mass flow rate of the target fluid that flow in helical coil and the bath water temperature.

2- The decreasing of the coil diameter causes enhancement the heat transfer characteristic, so that the heat transfer coefficient and the Nusselt number for the curvature ratio (d/D=0.144) are higher than for the curvature ratio (d/D=0.078) and (d/D=0.0625).

3- The overall amount of heat transfer increase as the coil pitch increases.

4- The overall correlation equation has been developed for outer Nussel number as a function of Rayleigh number .

Table (1) : Physical Dimensions of the Test Coils

coil	d (mm)	D(mm)	N	d/D	P(mm)
1	12.5	108	17.5	0.115	12.5,25,37.5
2	12.5	200	9.5	0.0625	12.5,25,37.5
3	12.5	250	7.5	0.056	12.5,25,37.5

Table (2) : Uncertainty of measured variables and calculated parameters

Independent Variable	Uncertainties %
Pressure drop (Δp)	2
Flow rate(m)	2.5
Length (L)	0.05
Density (ρ)	0.1
Specific heat (cp)	0.1
Viscosity (µ)	0.1
Thermal conductivity (K)	0.01
Coil diameter (D)	0.01
Tube diameter (d)	0.01
Average bulk temperature (ΔT_b)	0.706
Average film temperature (ΔT_F)	0.706
Heat transfer coefficient (h_o)	$2.4167 *10^{-2}$
Nusselt number (Nu_o)	$4.3567 *10^{-3}$

Table (3) : Coefficient of the Fluid Properties

Prop-erties	**a_o,b_o,c_o,d_o**	**a_1,b_1,c_1,d_1**	**a_2,b_2,c_2,d_2**	**a_3,b_3,c_3,d_3**	**a_4,b_4,c_4,d_4**
CP	4209.73	2.07489	0.0367315	0.00019683	3.67419E-07
K	0.557952	0.00277901	3.14102E-05	2.59877 E-07	1.01454E-09
µ	0.00168434	4.58535E-05	7.0071E-07	5.68919E-09	1.87682E-11
ρ	1001.08	0.151211	0.000151913	6.46818E-05	3.61999E-07

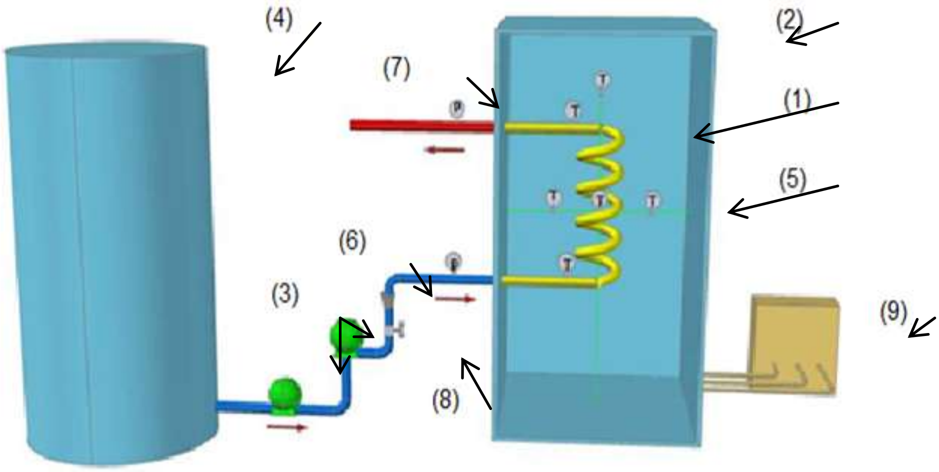

Fig. (1) : schematic diagram of the experimental apparatus.
(1) Test Section (Helical Pipe) , **(2)** Rectangular Tank, **(3)** Two series centrifugal pump, **(4)** Reservoir tank, **(5)** Thermocouple, **(6)** The volumetric flow meter, **(7)** Pressure tap, , **(8)** Valve , **(9)** Electric heaters.

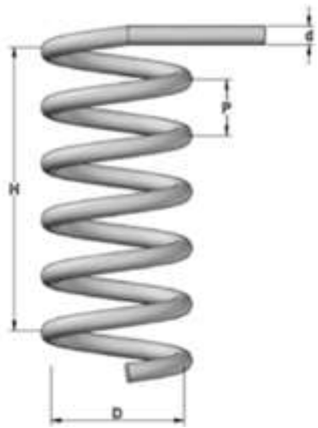

Fig. (2) : physical Dimensions of this Coil

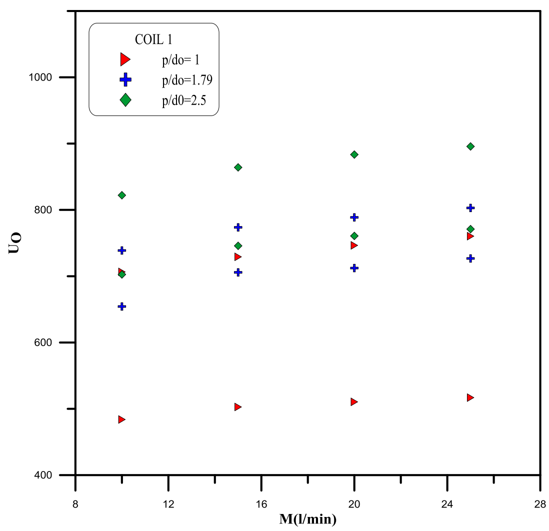

Fig. (3): Effect of the coil pitch on the overall heat transfer coefficient for coil 1

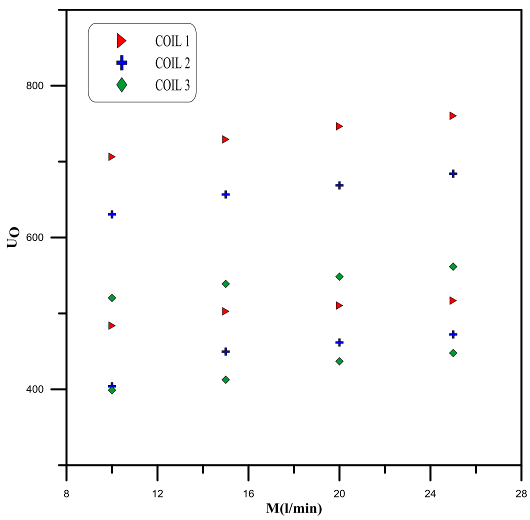

Fig. (4): Effect of the curvature ratio on the overall heat transfer coefficient

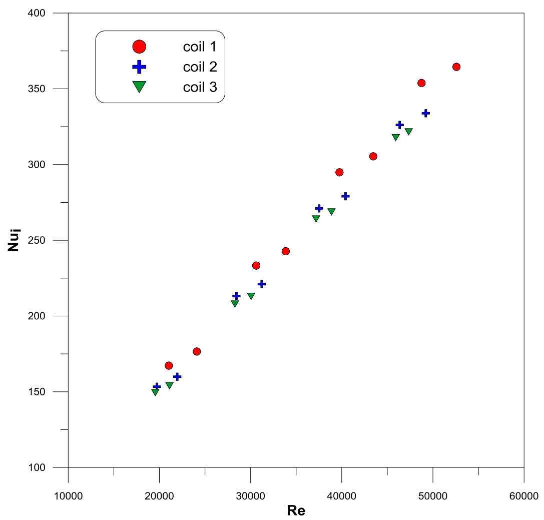

Fig. (5) : Inner Nusselt number versus Reynolds number for different Curvature ratio

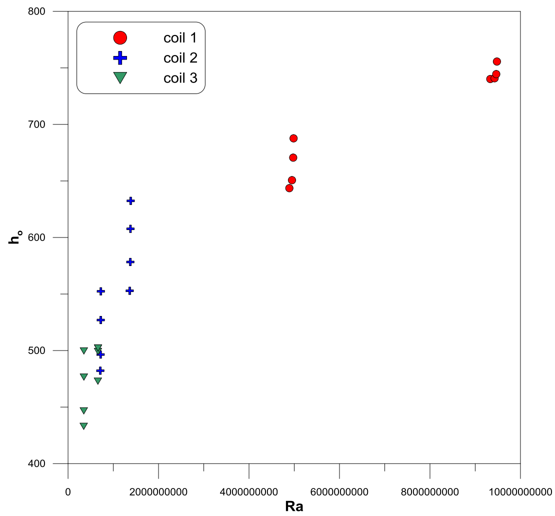

Fig. (6) : Outer heat transfer coefficient versus Rayleigh number for different curvature ratio

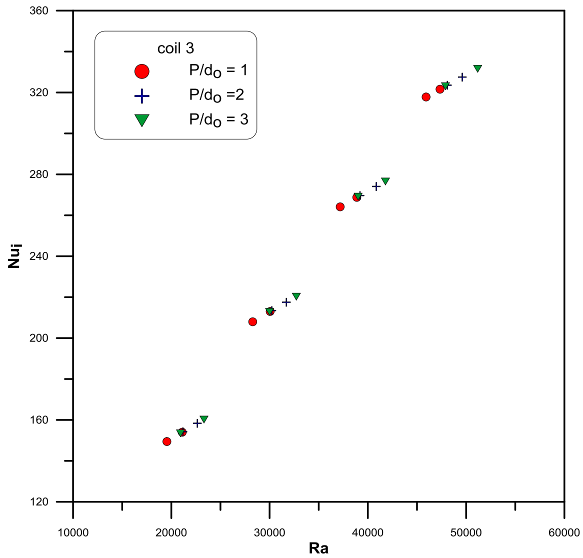

Fig. (7) : effect of the coil pitch on inner Nusselt number for coil 3

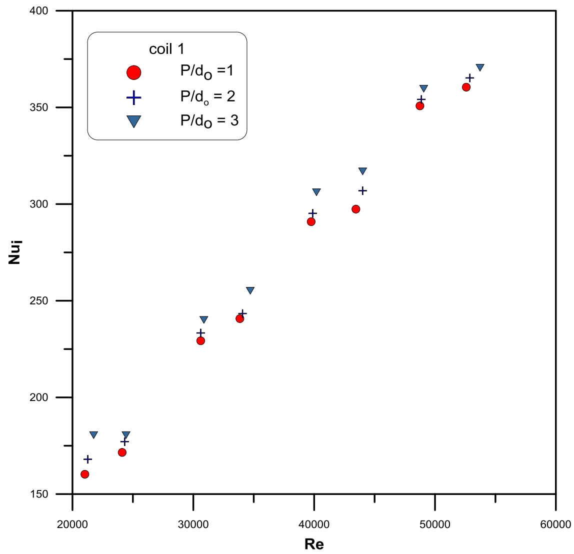

Fig. (8) : effect of the coil pitch on the inner Nusselt number for coil 1

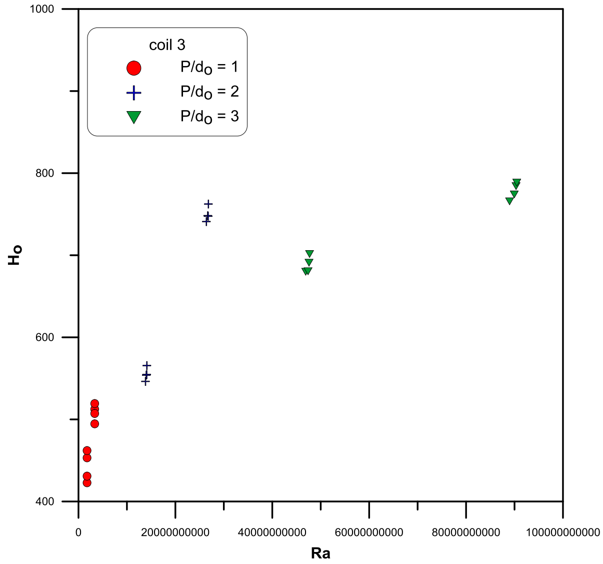

Fig. (9) : Effected of the coil pitch on the outer heat transfer coefficient for coil 3

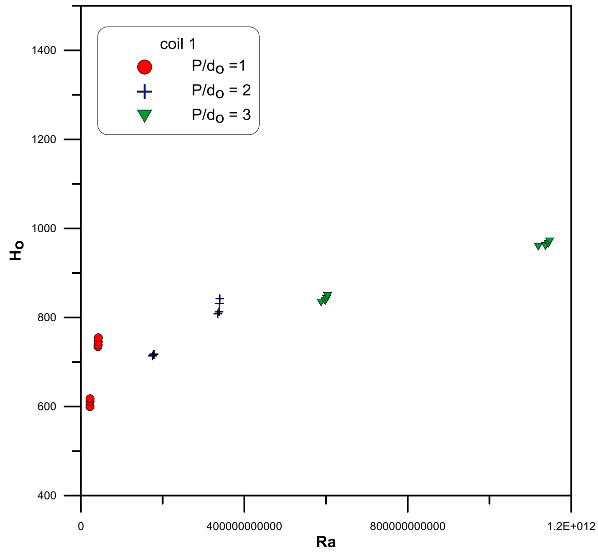

Fig. (10) : Effected of the coil pitch on the outer heat transfer coefficient for coil 1

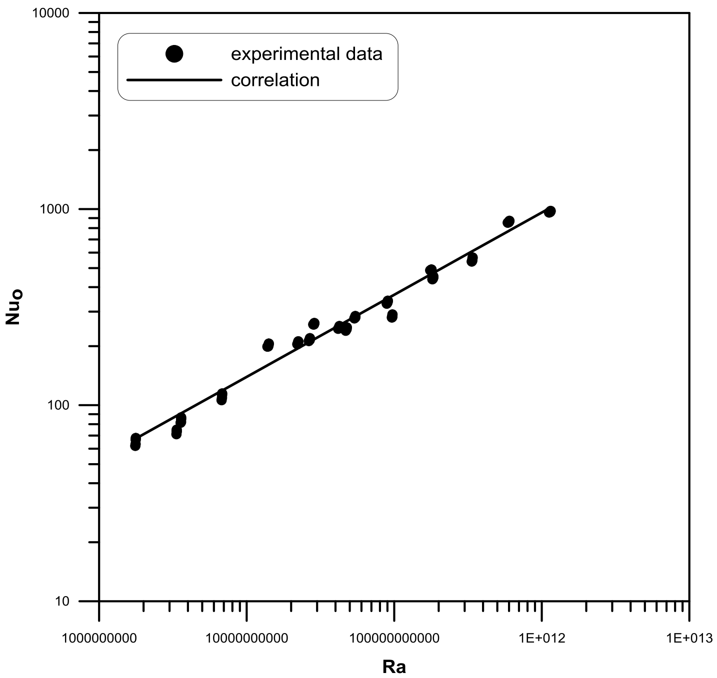

Fig. (11) : Average turbulent heat transfer correlation for vertical coils in water with coil height as a characteristic length

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