EXPERIMENTAL STUDY OF HEAT TRANSFER AND FLOW THROUGH ANNULAR TUBE WITH DISCONTINUOUS LONGITUDINAL WAVY FINS +

Deyaa Mohammed N. Mahmood *

Abstract :

In the present study the heat transfer and the flow characteristics for the annular side of concentric tube in tube heat exchanger was experimentally investigated. The single phase water was used as working fluid for both sides of the heat exchanger. The passive technique was used to enhance the heat transfer. The discontinuous longitudinally wavy fins was used for this purpose. The water flow rate of the annular side was ranged between (3.33×10^{-5} to 8.33×10^{-5} m^3/s), while the water flow rate of the tube side was kept at a constant level. The laminar flow was achieved in annular side for all water flow rate testes levels and the flow was turbulent for tube side. The heat transfer rate and the convective coefficient were augmented but this was accompanied by an increase in pressure drop and friction factor. The heat transfer rate was improved by a range of (15 to 25%). The friction factor obtained with finned tube was higher than that of the plain tube by (8 to 18%). The overall enhancement factor ranged between (1.11 to 1.19).

Keywords: **annular; heat exchanger; heat transfer; pressure drop; longitudinal wavy fin; enhancement**

دراسة عملية لانتقال الحرارة والجريان خلال الأنبوب الحلقي باستخدام زعانف طولية متموجة غير مستمرة

ضياء محمد نوري محمود

المستخلص:

في الدراسة الحالية تم تحري خصائص انتقال الحرارة والتدفق عمليا للجانب الحلقي لمبادل حراري نوع الأنبوب المتمركز في أنبوب. استخدم ماء أحادي الطور كمائع تشغيل في كلا جانبي المبادل الحراري. استخدمت تقنية (passive) لتحسين انتقال الحرارة. حيث استخدمت زعانف طولية متموجة غير مستمرة لتحقيق هذا الغرض. معدل تدفق الماء في الجانب الحلقي تراوح بين (3.33×10^3 إلى 8.33×10^3 ^{3}م/ثا)، في حين تم إبقاء معدل تدفق الماء في جانب الأنبوب عند مستوى ثابت. حصل جريان طباقي في الجانب الحلقي لجميع مستويات معدل تدفق الماء وكان الجريان مضطرب لجانب الأنبوب. ازداد معدل انتقال الحرارة ومعامل الحمل الحراري ولكن ذلك كان مصحوب بزيادة في هبوط الضغط ومعامل الاحتكاك. تم تحسين معدل انتقال الحرارة بمدى تراوح بين (15 إلى 25%). معامل الاحتكاك الذي حصل عليه للأنبوب المزعنف كان اكبر من الأنبوب الاعتيادي بمقدار تراوح بين (8 إلى 18%). معامل التحسين الكلي تراوح بين (1.11 إلى 1.19).

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* **Assistant Lecturer / Technical Instructors Training Institute / Middle Technical University**

Nomenclature:

A surface area, m^2
C_p specific heat at constant pressure, J/kg K
d inner tube diameter, m
D outer tube diameter, m
D_H hydraulic diameter, m
f friction factor
Nu Nusselt number
k thermal conductivity, W/m K
L tube length, m
$\dot{m}$ mass flow rate, kg/s
Pr Prandtl number
Q heat rate, W
Re Reynolds number
T temperature, K
u mean velocity, m/s
$\dot{V}$ volumetric mass flow rate, m^3/s
Δp pressure drop, pa
ρ density, kg/m^3
µ viscosity, pa.s

subscripts

b bulk
c cold water
e enhanced
h hot water
in inlet; internal
out outlet; external
p plain
s surface

Introduction:

The thermal performance of the heat exchanger can be improved by heat transfer enhancement techniques. It can be divided in two groups; passive and active, the passive method without stimulation by the external power such as a surface coating, rough surfaces, extended surfaces, turbulent swirl flow devices. While the active method required extra external power source for example mechanical aids, surface fluid vibration, injection and suction of the fluid, and use of electrostatics fields [1]. The annular region has relatively low heat transfer coefficient because it is often laminar flow, so it important to focus our interest in improving it. The study referred to a number of studies performed on the annular region only. Zahang et al. [2] experimentally investigate the thermal performance of the shell side of double pipe heat exchanger. The performance was improved by using helical fins and some vortex generators. Lin et al. [3] present experimental and numerical study of laminar heat transfer of annular tubes with sinusoidal wavy fins under constant heat flux boundary conditions. Joshi and Kriplani [4] have been carried out experimental investigation to study the effect of inner twisted tape and annular inserts on the performance of double pipe heat exchanger. Annular protrusions are used to augment the heat transfer by creating turbulence in the fluid flow. yu and Tao [5] experimentally studied the pressure drop and heat transfer coefficient characteristic in annular tube by using waves like longitudinal fins. Syed et al.[6] numerically study the convection laminar flow heat transfer in double pipe heat exchanger with straight longitudinal fins. In his other numerical study he investigate optimal configuration of finned annuls in concentric tube in tube heat exchanger with trapezoidal fins augmented to the outer surface of the inner pipes [7]. The triangular fins in the annular of double pipe was also numerically simulated using finite element method [8]. Ashaq et al [9] numerically simulated laminar convection in an annulus with triangular fins of different height. Iqpal et al. [10] investigate optimal configuration of fins annulus with parabolic fins. Pachegaonkar et al. [11] experimentally analysis the heat transfer and pressure drop characteristics of double pipe heat exchanger by using annular twisted tape insert. The results revealed increased in both heat transfer coefficient and pressure drop modified tube compared to that of plain tube. Sahiti et al [12] experimentally investigated the heat transfer enhancement for double pipe heat exchanger with pin fins. It was found that by a direct comparison of Nusselt number and friction factor no conclusion regarding the relative performances could be made. In the present study the enhancement technique employed is discontinuous longitudinal wavy fins welded to the outer surface of the inner tube. The fins distributed as pair in horizontal and vertical position alternately along the tube. The purpose of this arrangement is augmented the convective heat transfer in addition of the wavy shape by disrupting the boundary layer due to repeated changes in the surface geometry.

Experimental Apparatuses:

The experimental system shown in figure (1) is composed of double pipe heat exchanger which is the test section, water tank, centrifugal water pump, electrical heater and measurement devices. The hot water flows in the annular side of the heat exchanger, while the cold water flows in the inner pipe in counter flow. The inner tube of heat exchanger was made of copper with length of (2000 mm), thickness (1.06 mm) and external diameters is (19 mm). While the outer tube with length of (2000 mm), thickness (1.27 mm) and external diameters is (54 mm). The heat exchanger was insulated firmly with glass wool insulation. The water tank, with capacity of (0.02 m^3) was used to control the hot water flow rate by using bypass loop. The centrifugal water pump was used to circulate the hot water through the annular side of heat exchanger. While the over head water tank with capacity of (1 m^3) was used to supply cold water to the inner tube of the heat exchanger. The water in the hot loop was heated with electrical heater of (2 kW) power. The temperature around the system was measured by using a K type thermocouple. The thermocouples were installed on the inlet and exit ports of the hot and cold water streams. The hot water flow rate was measured by using rotameter, while the cold water flow rate was measured by using known volume container and a stop watch. The pressure drop for the annular side was measured by using differential pressure gauge. All the measuring devices have been calibrated before the installation in the experimental test rig. The photograph of the experimental system is shown in figure (2).

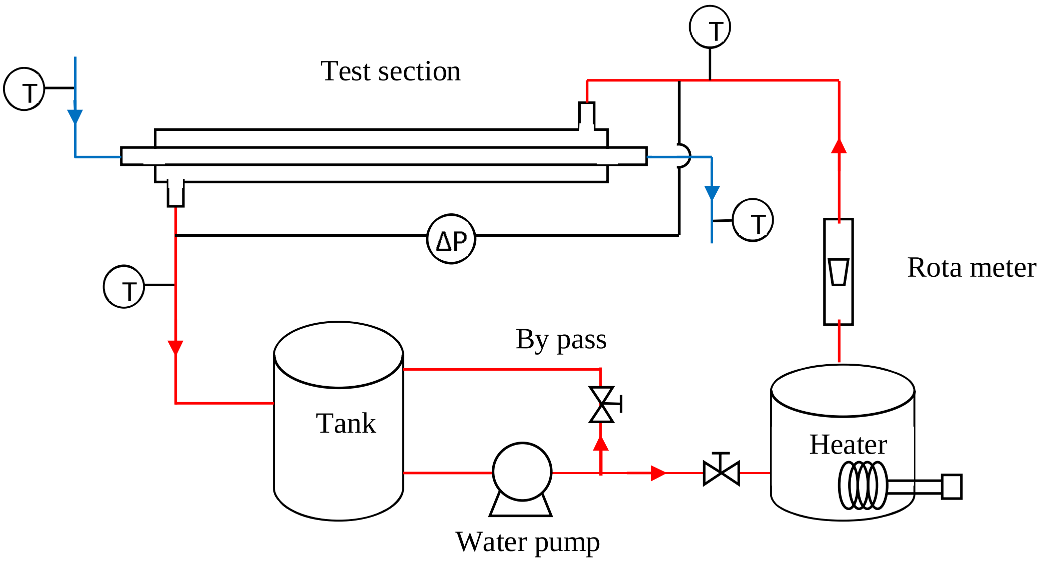

Figure (1): Schematic Diagram of Experimental System Set-up

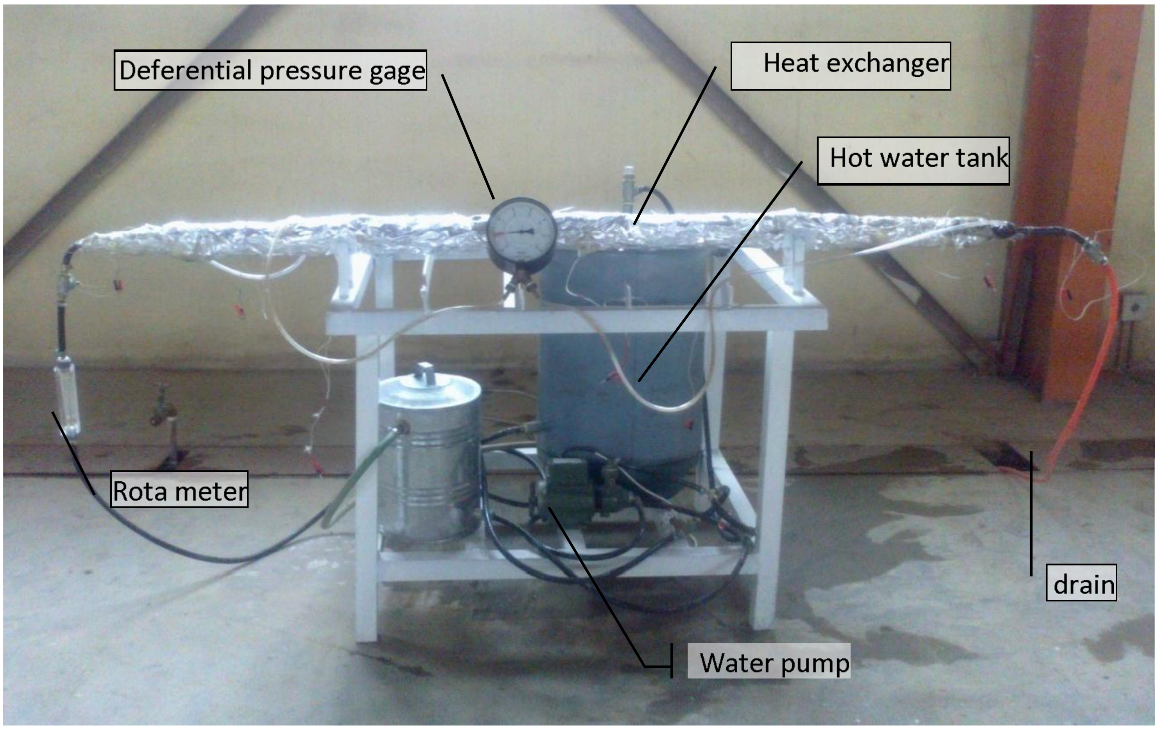

Figure (2): Photographic view of the Experimental Test Rig.

1. <u>Fin Geometry and Arrangement</u>:

The photographic view of the fin used in the present experimental work is shown in figure (3). The fins were made of copper. The detailed physical dimensions of the fin geometry are presented in table (1) and figure (4). The fins were welded longitudinally to the external surface of the inner tube of heat exchanger, which is distributed as a pair in horizontal and vertical positions alternately along the tube as illustrated in figure (5). There will be small dead zones in the bottom of each corrugation which will decrease the flow area giving a higher mean velocity. It is known that secondary flows induced by the corrugations assist the augmentation in addition to the partial restarts of the boundary layers [13].

Figure (3): Photographic view of wavy fins used in the present work

Table (1): Fin Physical Dimensions

Dimension specification	Value
Fin type	wavy
Fin thickness	0.6 mm
Fin length	88 mm
Fin height	12 mm
Number of fins	36 fins
height of corrugation	5 mm
Pitch (wave length)	8 mm

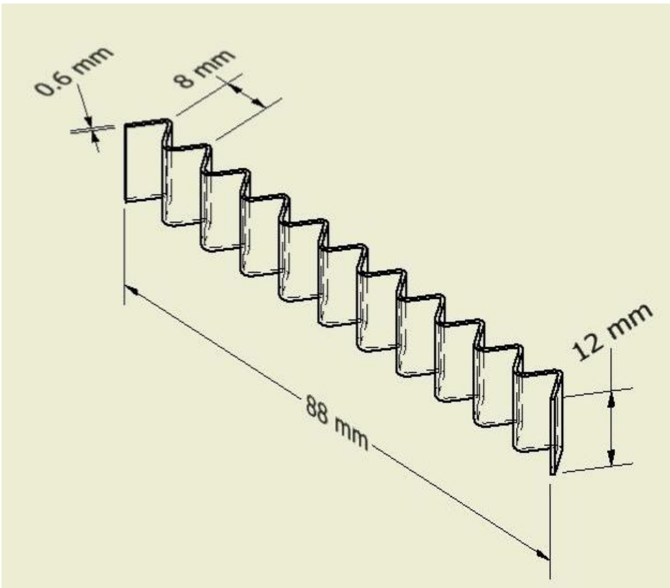

Figure (4): schematic sketch of typical wavy fins used in the present work

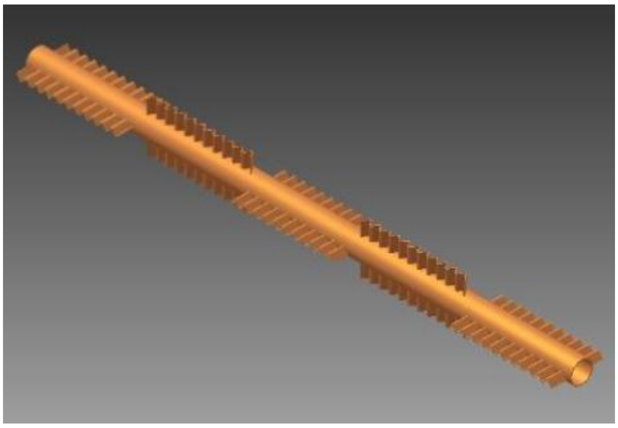

Figure (5): inner tube with discontinues longitudinal wavy fin for the first five pears

Data Reduction:

The experimental measured data obtained from the present test rig are used to determine the following parameters
The heat transfer rate of the hot water for the annular side calculated from

$$Q_h = \dot{m}\, c_{p,h}(T_{h,in} - T_{h,out}) \quad \text{.......(1)}$$

Where $\dot{m}$ represent the mass flow rate of the hot water estimated as $\dot{m} = \rho\, \dot{V}$
$T_{h,in}$ and $T_{h,out}$ represent the inlet and outlet hot water temperature respectively.
The heat transfer rate of the cold water for the tube side calculated by

$$Q_c = \dot{m}\, c_{p,c}\, (T_{c,out} - T_{c,in}) \quad \text{.......(2)}$$

The Nusselt number and convection heat transfer coefficient for the turbulent flow in the tube side for heating process was estimated from *Dittus-Boelter* correlation, [14] as the following:

$$Nu_d = \frac{h_{in}\, d_{in}}{k} = 0.023\, Re_d^{0.8}\, Pr^{0.4} \quad \text{.......(3)}$$

Where $Re_d = \frac{\rho\, u\, d_{in}}{\mu}$ and $Pr = \frac{\mu\, c_p}{k}$

By using equation (2) the internal surface temperature of the inner tube can be estimated

$$Q_c = h_{in}\, A_{in}\, (T_{si} - T_{b,c}) \quad \text{.......(4)}$$

Where A_{in} represent the internal surface area of the inner tube calculated as

$$A_{in} = \pi\, d_{in}\, L$$

And $T_{b,c}$ represent the bulk temperature of the cold water estimated from the mean values of the inlet and exit measured temperatures of the cold water.

$$T_{b,c} = \frac{T_{i,c} + T_{o,c}}{2}$$

By using equation (1), the annular side convection heat transfer coefficient can be estimated

$$h_{out} = \frac{Q_h}{A_{out}\,(T_{b,h} - T_{so})} \quad \text{.......}(5)$$

Where A_{out} represent external surface area of the inner tube calculated as

$$A_{out} = \pi\, d_{out}\, L$$

The external surface temperature (T_{so}) was obtained experimentally.
$T_{b,h}$ represent the bulk temperature of the hot water estimated from the mean values of the inlet and exit measured temperatures in the hot water flow path.

$$T_{b,h} = \frac{T_{i,h} + T_{o,h}}{2}$$

The thermodynamics and transport properties for the water based on bulk temperature.
The hydraulic diameter of the annular tube calculated from

$$D_H = D_{in} - d_{out}$$

The friction factor for the annular side calculated from equation (6) [13].

$$f = \frac{\Delta p}{\left(\frac{L}{D_H}\right)\left(\rho\, \frac{u^2}{2}\right)} \quad \text{.......}(6)$$

Where Δp represent the Pressure drop obtained from the experimental measured data and (u) represents the mean velocity water flow rate calculated as.

$$u = \frac{\dot{V}}{\frac{\pi}{4}\left(D^2{}_{in} - d^2{}_{out}\right)}$$

<u>Results and Discussion</u>:

The experimental heat transfer coefficient was compared with Mahdi et al. [14] results as shown in figure (6) for plain tube. The comparison shows good agreement between the results. Figure (7) shows the variation of heat transfer rate with Reynolds number for plain and modified wavy finned tubes. The heat transfer rate increased with increasing of Reynolds number. It is also shows that curve trend of the finned tube is slightly higher than that of plain tube. the explanation of this behavior is the turbulent intensity more increased in enhanced tube compared with the plain tube. The wavy fins improved the heat transfer rate by (15 to 25 %).

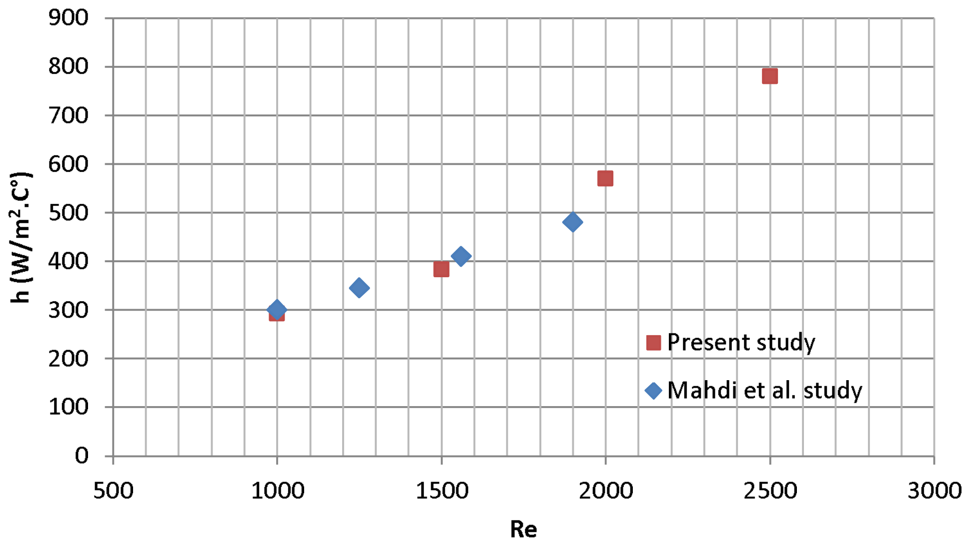

Figure (6): comparison of experimental heat transfer coefficient for plain tube.

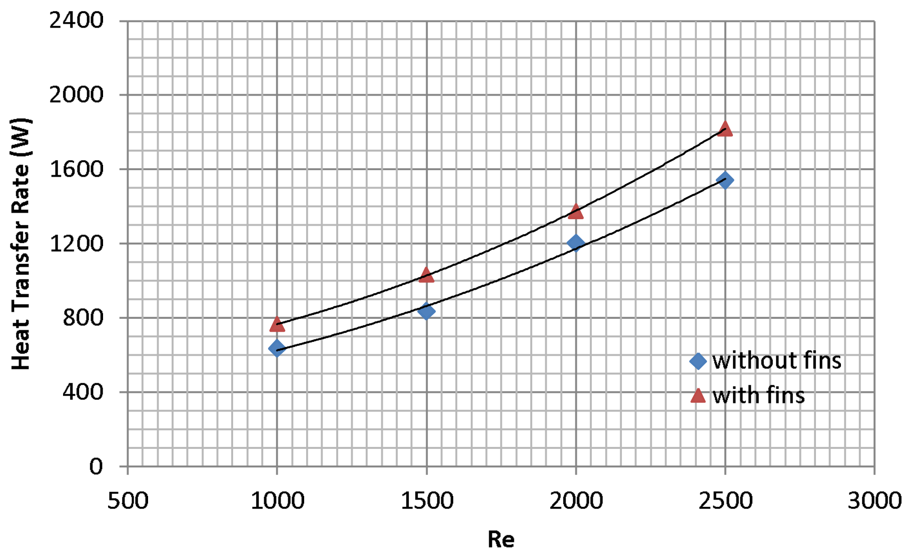

Figure (7): variation of the heat transfer rate with Reynolds number

The variation of the Nusselt number with Reynolds number is illustrated in figure (8). It can be seen Nusselt number increased with increasing of Reynolds number. Nusselt number for the modified tube was greater than that of plain tube. Figure (9) shows the variation of the convection heat transfer coefficient for the annular side of heat exchanger with Reynolds number, which is increased with increasing of Reynolds number. The heat transfer coefficient for the modified tube is greater than that of the plain tube, it is improved by the value of (15 to 25 %). This is attributed to the enhancement in fluid mixing and increasing of turbulent intensity due to effect of employing of fins and there distributed method.

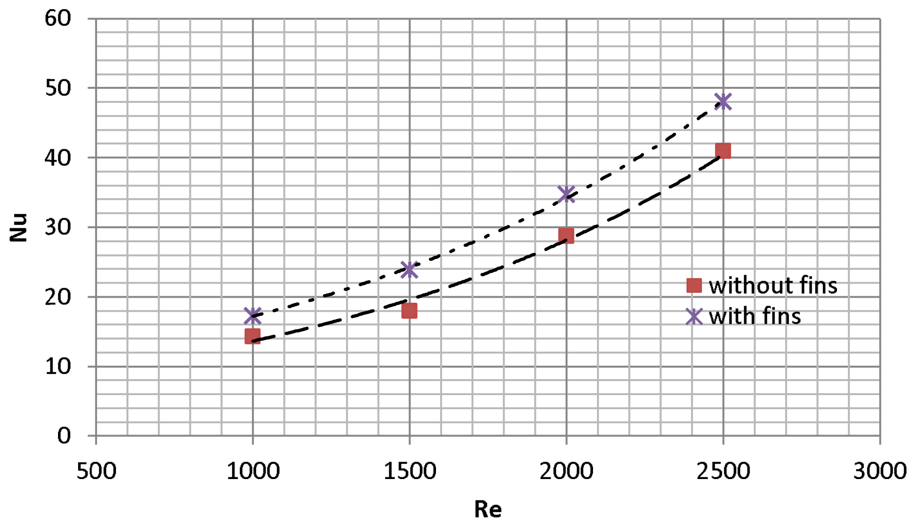

Figure (8): variation of Nusselt number with Reynolds number for the annular side.

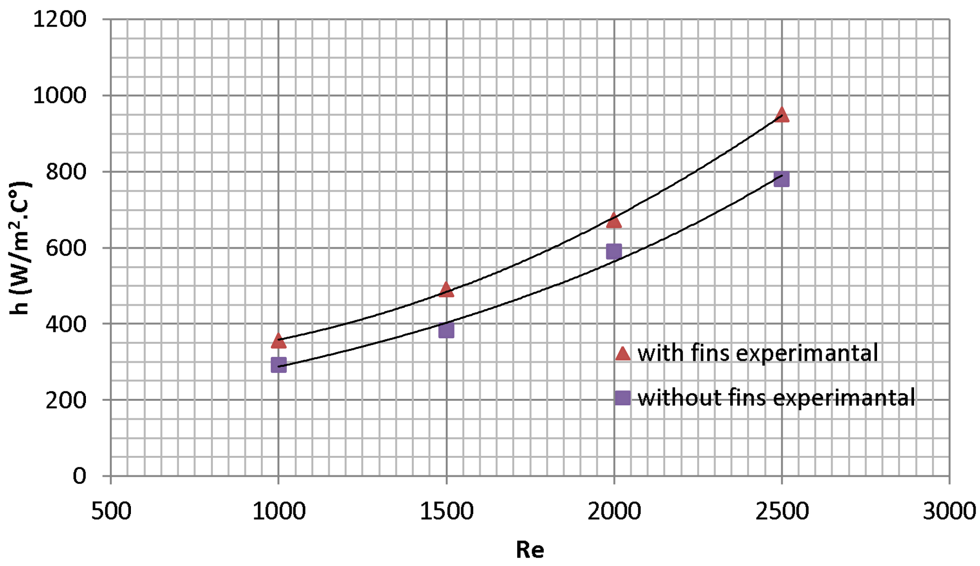

Figure (9): variation of the annular side heat transfer coefficient with Reynolds number

Figure (10) shows the relationship between the friction factor and Reynolds number for plain and enhanced tubes. The friction factor decreases with increasing of Reynolds number. Both curves (with and without fins) reveal the same trend. The friction factor of modified finned tube is greater than that of plain tube by (8 to 18 %). This is because of extra resistance to water flows for the annular side which is created by the fins. Figure (11) illustrates the variation of the pressure drop with Reynolds number, the pressure drop increased with increasing of Reynolds number for both the plain and enhanced tubes. The pressure drop of the enhanced tube was greater than that of plain tube by the range of (11 to 15 %), for the same reason mentioned above that concerning the friction factor can be given here.

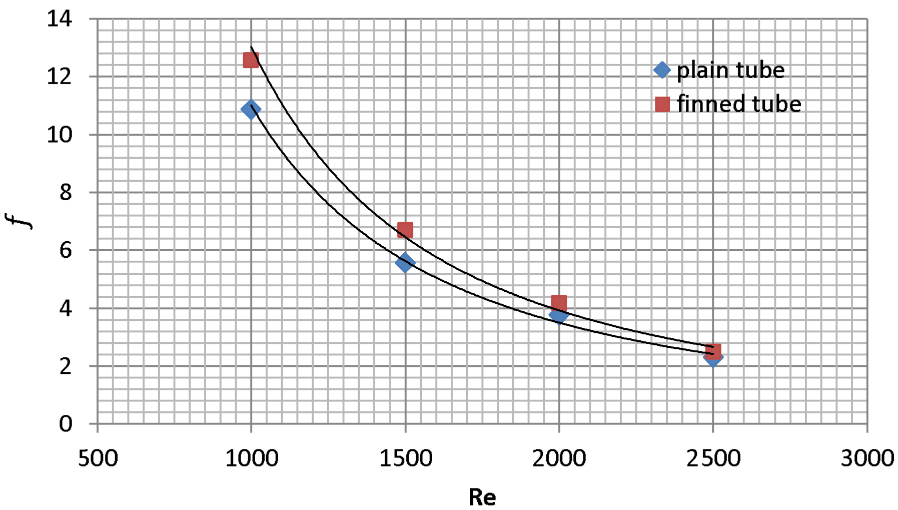

Figure (10): variation of the friction factor with Reynolds number

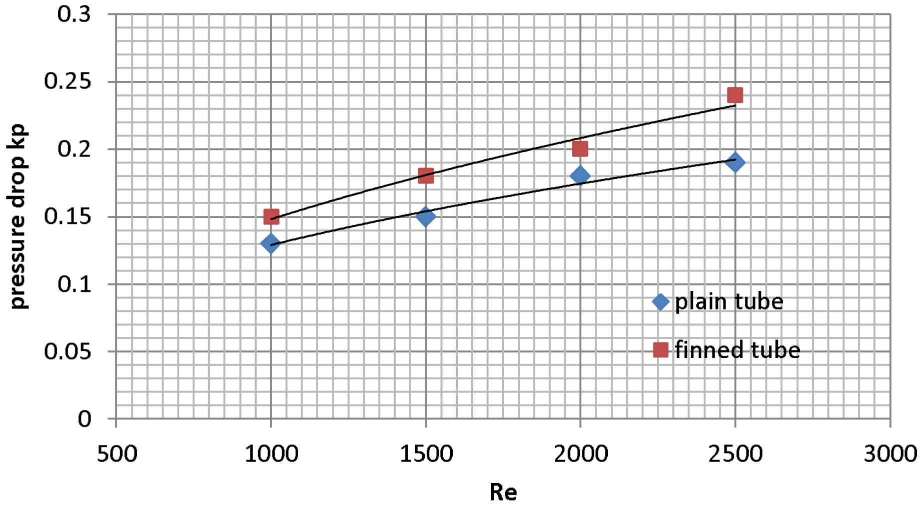

Figure (11): variation of the pressure drop with Reynolds number.

In order to have a comprehensible assessment of the performance of the heat exchanger, both thermal and flow characteristics has been compared. Table (2) presents the enhancement ratios concerning different water flow rate in the four tests are; heat transfer rate ratio (Q_e/Q_p), Nusselt number ratio (Nu_e/Nu_p) in comparison with friction factor ratio (f_e/f_p) and it is also shows the over enhancement factor. The enhancement ratios of heat transfer rate and Nusselt number were within the range (1.25 to 1.15) and (1.26 to 1.14) respectively, while the friction factor ratio increased from (1.11) to (1.20). The enhancement factor (EF) was calculated as the ratio of the measured Nusselt number for the modified finned tube to the Nusselt number for the plain tube of double pipe heat exchanger. The EF was within the range of (1.11) to (1.19).

Table (2): enhancement consideration

Test No.	Re	Q_e/Q_p	$\frac{Nu_e}{Nu_p}$	$\frac{f_e}{f_p}$	$\frac{Nu_e}{Nu_p}/\left(\frac{f_e}{f_p}\right)^{\frac{1}{3}}$
1	1000	1.25	1.26	1.18	1.19
2	1500	1.18	1.20	1.14	1.14
3	2000	1.16	1.15	1.10	1.11
4	2500	1.15	1.14	1.08	1.11

Conclusion:

The present experimental study of the heat transfer enhancement and flow characteristics for the annular tube of heat exchanger was carried out. The discontinuous longitudinal wavy fins was used to augment the heat transfer rate, it is improved by the rang of (15 to 25%). The convective heat transfer coefficient enhanced by average value of (20%). On the other hand the friction factor for finned tube was higher than that of the plain tube by (8 to 18%). Although the heat transfer rate is much higher at high water flow rate but the corresponding pressure drop and friction factor is higher too. The overall enhancement factor was ranged between (1.11 to 1.19)

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