Assessment for Heat Transfer Enhancement in Heat Exchangers by Using Dimples geometry

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Abstract:-

The pressure drop and heat transfer characteristics of a multi dimpled tubes heat exchanger were studied experimentally and numerically. Two different models of heat exchanger (plain and circular dimpled tubes) have been fabricated and tested. Instrumented heat exchangers were building up to cover the experiments for the selected range of Reynolds number (7000-18000). In the numerical study a software ANSYS (Fluent 14.5) with a Shear Stress Transport (SST) turbulent model have been used to calculate pressure drop and heat transfer coefficient for three models of heat exchangers(plain, circular dimpled and triangle dimpled tubes). The experimental results show that Average Nusselt number and the Pressure drop for the model of circular dimple tubes is higher than that for the plain tubes while the numerical results show that the triangular dimples have better results than the others. Comparisons between the experimental and numerical were done and show a good agreement between them.

Keywords: Heat exchanger, dimple, Nusselt Number, friction factor, triangle, circular.

1-Introduction

The technique of the heat transfer augmentation aims to increase convective heat transfer by reducing the thermal resistance in a heat exchanger. Use of heat transfer enhancement techniques lead to increase in heat transfer coefficient but with increase in cost and pressure drop. So, while designing a heat exchanger using any of these techniques, analysis of heat transfer rate and pressure drop has to be done [4].Rough surfaces are generally surface modifications that promote turbulence in the flow field, primarily in single-phase flows, and do not increase the heat transfer surface area , where they are employed to enhance heat transfer in flows both inside tubes and outside tubes, rods, and tube bundles [6]. Dimples have found their application in heat transfer augmentation techniques. In comparison, the use of dimples has also shown to significantly improve heat transfer coefficient with lesser pressure drop penalties [7]. Most of the researchers have studied the effect of dimples geometry, on the heat transfer enhancement numerically. Gururatana (2012) [8] presented a finite difference simulation for a micro type channel heat sink bounded by a dimpled surfaces, the rectangular shape was considered as a micro channel. The results show that the enhancement in heat transfer for the dimpled micro channel heat sink is helpful when the Reynolds number is more than 125. Heat transfer enhancement in mini-channel heat sinks with dimples was studied numerically by Tang, et al. (2013) [3] show that the dimple surface presents the highest performance in the heat transfer enhancement. Pisal and Ranaware (2013) [5] performed an experimental and numerical investigation using two different types of dimples circular (spherical) dimples and oval (elliptical) dimples. The results show that factor for the oval type dimpled plate was larger than that of the circular dimpled plate for all cases.

A new process surface designs were developed by Kukulka and Smith(2013)[2] which produce performance enhancement on the outside of the dimpled tubes for Reynolds numbers up to 215,000. Ghanim [1] study the thermal performance of a dimpled tube heat exchanger. The study concentrated on the effect of the some geometric dimensions of dimples on the heat transfer and pressure drop across a line type heat exchanger. The results show that Average Nusselt number for the model of four circular dimples per periphery is about 38.5 % higher than that for plain model while the pressure drop in circular dimple heat exchanger is about 69 % higher than that for plain model. The aim of the present work is to study the effect of using different dimples geometry as a method for heat transfer enhancement in a multi tube heat exchanger experimentally and numerically.

2-Experimental apparatus

The schematic diagram of the test rig of the present work is shown in Fig. 1 which consists mainly from a subsonic wind tunnel (duct) having a Parallel test section manufactured from clear acrylic, axial ventilation 3 phase motor (Power =2.2 kW), and the measurement devices.

A rectangular sheet copper material of conductivity (364 W/m.^{o}C) were selected for fabrications the heat exchanger tubes. A number of iron dies were manufactured and used for inserting the dimples as easy as possible. The dimples were placed on the outer surface of the copper sheet by using the die and the hand puncher. Two models of heat exchangers including plain and circular dimples were fabricated and tested in the present work as shown in Table. 1. The tubes have been prepared to be staggered heat exchangers and have an equal longitudinal and transverse pitch of tube. Fig. 2 shows the shapes and the dimensions of heat exchanger.

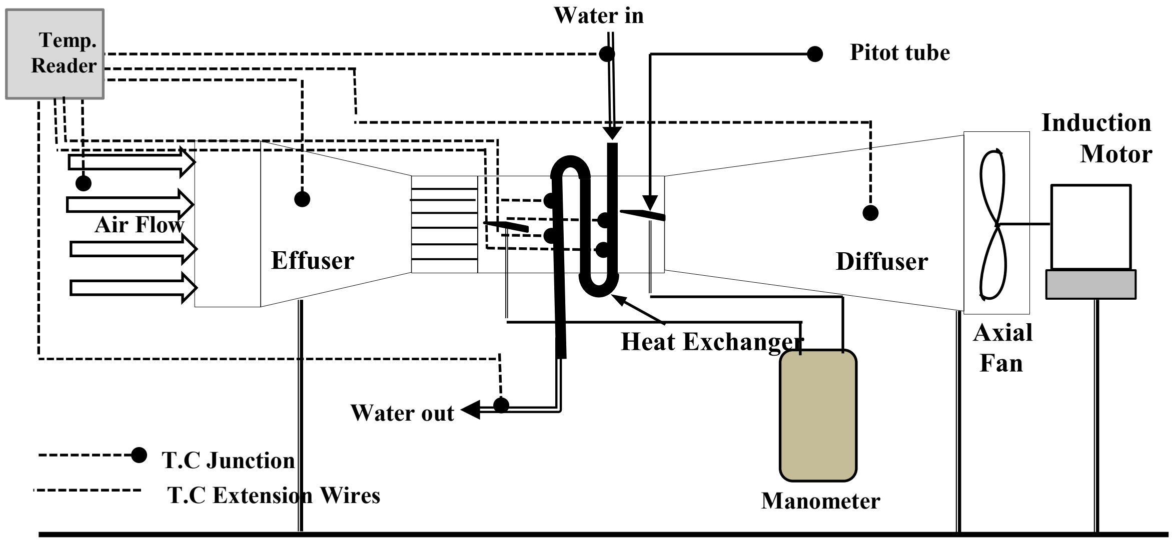

Fig 1. Schematic Diagram of the Experimental Rig

Many calibrated thermocouples of type (K) were used with a digital thermometer having an accuracy of ± 0.5 % of full scale division for the range of (0 - 999) ^{o}C, to measure the temperatures by connecting the thermocouples to digital thermometer in parallel by leads through a selector switch. The thermocouples were distributed within the rig and the heat exchanger to measure the inlet and outlet temperatures of the air at entry and at exit of the test section. Fifteen thermocouples were mounted on the heat exchanger surface, connected around the pipe, and distributed to measure the temperature of the hot surface, where each five thermocouples were distributed on the inlet, middle, and outlet of the heat exchanger. Static ellipsoidal nosed Pitot static tube has been used to measure the static pressure in the tunnel while a portable manometer has been used to measure the differential pressure. A digital anemometer vane-type (model AR826) has been used to measure the average air velocity at entry and exit of the test section.

As a different value of air velocity through the duct is required then an inverter used to control fan motor velocity. For a specified type of heat exchanger, the room temperature is adjusted in the front panel to be 25^{o}C then the water in the tank is heated to 50°C.

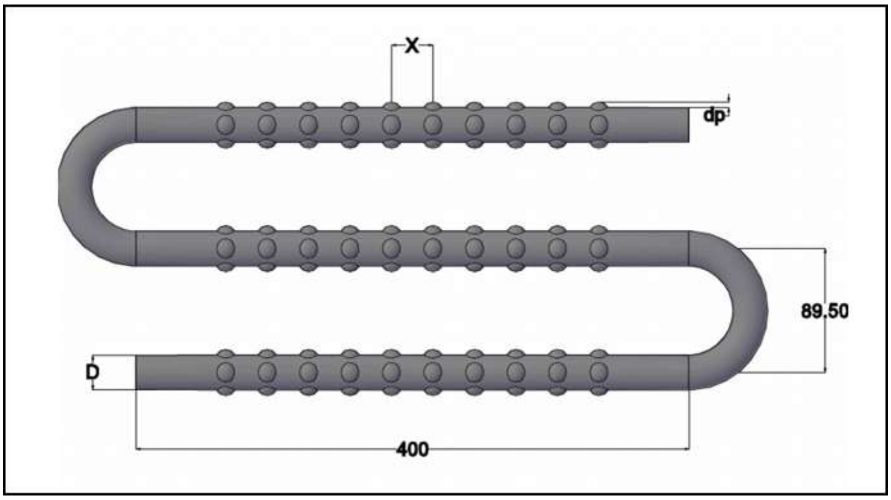

Fig 2. The Geometry of the Circular Dimple Heat Exchanger

A digital differential manometer is switched on to measure the pressure drop (ΔP) and the motor induction is switched on to inhale air inside the duct through the test section. The water pump is switched on to push the water through the heat exchanger with aid of the control valve to adjust the specified water flow rate. For each run, the air velocity and water flow rate are measured every 10 minutes to be sure of their stability. The thermocouples readings are taken also at the same period until the reading becomes constant, a final set of measurements are then recorded. Checking the water delivery by heating the water at specific temperature and the time required to reach this temperature is determined which represents the time of experiments. For each run, the inlet and outlet air and water temperature.

The pipe surface temperature, and the pressure drop (ΔP) between the inlet and outlet were measured.

3-Data Analysis

The rate of heat transfer from the hot fluid be equal to the rate of heat transfer to the cold one [9]:

$$Q = \dot{m}_a C_{pa}(T_{a,e} - T_{a,i})$$
$$Q = \dot{m}_w C_{pw}(T_{w,e} - T_{w,i}) \quad (1)$$

The rate of heat transfer to or from an isothermal surface is written as

$$Q = h_{av} A_s (T_{s,av} - T_\infty) \quad (2)$$

Then:

$$h_{av} = \frac{\dot{m}_w C_{pw}(T_{w,e} - T_{w,i})}{A_s(T_{s,av} - T_\infty)} \quad (3)$$

in which

$$A_s = A_{pipe} + A_{dimple} \quad (4)$$

Average Nusselt number can be calculated as

$$Nu_{av} = \frac{h_{av} D}{k_a} \quad (5)$$

While Reynolds number is calculated as

$$Re = \frac{\rho_a V_{max} D}{\mu_a} \quad (6)$$

The maximum velocity is determined from the following equation [9],

$$V_{max} = \frac{S_T}{S_T - D} V \quad (7)$$

Overall enhancement ratio (thermal performance) is defined by the following expression [3];

$$TH = \frac{Nu/_{Nu_o}}{\left[f/_{f_o} \right]^{0.3}} \quad (8)$$

Where $Nu_\circ$ and $f_\circ$ are results of the average Nusselt number and friction factor of the plain channel. Several correlations, all based on experimental data, have been proposed for the average Nusselt number for cross flow over tube banks. More recently the average Nusselt number is calculated as[9],

$$Nu_\circ = 0.7(0.27 Re^{0.63} Pr^{0.36} (Pr/Pr_m)^{0.25}) \quad (9)$$

Pr_m is the mean prandtle number which is evaluated at the arithmetic mean temperature of the fluid The pressure drops over the test section in the model were measured. The pressure drop can be measured by using the following relation [6]:

$$\Delta P = f \frac{\rho V_{max}^2}{2} N \quad (10)$$

4-NUMERICAL SIMULATION

Numerical calculations were conducted by simulating three-dimensional air flow and heat transfer over plain and dimpled tube heat exchanger configurations. The CFD modeling, simulation and post processing are carried out in an ANSYS 14.5, Workbench environment with an ANSYS system of fluid flow (Fluent).

The Geometry

A three dimensional heat exchanger pipe was simulated by using AutoCAD software and preparation for simulation by using ANSYS Design Modeler software. The specification and the dimension of the dimple tube heat exchanger is shown in Fig.3 while Tables. 1 and 2 shows the specification and dimensions of considered cases under study. The governing equations solved for the flow field are the continuity (mass conservation), the Navier-Stokes equations of motion (momentum conservation) and the energy equations in three dimensions for the fluid domain (air). Defining the Models and material properties. To run the cases, the model properties must be set. Model properties include the internal ANSYS-FLUENT solver settings like air and thermal properties, as well as model operating conditions and grid boundary conditions must be defined. The following settings were used to build the model in ANSYSFLUENT, three dimension, steady state, enabling energy. Properties that can be specified in this section are density, viscosity, specific heat, and thermal conductivity.

Boundary Conditions

The heat exchanger models were simulated for different inlet free stream air velocities of 7,6,5,4,and 3 m/s at 25 °C through the duct and inlet water velocity 0.04 m/s at 50°C inside the pipe for validation of Nusselt number (Nu) and friction factor (f).No slip condition was applied to the outer tube. Free stream air properties were specified as viscosity (μ) 1.798 ⟦10⟧^(-5) kg/m-s Specific heat (Cp) 1006.43 J/kg-K, and thermal conductivity (K)0.0242 W/m.K as shown in Fig. 4.

Mesh Generation

Tetrahedral elements were used to mesh the flow domain because it is more flexible in representing complex geometry boundaries [8]. A mesh independency test was carried out by varying the number of elements from 4.5 to 5 million. Six types of mesh density used to study mesh independence as shown in Table.3

Table 1. The geometric properties Specifications

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Heat exchanger	Total dimple number	(dp) (mm)	(X) (mm)	D(mm)
Plain pipe	-	-	-	25
Circular dimple pipe	120	4	30	25
Triangle dimple pipe	120	4	30	25

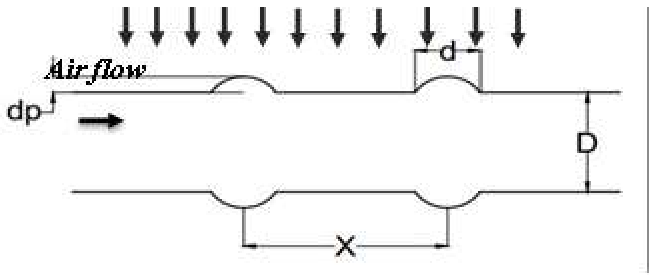

Table 2. The Dimensions the duct and the heat exchanger

Parameter	Symbol	Value
External Tube diameter	OD	25 mm
Internal Tube diameter	ID	21 mm
No. of tube	N	3
Duct width	W_D	20 mm
Duct height	H_D	110 mm
Duct length	L_D	300 mm

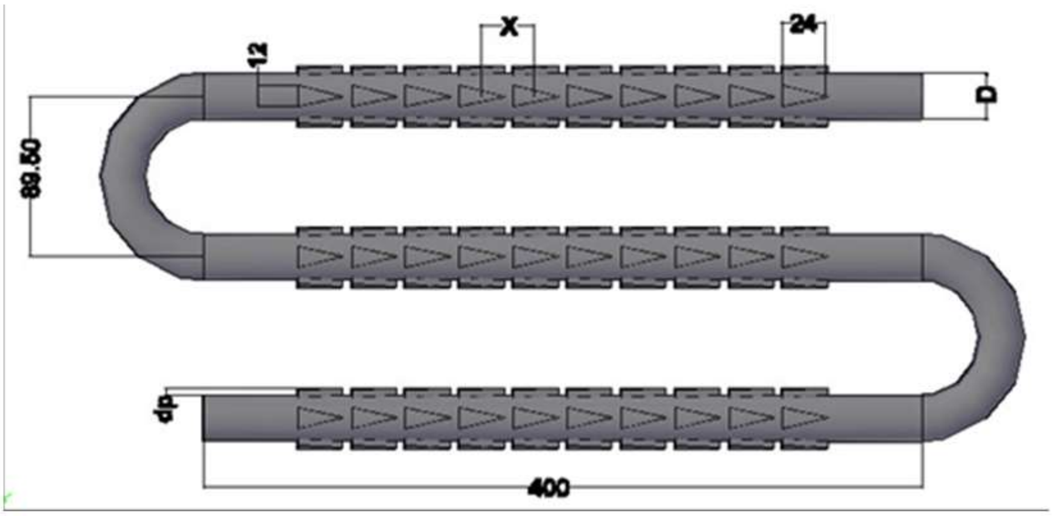

Figure (3) : The geometry of the triangle dimple heat exchanger .

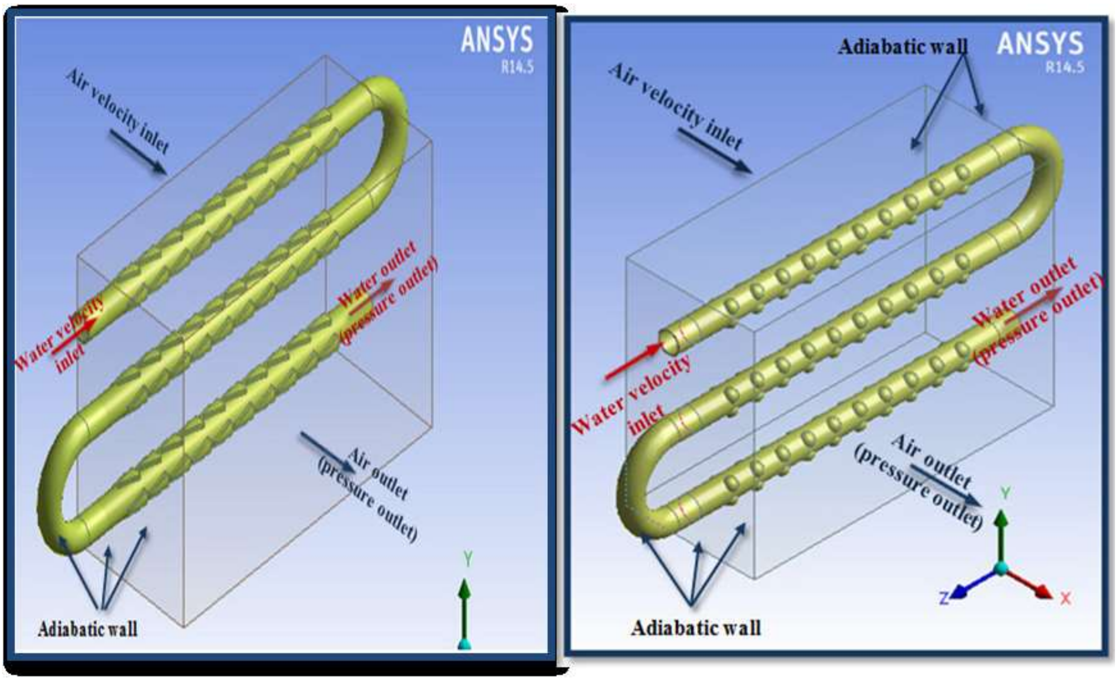

Fig 4. The Boundary Conditions.

As can be seen from this table and observing from Figs. 5 and 6 that after mesh (d) the difference in the values for both outlet temperature and pressure, the mesh number is very small and makes no signification therefore, and for more accuracy the (d) mesh has been chosen. Maximum skewness gained

in this simulation is 0.8, which indicates that the mesh has suitable quality and would not compromise the solution stability

Table 3. Mesh dependency results accuracy

No.	Number of elements (tetrahedral)	Outlet water temperature (°C)	Duct pressure drop (pa)
a	3304550	49.62	23.49
b	3759632	49.13	20.63
c	4138679	48.72	18.42
d	4633858	48.52	16.985
e	5025364	48.47	16.72
f	5407823	48.36	16.67

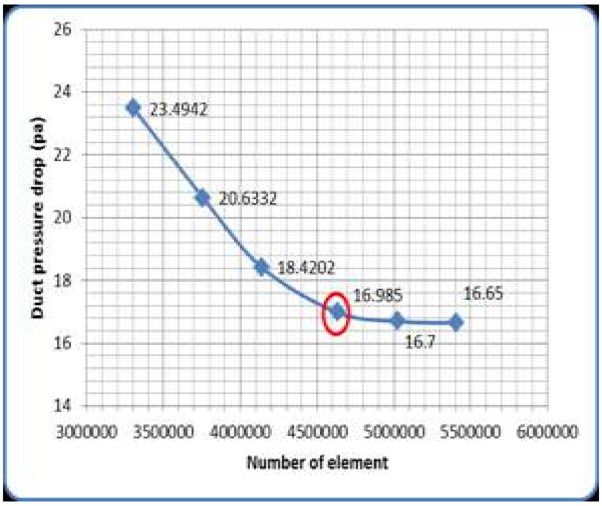

Fig 5. Variation of outlet water temperature with total element number.

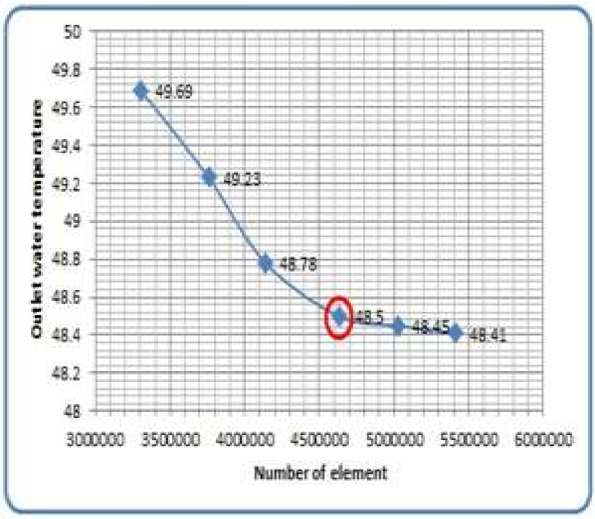

Fig 6. Variation of duct pressure drop with total element number.

5-Results and Discussion

5.1- Experimental Part

Fig. 7 represent the variation of Nusselt number with (Re) for plain heat exchanger and the circular dimple heat exchangers which shows that the Nusselt number is increased when dimple is added to the plain pipe that is due to the increasing of the turbulent air flow rate at downstream pipe.

The friction factor variation with Reynolds number is shown in Fig. 8. The friction factor value decrease with increasing of Reynolds number because addition of dimple on the plain pipe surface increase the friction factor as shown in the variation between the models under study which means that the magnitude of pressure drop increases in the dimple heat exchanger more than in the plain heat exchanger.

Fig. 9 shows the variation of the overall thermal performance for heat exchanger with the dimple tube. Thermal performance ratio is based on the experimental data of the Nusselt numbers and friction factors of those cases. The higher thermal performance can be attributed to the higher heat transfer coefficient relative to that in plain pipe heat exchanger.

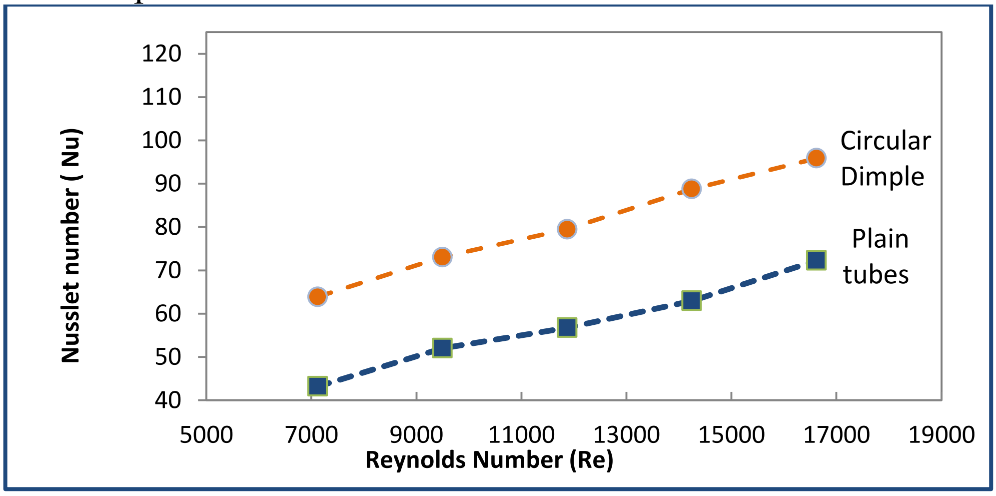

Fig 7. Experimental Average Nusselt number Vs. Reynolds number.

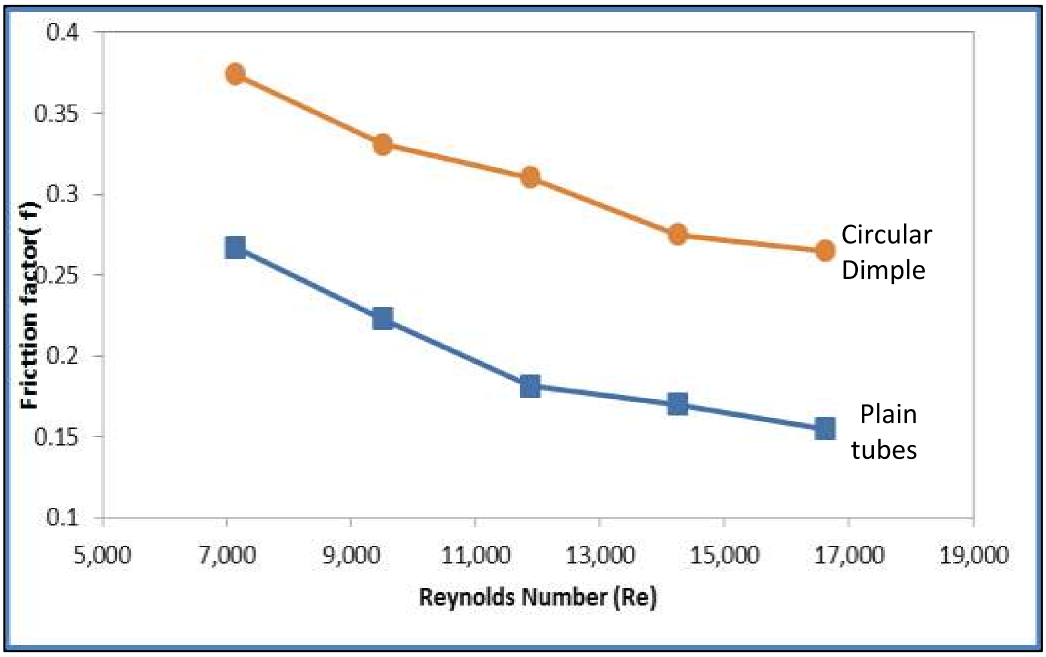

Fig 8. Experimental Variation of Friction Factor with Reynolds number

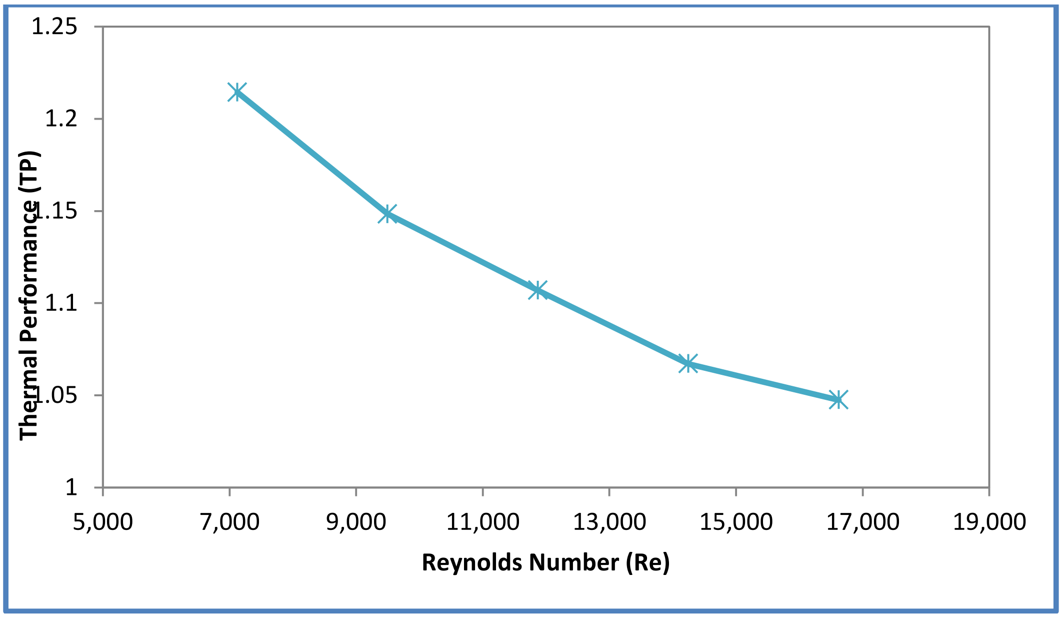

Fig 9. Experimental variation of thermal performance for circular dimple heat exchanger

5.2-Numerical Part

Fig. 10 represents the variation of Nusselt Number with Reynolds Number based on the numerical results which shows that the Nusselt number is increasing with the Reynolds number, and an increase about 45%and 12% for the Nusselt number was observed triangular dimple in comparison with the plain and the circular dimpled tubes respectively since dimples create a number of recirculation zones within and downstream the dimples which enhance heat transfer. The Nusselt number ratio (Nu/Nu_o) decreases when the triangular dimpled tube as shown in Fig. 11 in comparison with plain pipe heat exchanger.

The intensity velocity vectors for triangle dimple pipe are higher than that for plain case and this may attributed to the fact that, adding triangle dimple on the pipe circumference in the provided lower pressure in the downstream of the heat exchanger and air velocity will increase more in the recirculation zone. Fig. 12 shows the variation of friction factor data with Reynolds number obtained from the numerical simulation. The friction factor increases as pressure drop increased, similarly to the heat transfer results. The magnitude of friction factor increases in the dimpled heat exchanger more than other models by a 12.3% and 43% for the triangular, circular compared with plain tube respectively.

Fig. 13 shows the variation of friction factor ratio(f/f_o) with (Re). It shows that (f/f_o) is increasing for dimpled pipe over plain pipe. Figure(14) shows that the higher thermal performance is gained in the triangle dimple heat exchanger. The heat transfer enhancement is the gain upon what to force the flow through the channel .Hence increasing in pressure drop does not result in the same gain in Nusselt as the flow has little time to carry much heat from heat exchanger [5]. Fig. 15 shows the a comparison between the numerical and experimental results for the values of Nusselt Number of the plain tubes which indicate good agreement between both results with maximum difference 7% and 9% between them in the case of circular dimples shown in Fig. 16. Moreover the comparison between the experimental and numerical results for the values of friction factor in Figs 17 and 18 shows a difference of about 9%and 17% for the plain and circular dimple tubes. Unfortunately no similar cases were found in previous research for comparison.

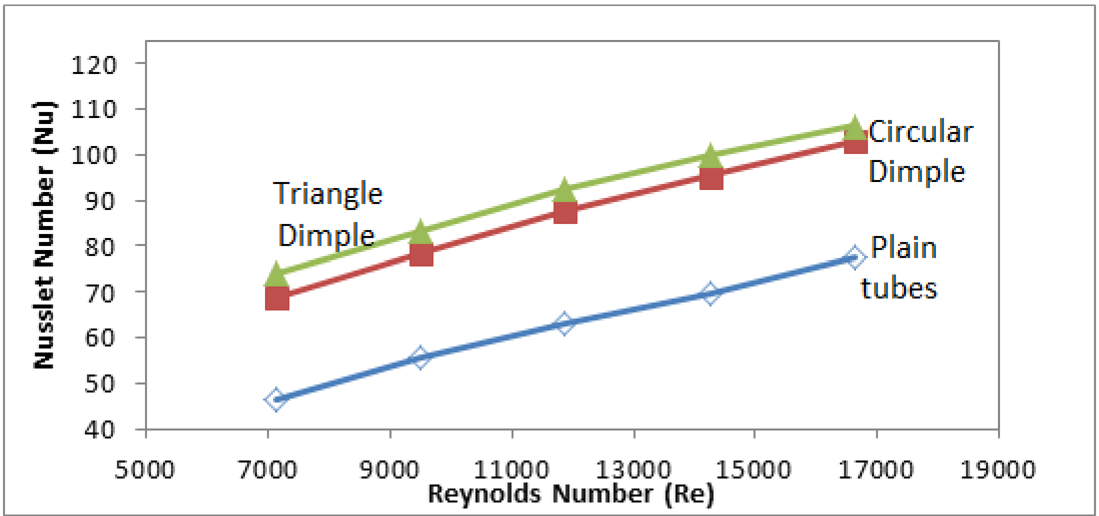

Fig 10. Numerical variation of Nusselt Number with Reynolds Number.

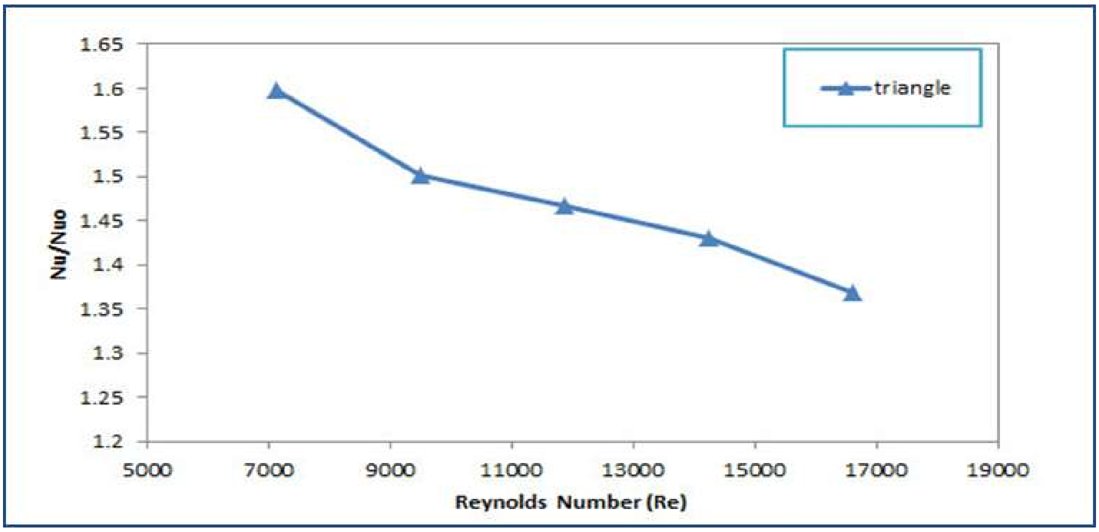

Fig 11. The variation of ($Nu/Nu_{\circ}$) with Reynolds Number for the triangular dimples

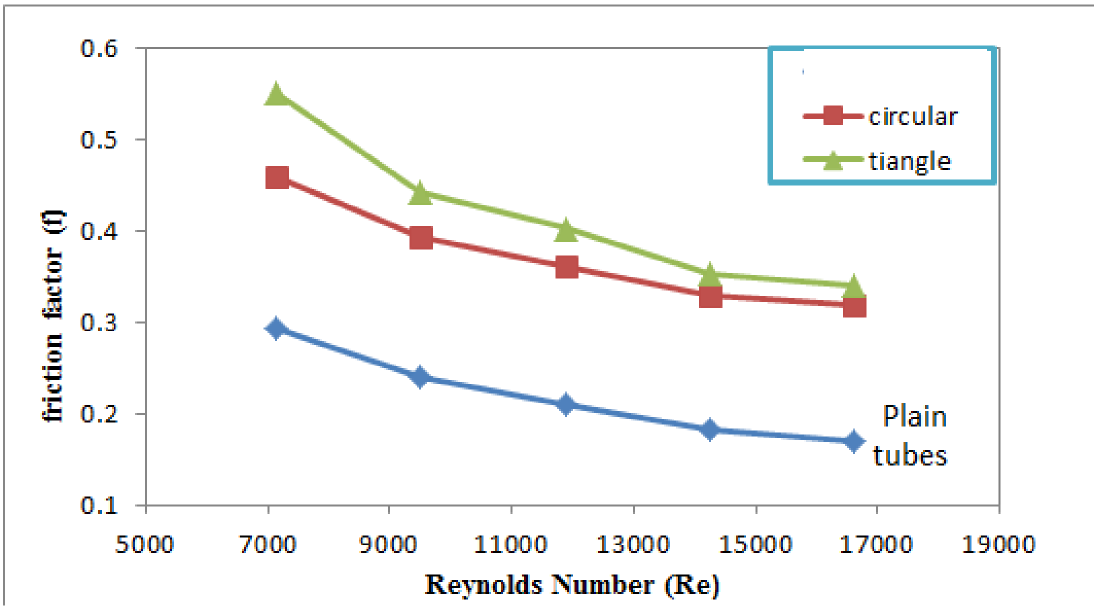

Fig 12. Numerical variation of Friction Factor withReynolds Number.

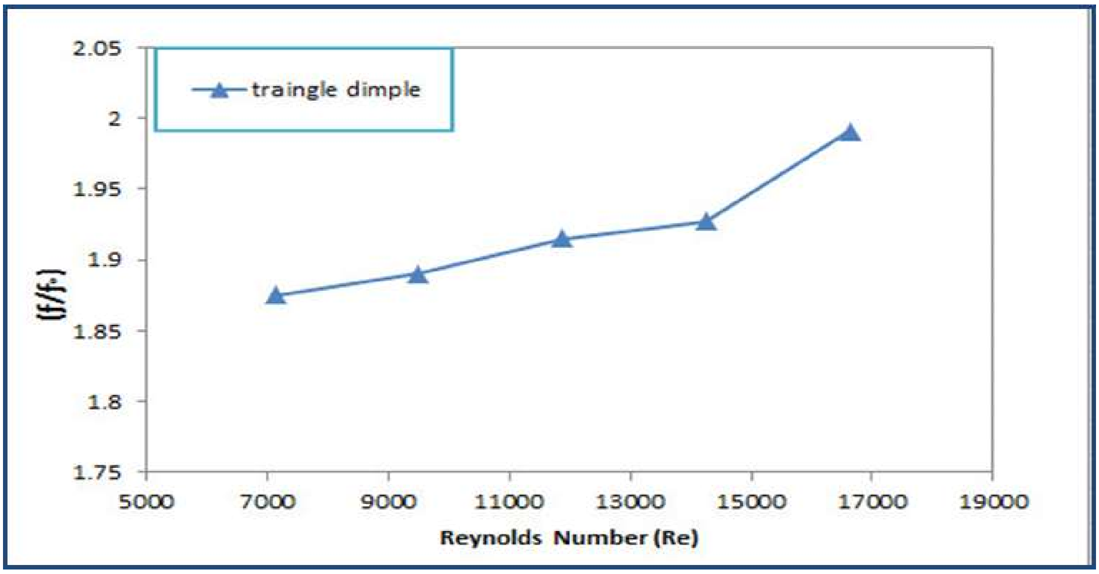

Fig 13. Numerical Variation of ($f/f_\circ$)withReynolds Number.

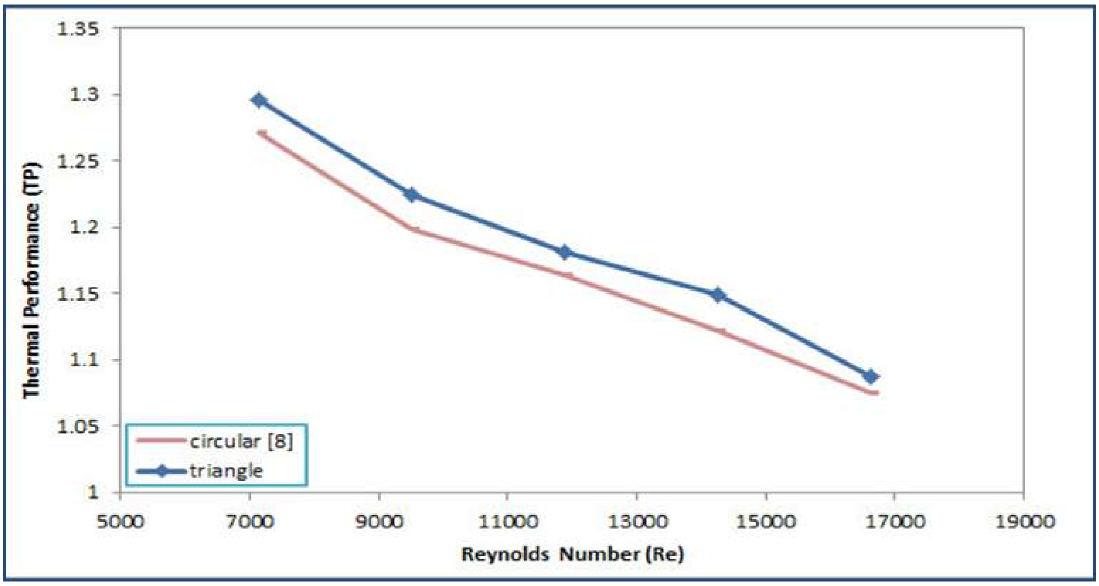

Fig 14. Numerical variation of Thermal Performance withReynolds Number.

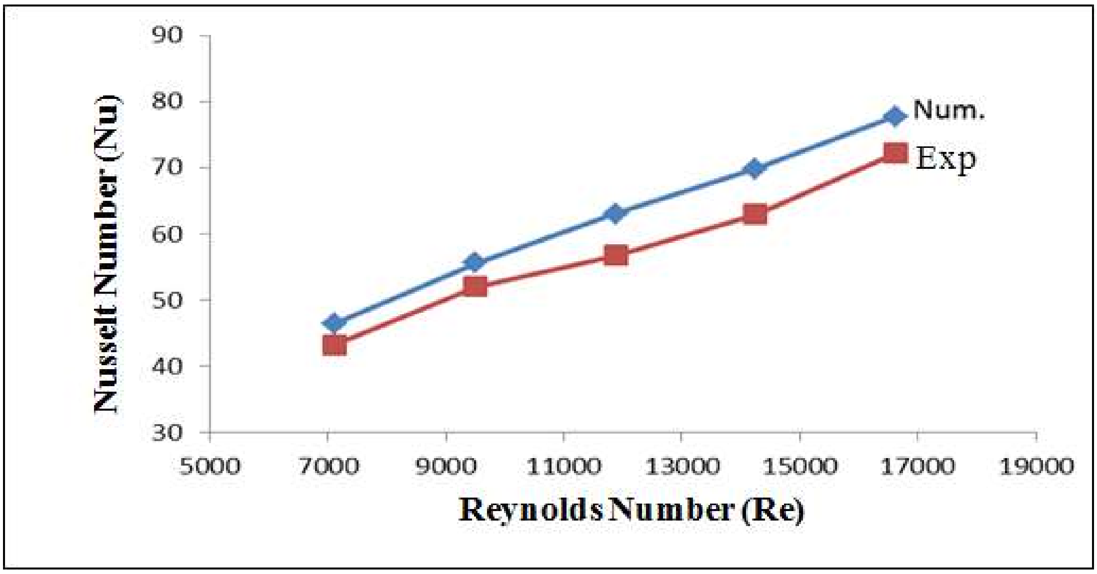

Fig 15. Comparison between the numerical and experimental results for the Nusselt Number of the plain tubes

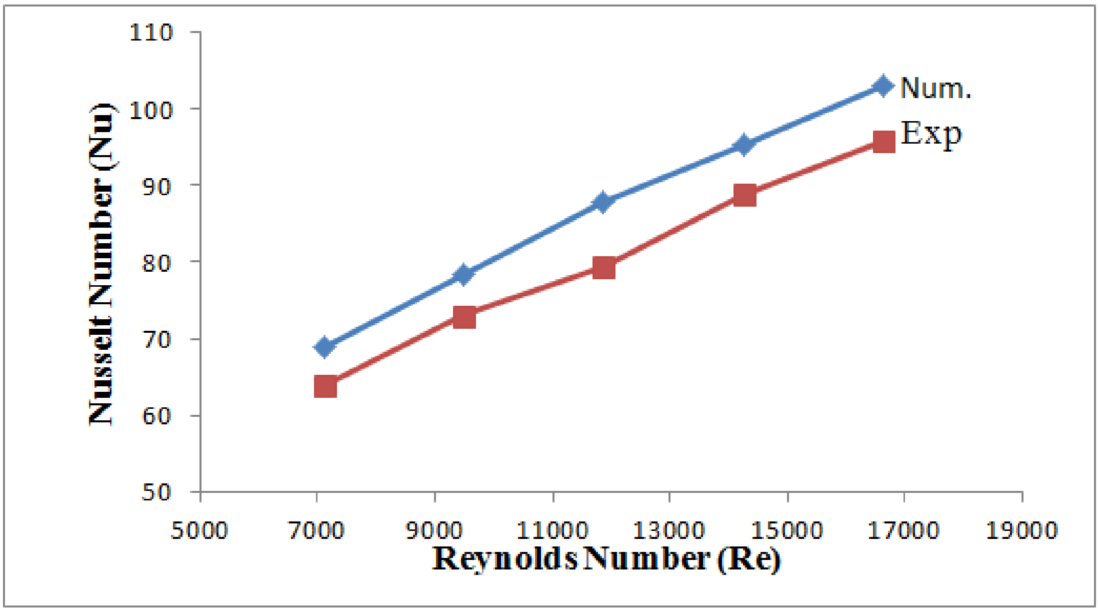

Fig 16. Comparison between numerical and experimentalResults for Nusselt Number of the circular dimpled tubes

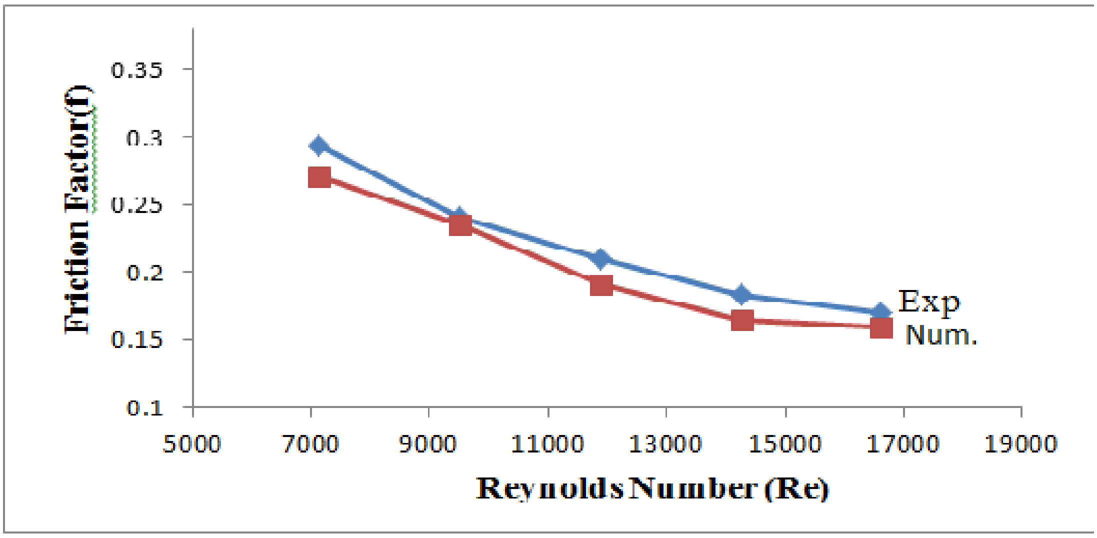

Fig 17. Comparison between numerical and experimental results for Friction Factor of the plain tubes

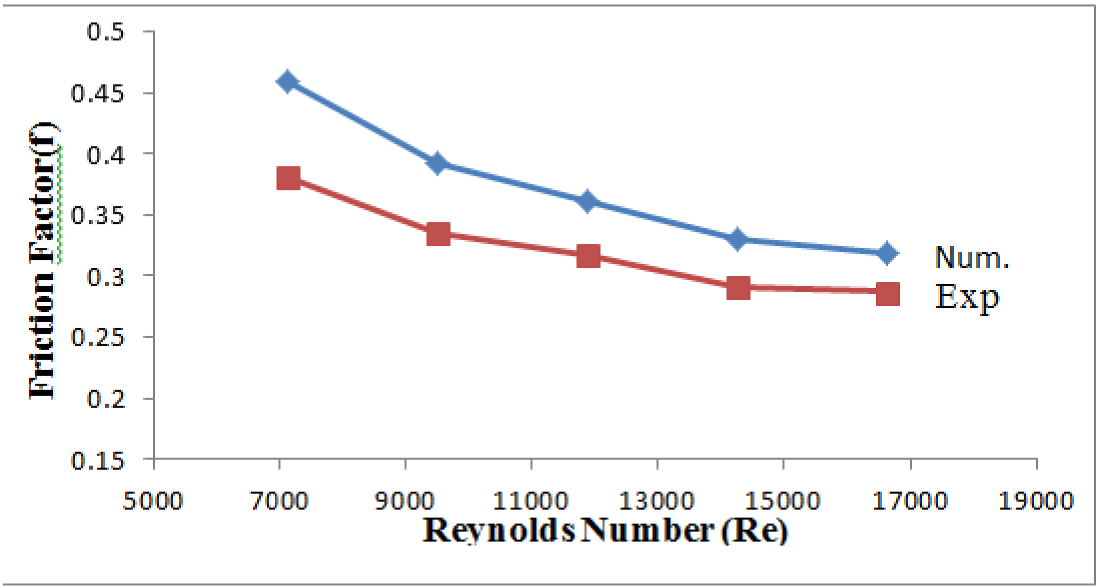

Fig 18. Comparison between numerical and experimental results for Friction Factor of the circular dimple tubes

6- Conclusions

From the previous discussion of the results obtained, the following conclusions can be extracted:

1-Average Nusselt number and the friction factor (i.e the pressure drop) increases by mounting dimples on the outer surface of the plain tubes.

2- Nusselt number of the triangular dimple tube is about 45 % higher than that for the circular dimple tube and 12% for the plain tube model while the pressure drop of dimple tube is about 12.3% higher than that for the circular dimple tube and 43 % for the plain tube model

3- Tube with dimple gives a higher reduction in surface temperatures than plain tube.

4-The present numerical results are seen to be in a good agreement with the experimental results (i.e. for enhancement of heat transfer have a maximum difference of ± 7% and for friction factor of ± 9%).

List of symbols

A_s= Surface area m^2
Cp= Specific heat J/kg.K
D=External Tube Diameter m
d= dimple diameter m
d_p= Dimple depth m
f= Friction factor
h=Convective heat transfer coefficient $W/m^2.K$
k= Thermal conductivity W/m. K
L_D=Duct Length m
L_p=Pipe Length m
$\dot{m}$= Mass flow rate Kg/s
N= Number of Tube Rows
n=number of Dimples
P=Pressure N/m^2
Q= Heat transfer W
q= Heat flux W/m^2

S_T= Transverse tube pitch
T =Temperature ^oC
V=Air Velocity m/s
W_D=Duct Length m
X = Transverse Dimple Center Pitch (mm)

Greek Symbols

μ : Dynamic viscosity (kg/m.s)
ρ : Density (kg/m^3)
∞:Surroundings

Subscript

a:air
av: average
i:inlet
o:outlet
s: surface
w:water

7 -References:

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تقييم لتحسن انتقال الحرارة في المبادلات الحرارية باضافة الانبعاجات

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احمد هاني غانم
باحث

الخلاصة :-

تم دراسه انخفاض الضغط وخواص انتقال الحرارة في مبادل حراري متعدد الانابيب مزود بانبعاجات على السطح الخارجي للانابيب عمليا ونظريا .تم تصميم وتصنيع نموذجين من المبادلات الحراريه(احدهما بانبيب ملساءوالاخرى تحوي على انبعاجات كرويه). تم تجهيز وتركيب عدد المبادلات الحراريه لاجراء التجارب ضمن حدود عدد رينولدز المختاره(7000-18500). في الدراسة النظرية تم استخدام برنامج ANSYS Fluent.14.5 باستخدام طريقة (SST) للجريان الاضطرابي لثلاث نماذج من المبادلات الحراريه(بانبيب ملساء،انابيب تحوي على انبعاجات كرويه و انابيب تحوي على انبعاجات مثلثيه).اظهرت النتائج التجريبيه ان متوسط عدد نسلت وانخفاض الضغط داخل المبادل الحراري ذات الانبعاج الكروي اعلى من المبادل بالانابيب الملساء بينما اظهرت النتائج النظريه ان افضل النتائج يمكن الحصول عليها باستخدام الانابيب ذات الانبعاجات المثلثيه.تم اجراء المقارنة بين النتائج العمليه والنظرية واظهرت تطابقا جيدا بينهما.