EXPERIMENTAL STUDY OF HEAT TRANSFER ENHANCEMENT BY USING POROUS MEDIA IN DOUBLE TUBE HEAT EXCHANGER +

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Abstract:

Forced convection heat transfer in counter flow heat exchanger which uses porous media to enhance heat transfer through it, has been investigated experimentally in the present work.

Theexperimental work included the design of double pipe heat exchanger with the dimension of (length 1.11m, 0.063 outer pipe diameter and 0.031m inner pipe diameter). Alumina(2.5mm diameter) as a porous media is added in heat exchangerat three case: inner porous tube (IP) , outer porous tube (OP) and inner- outer tubes (IOP) .Cold andhot water are used as working fluid to transfer heat through outer and inner pipe respectively. The experiment are conducted at hot water mass flow rates (0.0166-0.0833 Kg/s),cold water mass flow rates (0.05-0.116 Kg/s) ,inlet temperatures of hot water (47, 55°C) andinlet temperatures of cold water(20, 25 °C).

The results are obtained for range of Reynolds number to hot water (1144< Re<6241) and cold water (3939<Re<7254) and shown that theincreasing in mass flow rate ratio (hot water mass flow rate/ cold water mass flow rate) causes decreasing in effectiveness of heat exchanger at all cases. The effectiveness in (IOP,IP and OP) are higher at (39,34 and 25)% percent than (NP non porous added).Adding alumina beads lead to increase in effectiveness of heat exchanger, the maximum increment is obtained in (IOP) case due to increase in surface area needed to heat transfer, also increases in the turbulence in working fluid (water) destroys the boundary layer which considering as additional resistance to heat transfer .

Keyword: heat exchanger- porous media- forced convection- counter flow

دراسة عمليه لتحسين التبادل الحراري باستخدام وسط مسامي لمبادل حراري مكون من اسطوانتين متحدتي المركز

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المستخلص :

البحث الحالي هو دراسة عملية لانتقال الحرارة بالحمل القسري خلال مبادل حراري مملوء بوسط مسامي لتحسين انتقال الحرارة خلاله. تضمن البحث تصميم مبادل حراري من نوع اسطوانتي متحدتي المركز بأبعاده (1.11m طول، قطر الاسطوانة الداخلية 0.031m ، وقطر الاسطوانة الخارجية 0.063 m). أستخدمت كرات من من الالومينا بقطر (2.5 mm) كوسط مسامي في المبادل الحراري بثلاثه حالات: في الاسطوانه الداخليه (IP)، في الاسطوانه الخارجيه

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(OP)، في كلا الاسطوانتين(IOP).. تم استعمال الماء البارد والحار كمائع ناقل للحرارة من جانب الاسطوانه الخارجيه والداخلية على التوالي.

جرت الدراســـة بمعدل تدفق للماء الحار تتراوح بين (kg/s 0.833 – 0.0166) ومعدل تدفق للماء البارد (kg/s 0.116 – 0.05)و درجه حراره الدخول للماء الحار فهي (°C 47, 55) ودرجه حراره الدخول للماء البارد (25, 20 °C).

بينت لنتائج التي تم الحصول عليها هي ضمن رقم رينولد (1144< Re<6241) للماء الحار و (3939<Re<7254) للماء البارد، عند زيادة نسبه التدفق (معدل تدفق الماء الحار/معدل تدفق الماء البارد) تؤدي إلى انخفاض في فعاليه المبادل الحراري لكل الحالات. وتكون الفاعلية في حالة (IOP) ,(IP),(OP) تكون اعلى بنسبة (39)،(34)، (25) % من حالة(NP) (بدون اضافة وسط مسامي). عند أضافه الوسط المسامي للمبادل الحراري يؤدي إلى زيادة ملموسة في الفعاليه فاعلى زيادة يتم الحصول عليها في (IOP) لزيادة المساحة السطحية اللازمة للتبادل الحراري وكذالك زيادة الاضطراب في الوسط الناقل للحرارة (الماء) وقيام الوسط المسامي بتقليل سمك الطبقة المتاخمة الحرارية والتي تعتبر مقاومة إضافية لانتقال الحرارة

الكلمات ألداله: مبادل حراري- الوسط المسامي- الحمل القسري- جريان متعاكس .

Nomenclature

A	*Area*	m^2
Cp	*Specific heat*	*kJ/kg. K*
C_{min}	*Minimum heat capacity of cold or hot fluid*	*kJ/ K*
D, d	*Diameter*	*m*
D_h	*Heat exchanger hydraulic diameter ,*	*m*
D_P	*Particle diameter*	*m*
F	*Correction factor*	----
h_i	*Heat transfer coefficient of hot water*	$(W/m^2{\cdot}K)$
h_o	*Heat transfer coefficient of cold water*	$(W/m^2{\cdot}K)$
h_w	*Convection heat transfer coefficient at wall*	$(W/m^2{\cdot}K)$
IOP	*Inner – outer porous*	-------
IP	*Inner porous*	-------
K	*Thermal conductivity*	$(W/m{\cdot}K)$
K_d	*Thermal dispersion conductivity*	$(W/m{\cdot}K)$

K_{st}	*Stagnant thermal conductivity*	*(W/m·K)*
L	*Length of tube*	*m*
LMTD	*Logarithmic mean temperature difference*	*K*
M	*Mass flow rate*	*kg/s*
NTU	*Number of heat transfer units*	-------
NP	*Non porous*	-------
OP	*Outer porous*	-------
Q	*Heat transfer rate*	*W*
R	*Thermal resistance*	*$^oC/W$*
T	*Temperature*	*K*
U	*Overall heat transfer coefficient*	*$W/m^2\cdot K$*

Greek Symbols

$\in$	*Heat exchanger effectiveness*	----
μ	*Fluid viscosity*	*kg/m.s*
υ	*Kinematic viscosity*	*m^2/s*
ε	*Porosity*	-------

Subscripts

C	*Cold*	*R*	*Ratio*	*f*	*Fluid*
Ci	*Cold in*	*S*	*Surface, solid*	*h*	*Hot*
Co	*Cold out*	*T*	*Total*	*hi*	*Hot in*
Eff	*Effective*	*W*	*Wall*	*ho*	*Hot out*

Introduction:

Heat exchangers are extensively used in several industry equipment such as thermal power plants, chemical processing plants, air conditioning, equipmentetc. The mechanisms of enhancing heat transfer are classified as active or passive methods and there are several studies have reported that the use of porous media for enhancement heat transfer yields higher heat transfer performance than the other techniques. Porous media with high thermal conductivity have emerged as an effective method of heat transfer enhancement due to their large surface area to volume ratio and intense mixing of the flow. Examples of industrial application involving porous media are chemical catalytic reactors, petroleum reservoirs, and geothermal energy thermal storage. By using the porous media as transport medium, fluid mixing and the heat exchange area between the solid matrix and fluid phase will be increasedand leads to significant enhancement in heat transfer. Experimental study on multi-layered type of gas-to-gas heat exchanger using porous media by **Tomimura, et al [1]** an effective energy conversion method between flowing air enthalpy and thermal radiation air was examined. A series of experiments, on the parameter of inlet temperature on high temperature gas at (300-700 °C), the optical thickness of porous media of (0-15.4)mm, the number of layer (2-5) and two types of walls(bare or finned) placed in the system had been conducted. It was shown that heat recovery section is shown to play an important role in lowering an outer wall temperature of the system and at the same time in increasing the total heat recovery rate $H_{tot,N}$(total heat recovery rate of multi-layered gas to gas heat exchanger). The finned walls were quite effective to promote $H_{tot,N}$ under the present experimental conditions than the bare wall.

Heat transfer enhancement for gas heat exchangers fitted with porous media were studied experimentally and numerically by **Bogdan, et al [2].** He investigated the effect of commercial aluminum screen(wire of diameter 0.8mm, density 2770 kg/m^3, thermal conductivity 177 w/m^2 k) as a porous media with the range diameter and porosity of it(25.4mm-63.5mm), (0.966-0.993) respectively. The porous media was inserted in copper pipe of (ro=31.75mm) subjected at a constant with uniform of heat flux and the range of Reynolds number was (1000<Re<4500). The effects of porosity, porous material diameter and thermal conductivity as well as Reynolds number on the heat transfer rate and pressure drop were investigated. The results are compared with the clear flow case where no porous material was used. The results obtained lead to higher heat transfer rates could be achieved using porous inserts at the expense of a reasonable pressure drop and the rate of increasment in Nusselt number is (3-6) time when compared with no porous case.

Hayes, et al [3] studied heat transfer and fluid flow characteristics through square channel (38mm by38mm) numerically and experimentally at range of(100< Re<1150).For numerical simulation two models were created: a two-dimensional numerical model and a Fluent computational fluid dynamics (CFD) porous media model. The numerical model was modified to better simulate matrix heat exchanger. The numerical model of generated temperature profiles that were used to calculate heat transfer coefficient of the matrix heat exchanger and developing a correlation between the Nusselt number and the Reynolds number.

The performance of an air-cooled condenser (finned tube bundle) was numerically examined by .**Hooman [4]**.The range of Reynolds number of(600<Re<700). The finned tube bundles in the condenser are represented by a porous matrix, which is defined by its porosity (ε), permeability (K), and the form drag coefficient (CF). The dimensions of the fin are (thickness 0.6mm, and the tip being marked by a 60mm distance) .The porosity is equal to the tube bundle volumetric void fraction and the drag coefficient is found to be a function of the porosity. A close correlation is found between the form drag coefficient and the porosity with the drag coefficient decreasing with increasing porosity. Heat transfer and second law (of thermodynamics) with performance of the system have also been investigated. The volume averaged thermal energy equation is able to accurately predict the hot spots. It has also been observed that the average dimensionless wall temperature is a parabolic function of the form drag coefficient. For the porosity that is equal 0.769, the excess pressure drop is 0.22 while the increasment in heat transfer is 2.26 times that of the no-fin case.

Odabaee, et al [5]investigated numericallythe heat transfer from an aluminum metal foam-wrapped solid cylinder in cross-flow and use air as a working fluid and the Reynolds number that used is($Re < 2 \times 10^5$). Effects of the key parameters including the free stream velocity and characteristics of metal foam such as porosity, permeability, and form drag coefficient on heat and fluid flow are examined. Being a determining factor in pressure drop and heat transfer increment, the porous layer thickness is changed systematically to observe that there is an optimum layer thickness beyond which the heat transfer does not improve while the pressure drop continues to increase. Results have been compared to those of a finned-tube heat exchanger to observe much higher heat transfer rate with reasonable excess pressure drop leading to a higher area goodness factor for metal foam-wrapped cylinder.

Although there are many works in porous media of heat exchanger , but there are not many experimentally investigations on double tube heat exchanger contain porous media and forced convection with water as a working fluid.In the present study the forced convection in a double tube heat exchanger for various Reynolds number, various location of porous media at different entering temperature of hot and cold water were experimentally investigated.

Experimental Apparatus:

The experimental apparatus is shown in **Fig.(1-a)** and **Fig.(1-b)**diagrammatically and photographically respectively It isdesigned and constructed specially for investigating the effect of adding alumina beds as a porous media on heat transfer in double pipe heat exchanger. It consists of:

1-Test section: horizontal concentric double pipe made of aluminum with dimensions of outer diameter (63mm , 31mm) andinner diameter (58mm , 28mm) for outer and inner pipes respectively. It is shown in **Fig.(2).**

2-Porous bed: It is composed of alumina bed with average diameter (2.5mm).

3- Cold water cycle supplies the fluid passing through outer pipe.

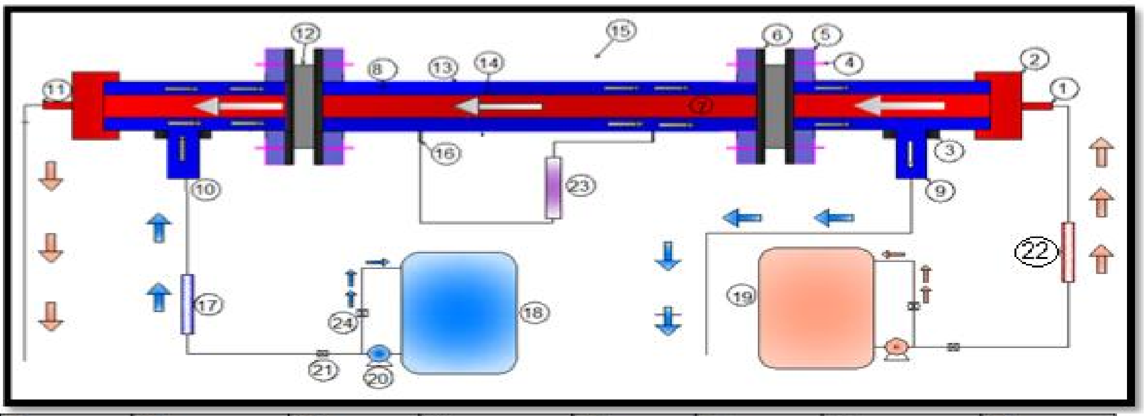
4- Hot water cycle supplies the fluid passing through inner pipe.

5- The measuring instruments.

The range of operating parameter is given in **Table(1)**. These parameters belong to four cases, the first without porous material (NP), the second pad additive in the inner pipe only (IP), in the third case pad is in the outer pipe only(OP) and the last the pad is to be in the both pipes (IOP).

Table(1) The range of operating parameter

parameter	Cold water (outer pipe) flow rate(kg/s)	Hot water (inner pipe) flow rate(kg/s)	Cold water inlet temperature ^{o}C	Hot water inlet temperature ^{o}C
Range	(0.116-0.05)	(0.0833-0.0166)	(20 and 25)	(55 and 47)



1 Position entering hot water	2 Connection to entering hot water	3 washer	4 screw	5 flange	6 rubber	7 Hot water at inner tube	8 Cold water in outer tube
9 Position of outlet cold water	10 position of entering cold water	11 Position of outlet hot water	12 Teflon connection to centering pipes	13 Outer tube	14 Inner tube	15 Test section	16 Pressure tap
17 Cold water flow meter	18 Cold water tank	19 Hot water tank	20 pump	21 valve	22 Hot water flow meter	23 Inclined manometer	24 By-pass

Fig. (1-a) Diagram of experimental apparatus

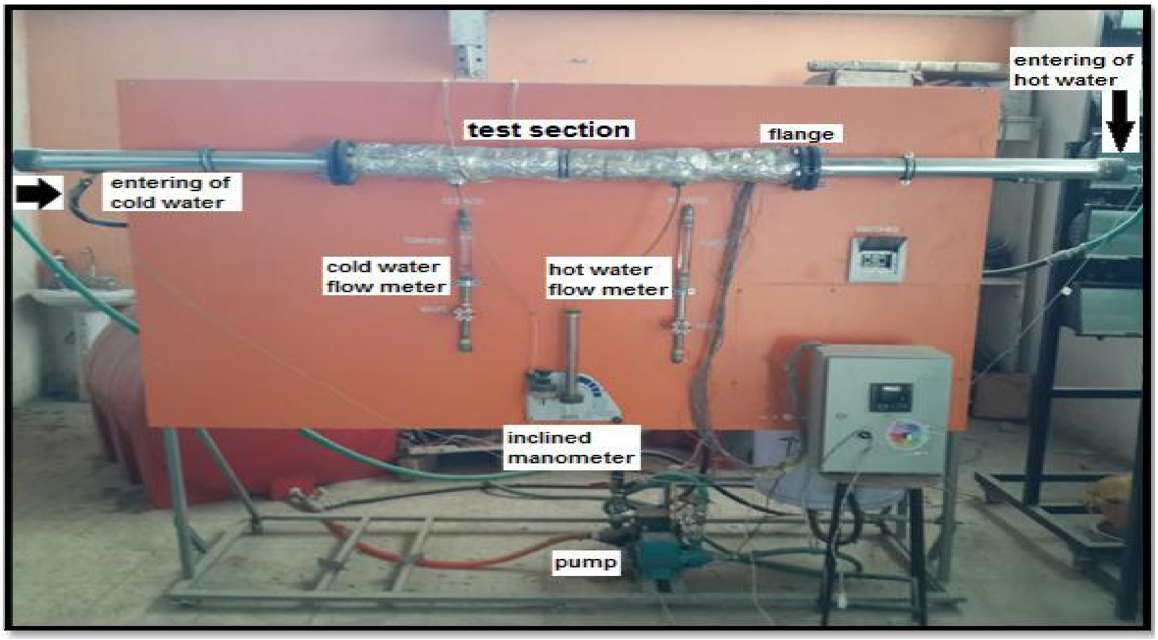


Fig.(1-b) Photograph of rig assembly

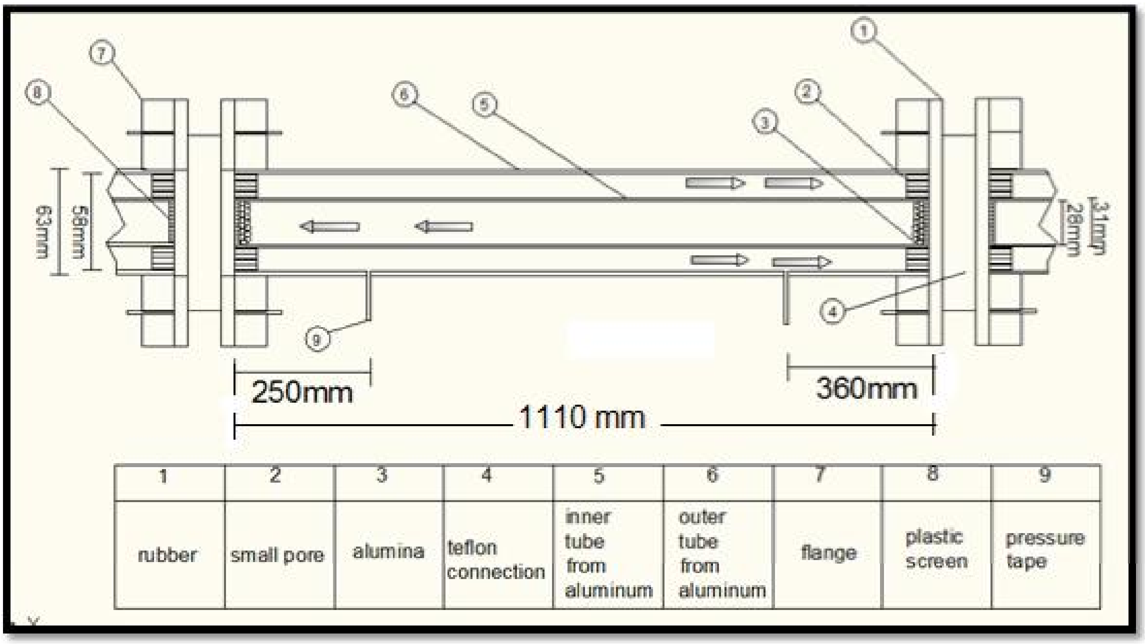


Fig. (2) Test section

The outside test section is thermally insulated with(100mm) thickness of glass wool to reduce heat losses from the test section to the surrounding, by conduction and radiation.

The temperature distribution through heat exchanger is measured by (18) type-K thermocouples .Two thermocouples are placed to measure temperature of cold and hot water before entering heat exchanger , four thermocouple wire penetrating the heat exchanger are put to measure entrance and exit of hot and cold water, ten thermocouple to measure surface temp.at test section five at outer surface and five ay internal surface of the tube , and two thermocouples are placed to measure temperature of hot and cold water tanks.

Adjustable differential manometer is used to measure the pressure difference with in the porous media. .Two water flowmeters are need to measure the water flow, one to measureflow rate of hot water and another to measure the flow rate of cold water.

Performance calculation:

The first law of thermodynamics requires that the rate of heat transfer from the hot fluid be equal to the rate of heat transfer to the cold one, or

*Heat transferred for the hot and cold water can be calculated from Eq. (1)(2).**[6]***

$$Q_h = m_h cp_h (T_{hi} - T_{ho}) \quad (1)$$

$$Q_c = m_c cp_c (T_{co} - T_{ci}) \quad (2)$$

The heat transfer coefficient in tube, h_{wi} can be calculated from:

$$h_{wi} = \frac{Q_h}{Ai(T_w - T_s)} \quad (3)$$

$$T_s = \frac{(T_{c_1} + T_{c_2} + \cdots + T_{c_n})}{n} \quad (4)$$

$$T_w = \frac{T_{hi} + T_{ho}}{2} \quad (5)$$

$\frac{1}{U} = \frac{1}{hw_i} + \frac{1}{hw_o}$ ***(6)***

$U = \frac{Q_c}{A \times F \times LMTD}$ ***(7)***

$h_{wo} = \left[\frac{1}{U} - \frac{1}{h_{wi}}\right]^{-1}$ ***(8)***

$Nu\_h = \frac{h_{wi}D}{K}$, $Nu\_c = \frac{hwoD_h}{K}$ ***(9)***

This law is applied when material is not found but when used porous change K by K_{eff} that calculated from this law ***[9]***

$Keff = k_{st} + K_d$ ***(10)***

where

$k_{st} = k_f\{1 - \sqrt{1-\varepsilon} + \frac{2\sqrt{1-\varepsilon}}{1-\gamma\beta} * \left[\frac{(1-\gamma)\beta}{(1-\gamma\beta)^2} ln\left(\frac{1}{\gamma\beta}\right) - \frac{\beta+1}{2} - \frac{\beta-1}{1-\gamma\beta}\right]\}$ ***(11)***

and

$\beta = 1.25[\left(1 - \frac{1-\varepsilon}{\varepsilon}\right)^{\frac{10}{9}}]$ $\gamma = \frac{K_f}{k_{st}}$ $K_d = o.1K_f * p_{ep}$

$Q_{max} = C_{min}(T_{hi} - T_{ci})$ $\in = \frac{Q_{act}}{Q_{max}}$ ***(12)***

$NTU = UA / C_{min}$ ***(13)***

Results And Discussion :

Results of the effect of porous media and the ratio of hot to cold water mass flow rate(m_r) on the temperature distribution of counter flow of heat exchanger are shown in **Figs**. **(3)** to **(6).**

Part **(a)** from **Figs**. **(3)** to **(6)** shows typical temperature distribution along the outer surface of inner tube of heat exchanger under four case of porous media(Np, IP, OP, IOP) respectively for the same conditions (m_r=0.332, T_{ci}=25°C, T_{hi}=47°C).

In general shape of all curve shows that the temperature decreases towards the end of heat exchanger due to heat transfer from hot water to the cold water . Also the surface temperature reduce in cases that use alumina as a porous media when compared with NP case, but the minimum temperature surface is obtained when porous media is added to both pipes (IOP). The drop in temperature on the surface when using alumina due to the mechanisms of heat transfer in porous media are convective heat transfer from tube wall to the water, convective heat transfer from alumina particles to water and conduction from wall to the alumina that have high thermal conductivity. The other cause for the drop in temperature surface is the use of porous media increasing surface area and mixing of fluids due to the presence of particles.

Part **(b)Figs.(3)** to **(6)** show the temperature distribution inside heat exchanger for four cases NP, IP, OP, IOP fixed inlet condition of (m_r=0.5, Tci=25°C, Thi=47°C) respectively.

And part **(c)** the figures also show the temperature distribution inside heat exchanger for four cases NP, IP, OP, IOP respectively for fixed condition (m_r=0.718, Tci=25°C, Thi=47°C).

These figures show the minimum m_r in part **(a)** so that these are high temperature distribution and less surface temperature but in parts **(b, c)** the increasing in mass flow rate leads to decrease in temperature distribution and increase in surface temperature when compared with part **(a)** in four cases NP, IP, OP, IOP because increase in inlet velocity in tube in tube heat exchanger that is leads to change of flow pattern inside heat exchanger and change vortex form that is developing in the heat exchanger the form of vortex in the heat exchanger has significant effect, or destroy thermal boundary layer.

Fig.(7)shows the effect of mass flow rate with using porous media in double tube heat exchanger on Qc ,Qh. The heat transfer rates are highly dependent on decreasing the thermal resistance of the tubes of heat exchanger. Physically, increasing the inlet inner and outer tube mass flow rate leads to decrease in thermal boundary layer thickness then decreasing the thermal resistance which causes an increase in the heat transfer. Heat transfer rate at (IOP) is higher than the rate in the cases (IP) and (OP) but all of them are higher than (Np) due to increasing surface area as a result of the filled tube of heat exchanger with porous media and increase in the turbulence water so that high heat transfer between hot and cold fluid will occurred.

Figs (8) show the effect of mass flow rate and use porous media in double tube heat exchanger on LMTD. It is obvious from this Fig that the LMTD increases with increase in mass flow rate. LMTD depends on the heat transfer rate which increases with increasing the mass flow rate. It is seen that in (NP) case (LMTD) is higher than (IOP) case. Also (IP, OP)cases are also less than NP case because the use of alumina causes decrease in the ((Thi-Tco) and (Tho-Tci)) and increase in overall heat transfer coefficient due to porous pad which increases in surface area of heat transfer between hot and cold water and causes heat transfer by conduction between particle and wall of tube in addition to heat transfer by convection between particle and wall and water.

In **Fig (9)** effectiveness is plotted as a function of mass flow rate ratio for counter flow tube in tube heat exchanger. Notice in this figure that the slope of the curve falls rapidly as the value of mass flow rate increases. This means for a certain heat exchanger, increment in inner and outer tube mass flow rate will always downgrade the effectiveness. The mass flow rate of hot and cold flow rates has an adverse effect on effectiveness of heat exchanger . Physically, increasing the mass flow rate ratio means more hot and cold mass flow rate leading to less temperature fall in the hot stream and consequently worsens effectiveness of the heat exchanger. On the contrary, little hot and cold mass flow rates lead to more temperature fall in the hot stream and consequently better the modified effectiveness of the heat exchanger.

But this figure shows the use of alumina as a porous media increase the effectiveness of heat exchanger when compared with NP case because the use of alumina causes resistance to water flow rate and destroys thermal boundary layer in addition the porous media increases the surface area of heat transfer between hot and cold water through the tube so that the effectiveness increases when use porous media in double tube and the increase in effectiveness at IP, OP less than IOP at (4 , 11)% percent and more than NP case at (34 ,25)% percent but the effectiveness in case of IP is higher than that of OP due to large amount of alumina in inner tube when compared with outer pipe.

Fig.(10) shows the relation between the effectiveness and NTU (number of heat transfer unit) and the effect of porous media. This figure shows the effectiveness increases with increase in NTU because the effectiveness and NTU depend on heat transfer coefficient which increases with mass flow rate because the NTU and effectiveness depend on the heat transfer coefficient so that this figure shows the (NTU) at (IOP) is more higher when compared with (IP, OP and NP) cases and the curve of (IP, OP) cases higher than that of NP case .

Fig.(11) depicts the relation between Nu- Re of hot and cold water for four cases NP , IP, OP and IOP respectively. It is shown that Nu increases with increasing Re. Any increment in the mass flow rate causes increase in velocity of hot or cold water that enters the heat exchanger and causes more turbulence in water leading to high heat transfer. when alumina filled the tubes of heat exchanger are resulting more increment in turbulence and then increasing Nu number. The (IOP) case has higher (Nu) than other cases. and the relation between Nu and Re numbers may be written in this way:

$Nu=aRe^{b}$ **(14)**

so that from this figures find correlation between Nu and Re numbers and the constants (a,b) are shown in Table **(2)**.

Table (2) Correlation between Nu and Re

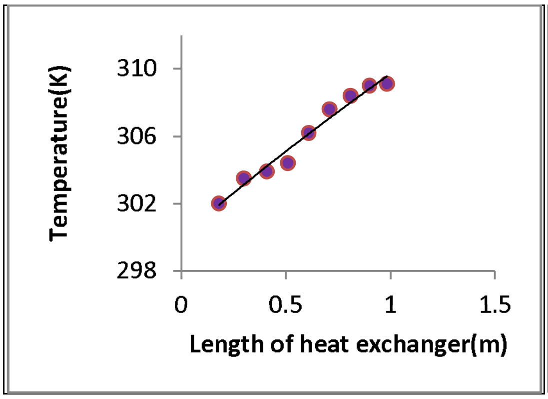
Hot water				
	NP	*IP*	*OP*	*IOP*
A	*12.766*	*16.86*	*12.23*	*20.06*
B	*0.1513*	*0.1502*	*0.191*	*0.183*
Cold water				
	NP	*IP*	*OP*	*IOP*
A	*0.8798*	*0.2131*	*0.796*	*0.777*
B	*0.507*	*0.7443*	*0.5891*	*0.612*

The experimental result of pressure drop along test section shows that alumina is added as a porous media in **Fig.(12)**show the relation between Reynolds number and pressure drop it is seen that the pressure drop increases with increase in Reynolds number along test section due to resistance applied of porous pad to water that flows through it. It increases with adding pad through outer tube.

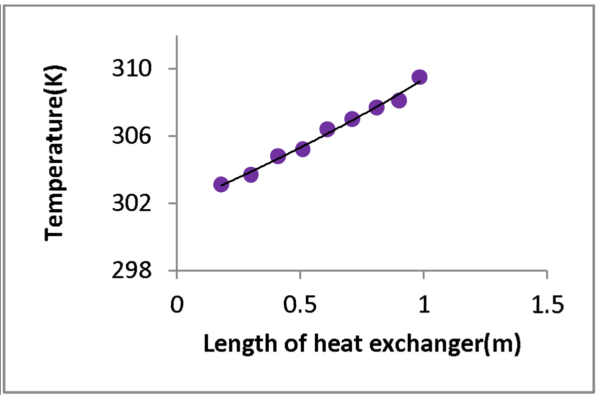
Table(3) show that effect of increasing the initial temperature of hot and cold water that is entering heat exchanger .

Table(3) Outer temperature at two cases.

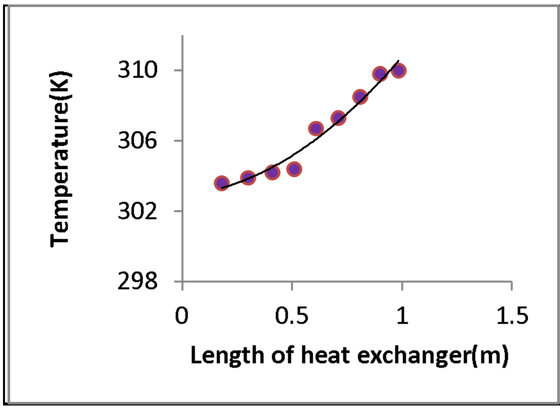
(25, 47°C)				
	NP	***IP***	***OP***	***IOP***
Tc-out	*29.2°C*	*30°C*	*30.1°C*	*30.9°C*
Th-out	*35°C*	*31.4°C*	*31.2°C*	*31°C*
(20, 55°C)				
	NP	***IP***	***OP***	***IOP***
Tc-out	*23.9°C*	*26.8°C*	*27°C*	*29°C*
Th-out	*43.3°C*	*35.4°C*	*35.8°C*	*29.8°C*



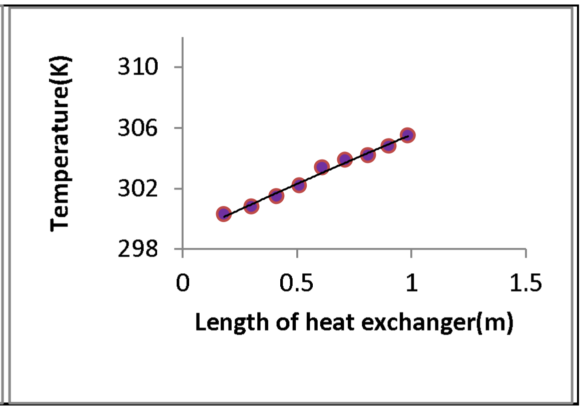
Fig(3-a) Axial surface temperature distribution along outer surface of tube at(mr=0.332 and NP case)



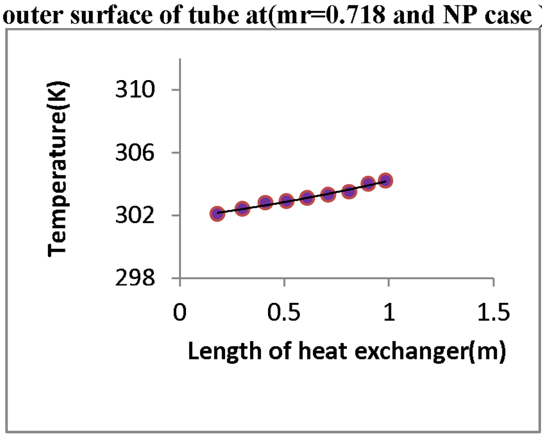
Fig(3-b) Axial surface temperature distribution along outer surface of tube at(mr=0.5 and NP case)



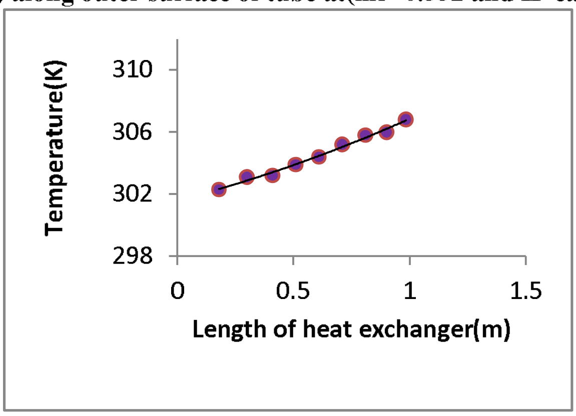
Fig(3-c) Axial surface temperature distribution outer surface of tube at(mr=0.718 and NP case)



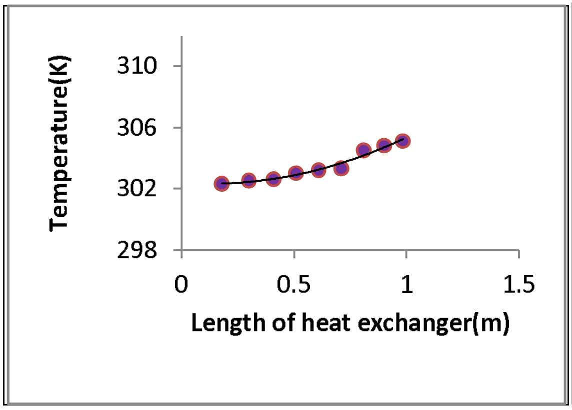
Fig(4-a) Axial surface temperature distribution along along outer surface of tube at(mr=0.332 and IP case)

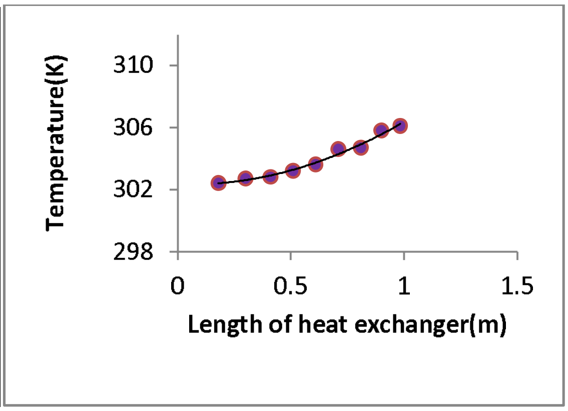


Fig(4-b) Axial surface temperature distribution along outer surface of tube at(mr=0.5 and IP case)



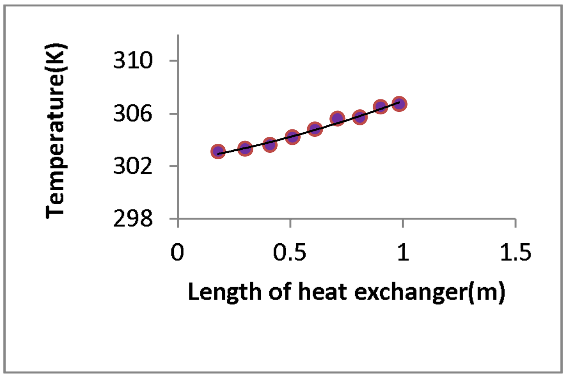
Fig(4-c) Axial surface temperature distributionalong outer surface of tube at(mr=0.718 and IP case)

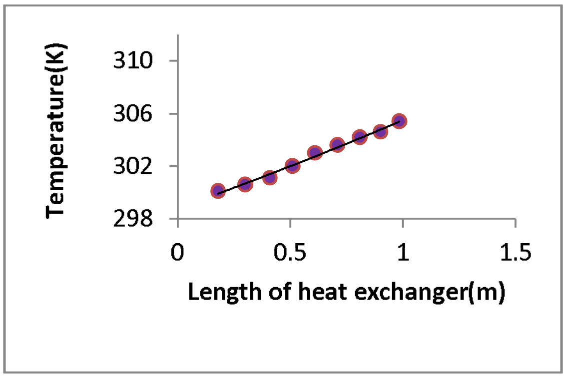




Fig(5-a) Axial surface temperature distribution along outer surface of tube at(mr=0.332 and OP case)

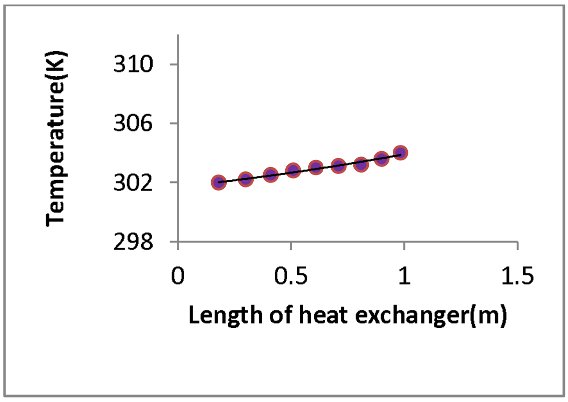
Fig(5-b) Axial surface temperature distribution along outer surface of tube at(mr=0.5 and OP case)

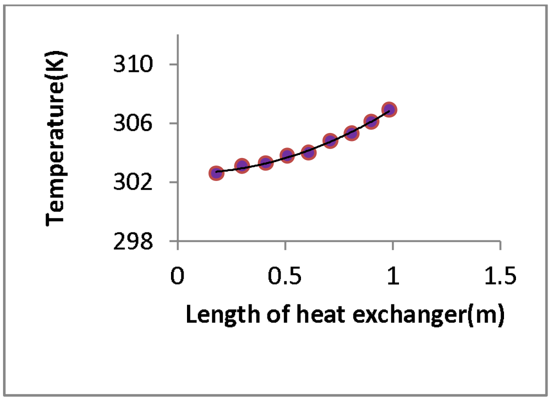




Fig(5-c) Axial surface temperature distribution along outer surface of tube at(mr=0.718 and OP case)

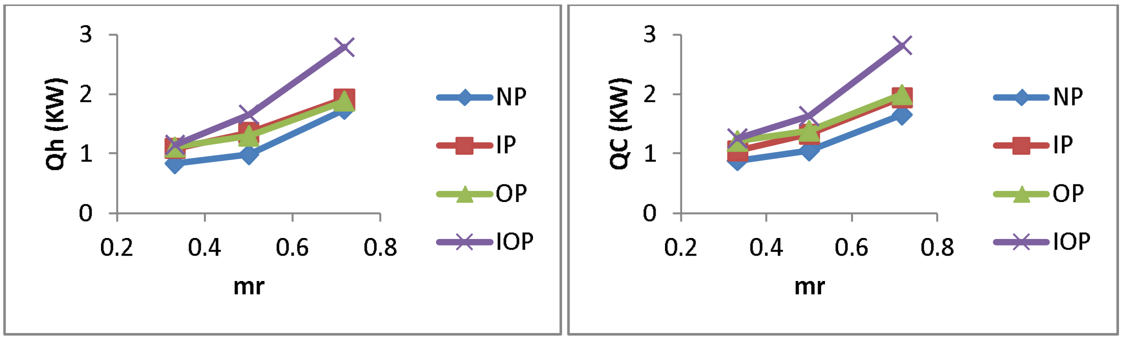
Fig(6-a) Axial surface temperature distribution along outer surface of tube at(mr=0.332 and IOP case)



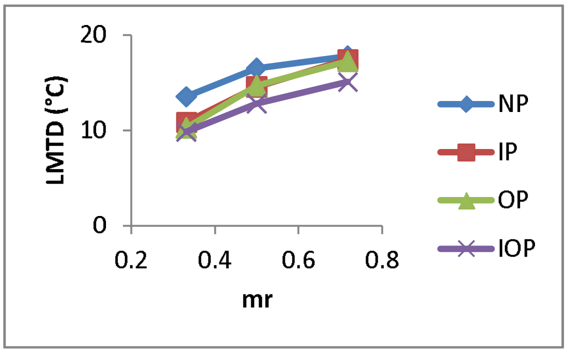


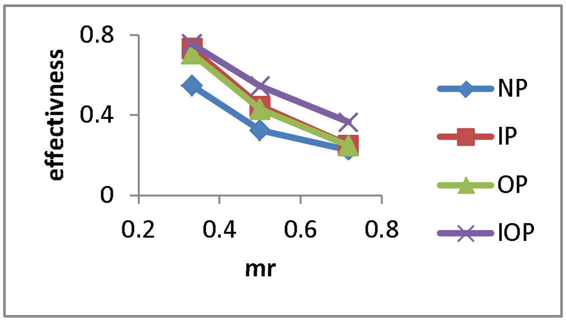
Fig(6-b) Axial surface temperature distribution along outer surface of tube at(mr=0.5 and IOP case)

Fig(6-c) Axial surface temperature distribution along outer surface of tube at(mr=0.718 and IOP case)



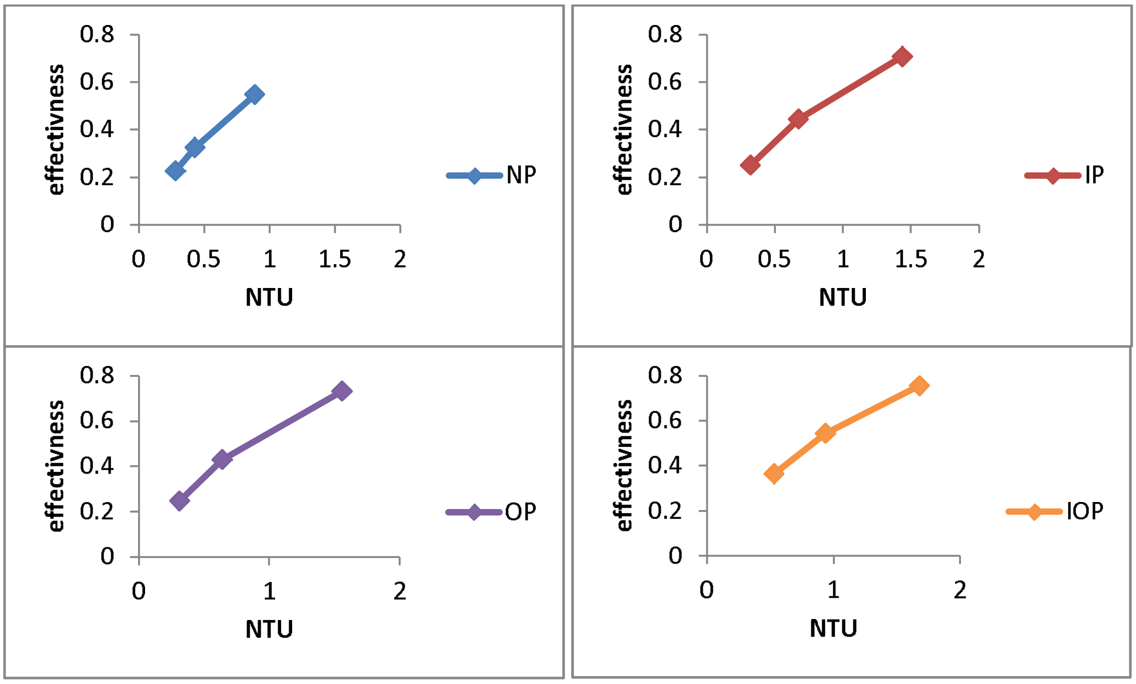
Fig(7) The effect mr and porous media on heat transfer rate at (mr=.0.332,0.5,0.718, Tci =25 °C, Thi=47°C)



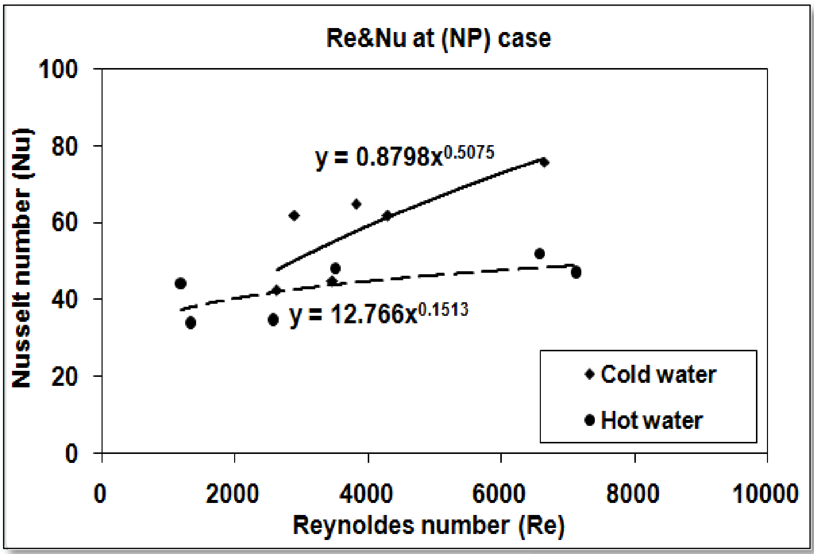


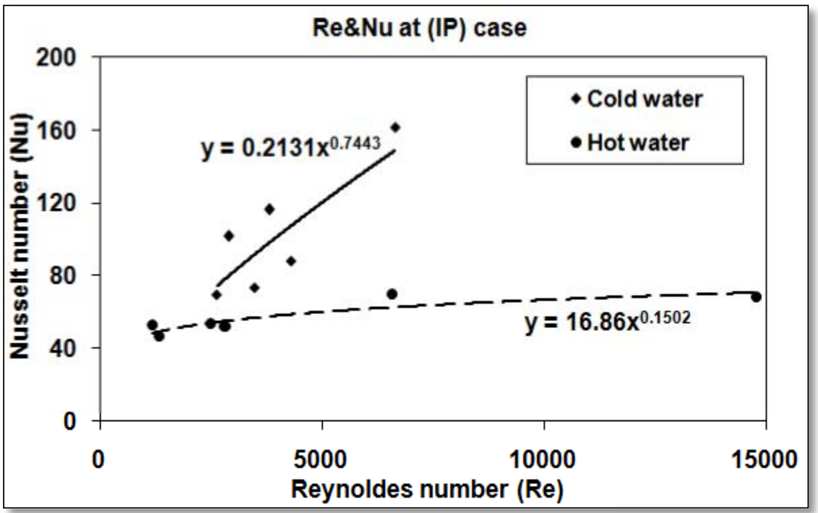
Fig(8) The effect mr and porous media on LMTD at(mr=.0.332,0.5,0.718, Tci =25 °C, Thi=47°C)

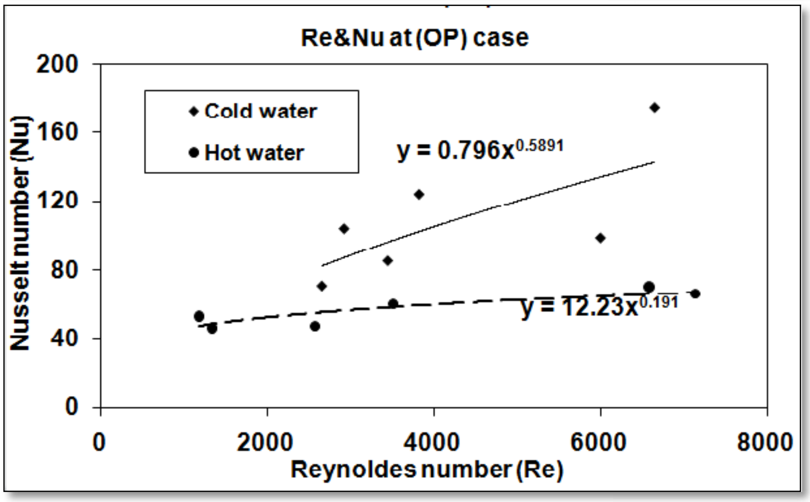
Fig(9) The effect mr and porous media on effectiveness at(mr=.0.332,0.5,0.718, Tci =25 °C, Thi=47°C).

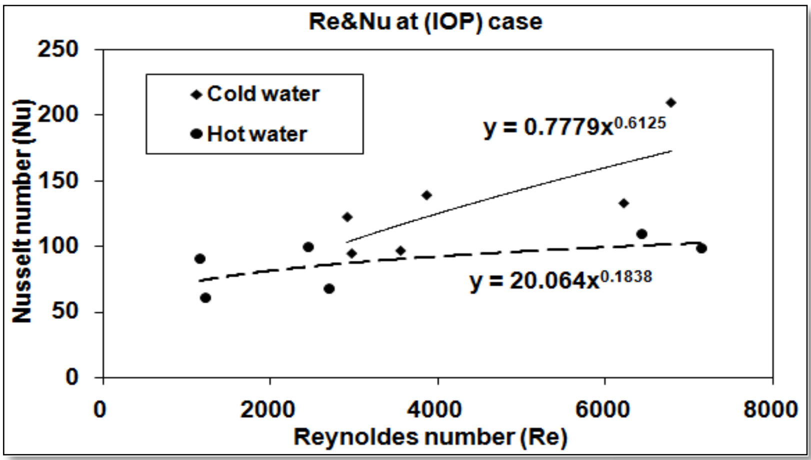


Fig(10) Experimental- effectiveness- NTU relation and effect of porous media at(Tci=25°C, Thi=47°C)

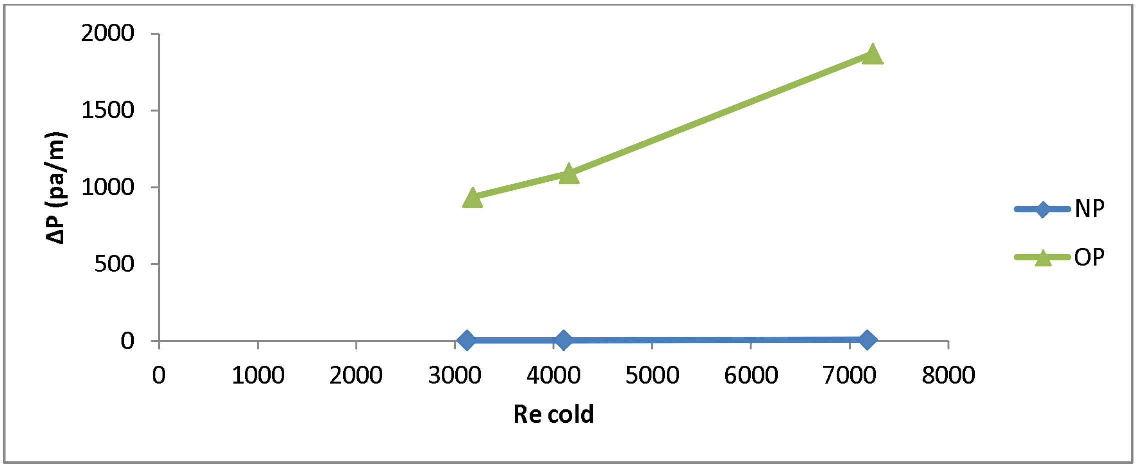

Re&Nu at (NP) case
y = 0.8798x^0.5075
y = 12.766x^0.1513
Cold water
Hot water
Nusselt number (Nu)
Reynoldes number (Re)
0
20
40
60
80
100
0
2000
4000
6000
8000
10000


Re&Nu at (IP) case
y = 0.2131x^0.7443
y = 16.86x^0.1502
Cold water
Hot water
Nusselt number (Nu)
Reynoldes number (Re)
0
40
80
120
160
200
0
5000
10000
15000


Re&Nu at (OP) case
y = 0.796x^0.5891
y = 12.23x^0.191
Cold water
Hot water
Nusselt number (Nu)
Reynoldes number (Re)
0
40
80
120
160
200
0
2000
4000
6000
8000



Fig(11) Nu-Re relation and effect of porous media –Experimental



Fig(12) Pressure drop-Re relations at L=0.5 m and (Tci=25°C, Thi=47°C)

Conclusions:

This work, present an experimental investigation into heat transfer parameter in counter flow heat exchanger enhanced by adding porous pad through it in three position (IP, OP, IOP). From the discussion of experimental results the following conclusions have been extracted:

1-The axial temperature profile of outer surface of inner tube decreases towards the ends of heat exchanger in the hot water direction and increases with decreasing mass flow rate ratio.

2- The axial temperature is also influenced by porous media and high increments at the cases: IOP < IP <OP < NP respectively. The increment ratio between (IOP,NP), (IP,NP) and (OP,Np) is (9.93 ,9.42 and 7.66) % respectively.

3-For the same inlet temperature for both fluids, outlet temperature decreases with increasing mass flow rate.

4- The heat transfer coefficient increase with increase mass flow rate and it is effected by using alumina , but the best value of heat transfer coefficient is obtained in (IOP) case. The lowest deviation (NP,IIOP) , (NP,OP) , (NP,IP) is (42.85,38.07,19.066)% respectively .

5- The heat transfer rate increases as the mass flow rate ratio increases also the heat load is effected by using alumina and the highest value are obtained when alumina is in double pipe .

6- Increasing the mass flow rate ratio lead to increasing LMTD. LMTD decrease when using porous media and the lowest are: IOP<IP<OP<NP.The reduction in devotion between (NP.IOP),The reduction in deviation between(NP,IOP) , (NP,OP) and (NP,IP) is (27.32, 24.09,19.8)% respectively.
7- Effectiveness decreases with increase in mass flow rate ratio but increases when using alumina as a porous media and the better value is obtained when alumina is in IOP, IP, OP and NP respectively.
8- Effectiveness Increased as the NTU increases.
9-Nu increases with increase in Reynolds number but the Nu is influenced by using porous media and it gives the better result get on it when alumina is used in double pipe.
10- Increase in the pressure drops when using porous media and when increasing mass flow rate ratio.
11- Adding porous media increases effectiveness of heat exchanger.

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