RESEARCHERID
THOMSON REUTERS

[Maid* 4(9): September, 2017] ISSN 2349-4506
Impact Factor: 2.785

Global Journal of Engineering Science and Research Management

EXPERIMENTAL STUDY OF HEAT TRANSFER ENHANCEMENT IN HEAT EXCHANGER USING POROUS MEDIA

Isbeyeh W. Maid*, Kifah H. Hilal, Jawaher K. Hassun
* Air- cond. Eng. Dept., Engineering Tech- Baghdad. College, Iraq
Mechanical Eng. Dept., Institute of Technology- Baghdad, Iraq
Air- cond. Eng. Dept., Engineering Tech- Baghdad. College, Iraq

DOI: 10.5281/zenodo.886915

KEYWORDS: heat exchanger- metallic pad- forced convection- pressure drop- performance ratio.

ABSTRACT
In present work experimental investigation of heat transfer enhancement in double pipe counter flow heat exchanger with insert metallic pad inner tube. Experimental work included the design of a tube in tube heat exchanger with the dimensions of (length 1.11m, 0.063m outer tube diameter and 0.031m inner tube diameter) Plain and saturated with pad heat exchanger . The examination were tested to evaluate their influence on effectiveness of heat exchanger, heat transfer coefficient, number transfer unit and pressure drop at steady-state condition. Water was used as a working fluid in the double pipe heat exchanger where hot water flow in inner tube and cold water flow in outer tube. The study was conducted at the hot water mass flow rates between (0.066-0.198 kg/s) and (0.1997 kg/s) cold water mass flow rate. The inlet temperatures of hot and cold water were (43 ^{o}C) and (18 ^{o}C) respectively. The results are obtained for range of Reynolds number to hot water (4862.9< Re<14633.8) and constant for cold water (Re=2912.2).

The experimental results show that the major effective factors on the axial temperature distribution of heat exchanger, the effectiveness, heat transfer coefficient and pressure drop are the mass flow rate and adding metallic pad, where, The inner heat transfer coefficient of heat exchanger increased with increase in Reynold number, heat transfer coefficient when add metallic pad (WP) the inner tube of heat exchanger higher of plain pipe, the enhancement factor of heat transfer coefficient in metallic pad comparison with plain case are (2.074). The effectiveness decreases with increase hot water Reynold number, and increase by 26.5% when use of metallic pad compared with plain tube (WOP). The performance ratios obtained are in the ranges of 0.1

INTRODUCTION
Heat exchangers are extensively used in several industry equipment's such as thermal power plants, chemical processing plants, air conditioning, equipment's, etc. 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 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 increase and leads to significant enhancement in heat transfer. Experimental study **Boomsma et al. [1],** experimental presented the heat transfer and pressure drop in metal foam heat exchanger manufactured from aluminum ally with average cell diameter (2.3mm).which foam were compressed and designed into compact heat exchanger measuring (40mm*40mm*2mm high), having a surface area to volume ratio (10000 m^2/m^3).they were setted and occupying the entire cross-section of the channel using water as the coolant. Heat fluxes ranged up to 688 kW/m^2 obtained from heater attached to aluminum foam through the heat spreader plate .Six types of metal foam were tested with permeability ranging between (44.4*10^{-10}-3.88*10^{-10} m^2). Nusselt increased with increasing coolant velocity and the aluminum foam heat exchanger (K=3.88*10^{-10} m^2) achieved the largest Nu values. **Hayes et al. [2],** Experimental and numerical investigated heat transfer and fluid flow in a matrix heat exchanger using porous media Two model were formed as numerical using finite- difference method and fluent computational fluid dynamics .The test section was consisted of flow channel (38 mm × 38 mm) with a porous media (porous fins) constructed of aluminum foam (k= 230 W/m.k) with two different pore size (10 ppI) (pores

in inch) and (50 PPI). The porous media dimensions were (0.0381 m * 0.0381 * 0.027 m) and heated from below by a heat source. Water was used as a working fluid at Reynolds number range between (100 < Re < 1150). The results were showed that the heat transfer coefficient evaluated from fluent was higher than Numerical and experimental ($h_{fluent} > h_{numerical} > h_{experimental}$) , the overall heat transfer increased as Re increased and porous media at (40 PPI) was greater (h) than (10 PPI) . **Narasimhan, et al. [3],** were investigated numerically by using finite volume method the effect of variable permeability of porous media (aluminum metal foam) pad in tube-to-tube inter-connectors in cross flow on the thermohydraulics of near-compact heat exchangers (NCHX, surface to volume ratio, α = 100–300 m^2/m^3. The cooling fluid with (Pr = 0.7) is under laminar flow (10 < Re < 100) through the NCHX with and without local porous medium inter-connectors of uniform permeability (UP), porous media case one (PMC1) and porous media case two (PMC2) with permeability ($K_i = 10^{-5}$ and 10^{-10} m^2 respectively) and variable permeability (VP), porous media case three (PMC3) and porous media case four (PMC4) with permeability (K_i varying along the flow direction from 10^{-5} to 10^{-10} m^2 and 10^{-10} to $10^{-0.5}$ m^2, respectively). At higher flow rates (Re > 70) it is shown that PMC4 registers less pressure-drop compared to PMC1, the UP case with highest K_i. Nusselt number variation for Re for VP cases (PMC3 and PMC4) is bounded by the two limiting UP cases PMC1 and PMC2, for identical porosity. Comparison between NCHX tube bank in the UP and VP cases is done using an overall enhancement ratio (ER), which was found to increase with increasing K_i of the PM inter-connectors. **Odabaee, et al. [4],** presented a numerical investigation to examine the 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 use 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 were 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. **Rezae.[5],** examined ten porous heat exchanger , four used metal foams and six used wire meshes of 10 and 40 PPI (pores per inch) to enhance the effectiveness of tube heat exchanger . The porous metallic material used external fins to increase the surface area for heat transfer. Each heat exchange manufactured of copper tube (6 mm outer diameter) and the overall length (482 mm) bending in (U) shape . Foam pieces were cut at the dimension (152 mm × 44 mm × 20 mm) . The screen wire mesh dimension (15 mm × 44 mm) positioning on the top of copper tube at single screen (10 PPI) and Double screen (40 PPI) on both side of tube . Distilled water passes through the copper tube to recovery heat from furnace. The water flow rate ranged between (0.013 kg/s – 0.032 kg/s) and furnace temperature between (300 – 800°C). The results showed that water coolant temperature raised for increasing furnace temperature and water mass flow rate. At wire mesh screen heat exchange the performance of having two screen was better than one at high temperature. For both wire mesh and foam at (10 PPI) was the highest due to air penetrated easily than (40 PPI) which increased heat transfer rate **Hilal et al .[6],** Studied experimentally forced convection heat transfer and pressure drop through square packed duct (12.5 * 12.5 * 100 cm.). The porous media made of (48) metallic wrapping coil unit at (ε= 0.98) and (K= 26 W/m. °C). The experiments were carried with Reynolds number (40339 – 54797) and constant heat flux (2.73 – 0.54 kW/m^2) placing on top surface , bottom surface or all surface in various conditions . The results showed that Nusselte number increase with increasing Reynolds number and increment percent of (Nu) between all surface duct heating and top surface duct heating only are (22.44%, 30.68%, 40.97%) at Reynolds number (54799, 48222, 40339) respectively and (16.94% , 24,115% , 33.03%) the increment percentage of (Nu) between top and bottom surface duct heating and top surface duct heating only at the same (Re) range . Packed Nusselte number is to be (1.2 , 1.19) times higher than empty ducts at heating all surface and top and bottom surface of porous duct respectively.

EXPERIMENTAL APPARATUS

The test heat exchanger which is consists double pipe configuration at concentric form, hot water flow inner tube and cold water flow annuli side as indicated diagrammatically in Fig.(1) and photographically in Fig.(2), dimensions of inner tube (31mm)outer diameter and (28mm) inner diameter, dimensions of annuli side (63mm) outer diameter and (58mm) inner diameter, (1110mm) length for external and internal pipes respectively, ,the configuration added at entrance and exit of heat exchanger respectively to remove eddies and obtained more uniform velocity profile. The heat exchanger is made of Aluminum because of its high thermal conductivity (202.4

W/m.K), available, cheap and for its easy machinability. Thermally insulated with one layer of (75) mm thickness of glass wool, which was mounted by means 4 of plastic tapes to reduce heat losses from the test section to the surrounding. Water circuits are controlled the water flow through the heat exchanger in control flows .the water come from two tanks each one is ($1m^3$),the first contains the cold water ,while the other contains hot water. Each circuit is used centrifugal pump for water circulation the specification of pumps are with head (30m) and volumetric flow rate (36 L/min) flow rate at 370W , 220V and 50Hz.

Controlling the cold water is done by make use of ice adding in tank according the temperature needed. To obtain hot water (3000 W) heating coil is used, the hot water temperature is controlled by thermostat setting with an error of ±1°C .Also the hot tank is insulated with 75 mm thickness of glass wool. Piping of water circuits are used piping rubber and four globe valves, two for each circuit. Also a globe valve is used for flow control. In the present experimental work, the temperature variation for the whole range of water 20 °C to 65 °C. Therefore, K-type thermocouple is used.

-Ten thermocouple to measure surface temperature at test section five at outer surface and five at internal surface of the tube. These ten thermocouple are fixed at distance of (500mm) in middle of heat exchanger.

- Two thermocouples were placed to measure temperature of cold and hot water before entering heat exchanger entering heat exchanger,
- Four thermocouple wire penetrating the heat exchanger are put to measure entrance and exit of hot and cold water
- Two thermocouples are placed to measure temperature of hot and cold water tanks.
- Two thermocouples inserting through the glass wool insulation to calculate the heat loss from the test section to surrounding by know the measure temperature drop and thermal conductivity of insulation.

Bourdon gauges are used to Measure the pressure of water, range of (0-160) kPa two pressure gauge fixed on inlet and outlet of heat exchanger to measure pressure drop. Two water flow meters are need to measure the water flow in each circuit, first one to measure flow rate of hot water and another to measure the flow rate of cold water, the specifications of hot flow meter and cold flow meter are [(0-20 LPM),(BLUE-WHITE),(F-400)].

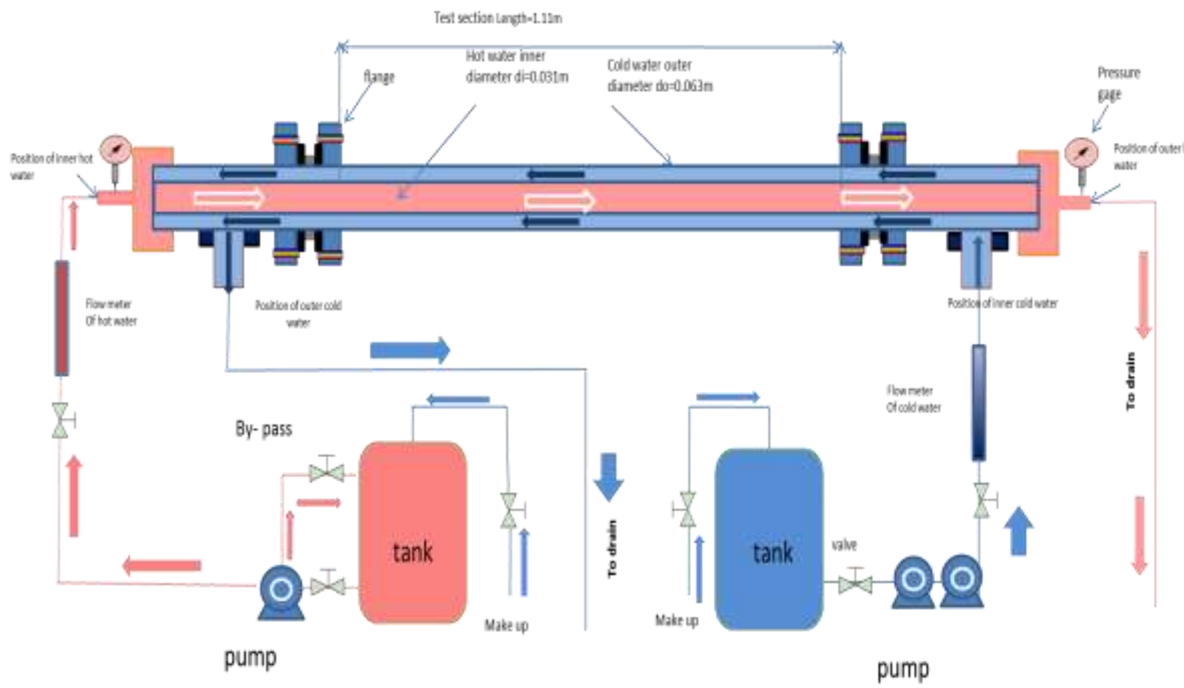


Figure (1) diagram of experimental apparatus

RESEARCHERID
THOMSON REUTERS
[Maid* 4(9): September, 2017] ISSN 2349-4506
Impact Factor: 2.785
Global Journal of Engineering Science and Research Management

Figure (2) Photograph of rig assembly

Metallic pad

Metallic pad dimension

The packed pad was made of metallic wrapping coil (MWC) (1mm) diameter as show in Figure (3) the ten of (MWC) were used to pad inner tube of heat exchanger. To obtain a certain porosity along the heat exchanger,(MWC) were arranged and compressed regularly in wire mesh housing (WMH) shown in Figure(4)to avoid any change in porosity during manufacturing test section and experimental. The (WMH) was divided into five equal part with diameter (27mm) and length (220mm) to facilitate the fill (MWC) process, then each part was close with a circular (WMH) pieces as seen in Figure(4) this five part were connect from its end and inserting inside the inner tube of heat exchanger to work as a porous media in this study. The wire mesh house(WMH)was made of commercial galvanized steel screen (wire diameter 0.5 mm , pore area 5.32 mm2 , density 2659 kg/m3 , thermal conductivity 164 W/m.K) [7].the wire mesh used as housing showing in Figure (5). The dimension of (WMH) was (1110mm) length and (84.78mm).the width of housing was equaled the inner tube circumference shape to insert tightly in the inner tube of heat exchanger as seen in Figure(6).

Figure (3) metallic pad

Figure (4) Wire mesh housing and metallic pad

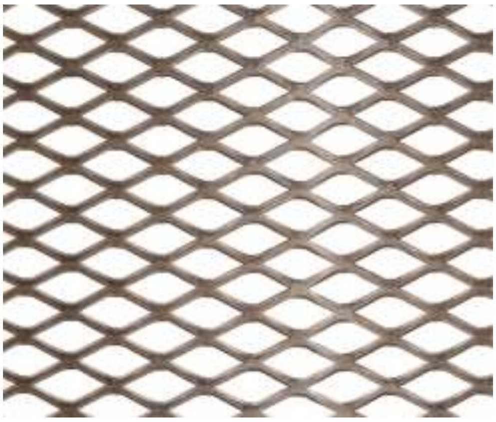


Figure (5) Wire mesh

Figure (6) Metallic pad insert inner tube

Thermal properties of metallic pad

The weight of each unit was measured by sensitive balance with a precision of (± 0.01 gram),the average weights of metallic pad unit are found to be (16 gram). A Measure cylinder with capacity (250±1 mL) was used to find the volume of each unit of metallic pad a known volume of water was place inside the cylinder then one unit metallic pad previously weighted was add .by measuring the total volume occupied by the water and by metallic pad unit and subtracting the initial volume of the water , volume of metallic pad unit could be obtained ,this measure is repeated for all unit of metallic pad used in this work .the average volume of metallic pad unit found to be (2.5 mL). Density was measured using the volume displacement method, by dividing the weight of metallic pad unit to volume water displaced .the average density is found to be (7945 kg/m3), and compart this average density with density of various kind of steel in Ref [21], which indicate that metallic pad made from Nickel steel (10Ni) with thermal conductivity (k=26W/m.K) Specific heat (CP=460 J/kg.°C).

***Porosity Measurements**:*

The (WMH) was divided into five equal part with diameter (27mm)and length(220mm)to facilitate the fill (MWC) process, take one five part and then finding its volume by a water displacement method at same method which measure water displacement when density measurements the volume water displacement is (33 mL)

The porosity is (MWC) pack pad used can be calculated as follow:

Test tube volume (V_t) = $(3.14/4) * 0.028^2 * 1.11 = 6.8313*10^{-4}$ m^3

Number the part (WMH) the fill (MWC) process insert in test section=5

Pack volume (V_p) $=5*6.6*10^{-5}= 33*10^{-5} m^3$

Porosity (ε) = (V_t- V_p)/ V_t=$(6.8313*10^{-4}-33*10^{-5})/ 6.8313*10^{-4}=0.951$

Heat Exchanger Analysis

To analyze heat exchanger this includes many important assumptions, in this study the working fluid is water for the two sides of heat exchanger. Therefore the flow characteristics are assumed to be as follows:

- Steady state conditions.
- Incompressible fluid and there is no change and in fluid phase.
- Perfect insulation of heat exchanger and heat transfer between the cold and hot fluids only.

Under the above assumptions, the heat dissipation of both sides are[8]:

$$Q_c = m_c Cp_C \; (T_{co}\text{-}T_{ci}) \quad (1)$$

$$Q_h = m_h Cp_h (T_{hi}\text{-}T_{ho}) \quad (2)$$

Where:

Q_h=Heat transfer rate from hot water (W)

Q_c=Heat transfer rate gained by cold water (W)

m_h=Mass flow rate of hot water (kg/s)

m_c=Mass flow rate of cold water (kg)

C_p= Specific heat of water at constant pressure (kJ/kg. °C)

T_{hi}=Hot water inlet temperature (°C)

T_{ho}= Hot water outlet temperature (°C)

T_{ci}=Cold water inlet temperature (°C)

T_{co}==Cold water outlet temperature (°C)

For heat exchanger analysis, it is convenient to share the product (m. C_p) of fluid in single quantity, which named the heat capacity rate, and can be rewrite for hot and cold streams as:

$$C_c = m_c Cp_C \quad \text{and} \quad C_h = m_h Cp_h \quad (3)$$

With heat capacity rates, equations (1) and (2) become as:

$$Q_c = C_C \; (T_{co}\text{-}T_{ci}) \quad \text{and} \quad Q_h = C_h (T_{hi}\text{-}T_{ho}) \quad (4)$$

The average heat transfer rate,Q_{av} used in the calculation is determined from the hot and cold water sides as follow: [9]

$$Q_{av} = \frac{Q_h + Q_c}{2} \quad (5)$$

The overall heat transfer coefficient, U, can be determined from:

$$Q_{av} = U_i A_i \, LMDT \quad (6)$$

$$A_i = \pi \, d_i \, L$$

Where

A_i = Inside surface area of inner tube m^2

d_i= Inner diameter of inner tube m

L= Length of tube m

LMTD=the logarithmic mean temperature difference °C

- The average heat transfer coefficient of inner tube, h_i, can be calculated from the average heat transfer rate obtained from[7]:

$$h_i = \frac{Q_{av}}{Ai(T_w - Ts_{av})} \quad (7)$$

Where: $Ts_{av} = \frac{(T_{s1} + T_{s2} + \cdots + T_{s9})}{9}$ and $T_w = \frac{T_{hi} + T_{ho}}{2}$

T_{s_av}=Heat transfer coefficient of hot water (W/m^2·°C)

T_{s_av}=Average Surface Temperature of hot water tube (°C)

T_{w}=Average Temperature of hot water (°C)

- The overall heat transfer coefficient, **U**. can be determined from:

$$U = \frac{Q_{av}}{A \times LMTD} \quad (8)$$

In order to develop a relation for the equivalent average temperature difference between the two fluids, the log mean temperature difference can be determined from:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (9)$$

For counter flow heat exchanger, $\Delta T_1 = (T_{hi} - T_{co})$ and $\Delta T_2 = (T_{ho} - T_{ci})$

Then, thermal resistance net-work can be determined formed as [8]:

$$R_{total} = R_i + R_{wall} + R_o = \frac{1}{hiA_i} + \frac{\ln(r_o/r_i)}{2\pi KL} + \frac{1}{hoA_o} \quad (10)$$

where: $A_i = \pi d_i L$ and $A_o = \pi d_o L$

and the thermal resistance of the tube wall in this case is:

$$R_{wall} = \frac{\ln(r_o/r_i)}{2\pi KL}$$

$$Q_{av} = \frac{LMTD}{R_{total}} = U_i A_i LMTD = U_o A_o LMTD \quad (11)$$

Thermal resistance in aluminum tube wall can be neglected $R_{wall} = 0.0000722836$ (°C/W) , so that $R_{wall} \approx 0$, and we can approximate the equality of surface areas ($A_i \approx A_o$)

- Calculation heat exchanger thermal performance indices, maximum possible heat transfer, Q_{max}, effectiveness, ε, and number of transfer unit, NTU, and C_r from the following relations[8]:

$$Q_{max} = C_{min}(T_{hi} - T_{ci}) \quad (12)$$

Where:

C_{min}= (ṁ cp) minimum heat capacity rate of cold or hot fluid (kJ/ °K).

$$\in = \frac{Q_{act}}{Q_{max}} \quad (13)$$

$$NTU = UA_i / C_{min} \quad (14)$$

RESULTS AND DISCUSSION

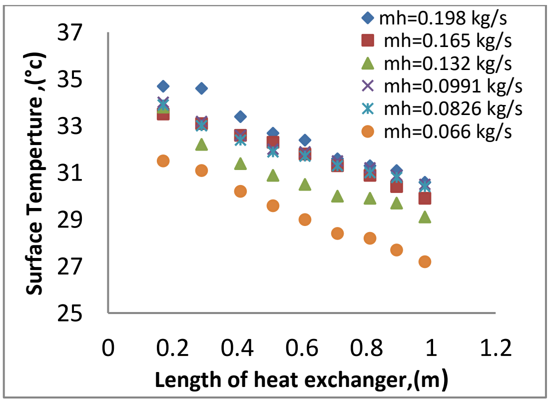
Results of the effect of porous media and hot water mass flow rate at constant cold water mass flow rate on the temperature distribution of counter flow of heat exchanger Figures (1) and (2) show the experimental measured temperature of inner tube surface distributions as a function of heat exchanger length and hot water mass flow rate, at constant cooling water flow rate. Figures **(1)**, shows the distribution for plain tube (case without porous media, WOP), under the above considered conditions. Figures **(2)** shows the temperature distribution when the heat exchanger is saturated with metallic porous pad, WP. These figures are indicated the decrease in surface temperature with axial distance due to heat transfer from hot water to the cold water. The surface temperature increased with increase in mass flow rate of hot water, in the same form for all cases, with (2 %) for maximum flow in comparison of the minimum one Also the surface temperature increase by (10.5%) in cases that use metallic pad as a porous media when compared with WOP case.

The increase in surface temperature with using metallic pad is due to increase of the turbulence that result in destroy of thermal boundary layer because of generated channeling near the wall surface and increase of radial thermal conductive as a result of the porous medium.

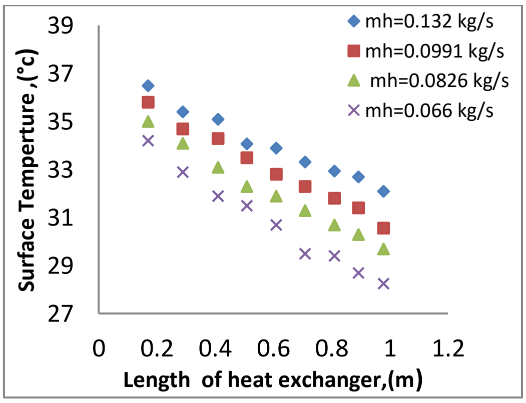
Figures **(3)** show effect of different hot water $\mathbf{Re_h}$ on inner tube heat transfer coefficient at Re_c=2912.2 of cold water, which flow in annular side. The experimental results show that coefficient is a function of Re_h as noted from figures. This because of any increase in the mass flow rate causes increase in velocity and which give more turbulence in water flow that lead to high heat transfer coefficient. Also, can be noted that heat transfer coefficient when add metallic pad in the tube, is improved in comparison of plain one, due to the increment in turbulence and its effect on thermal boundary layer. The enhancement in heat transfer coefficient with metallic pad in comparison with plain case, are (2.074). In Figure **(4)** the variation of heat exchanger effectiveness, Ɛ, is plotted as a function of a hot water $\mathbf{Re_h}$, The result show that the effectiveness decreases with increase hot water $\mathbf{Re_h}$

which mean that it has an adverse effect on effectiveness of heat exchanger. Physically increase hot water Reynolds number mean increase mass flow rate of hot water this lead to mass ratio (m_h/m_c) increase leading to less temperature fall in the hot stream and consequently worsens effectiveness of the heat exchanger. On the contrary, little hot mass flow rates lead to more temperature fall in the hot stream and consequently better the modified effectiveness of the heat exchanger. Also those figures shows the use of metallic pad as a porous media increase the effectiveness of heat exchanger when compared with plain one because the use of metallic pad causes resistance to water flow rate and destroys thermal boundary layer in addition the porous media increases the heat transfer in fluid that saturated with porous and as a result, between hot and cold water through the tube. The experimental results show that effectiveness is increased when porous media is use in double tube in the percentage of 26.5%. Figure (5) Shows the experimental relation between the hot water mass flow rate represented by Re_h and the heat exchanger number of heat transfer unit, NTU for deferent orientation of heat exchanger with and without porous media. NTU is decreased with increase in hot water Re_h because the mass flow rate for hot water increase the C_{min} which lead to decrease NTU. The experimental results show that NTU is increased when porous media is used in double tube of in the percentage of 27%. in **Figure (6)** shows the experimental relation between the effectiveness and NTU for the cases of plain tube and saturated with porous media. This figure shows that the effectiveness increases with increase in NTU because the effectiveness and NTU depend on the same parameters (C_{min}), so the change in any one of them leads to the same effect on the others. Also, this figure shows the effectiveness and NTU at saturated porous is greater than that plain case. Pressure drop in heat exchanger gives the main indication to the heat exchanger consumption of energy because it represent the power required to pump fluid through the heat exchanger.

Figures **(7)** show the relation between the pressure drop and the hot water **Re_h**. The experimental results show that pressure drop is a function of Re_h. This because of any increase in the mass flow rate causes increase in velocity and as a result increase the friction losses and turbulent eddies of water flow that lead to high pressure drops. Also, can be noted that pressure drops when adding metallic pad in the tube, is developed in comparison of plain one, due to the turbulence and losses of porous media. The enhancement in pressure drops with metallic pad in comparison with plain one, are 94 %. The quality of heat transfer enhancement concept is derived from the performance ratio or heat transfer efficiency. The variation of performance ratio with Re_h for experimental result is plotted in Figure (8) The performance ratios obtained are in the ranges of 0.1

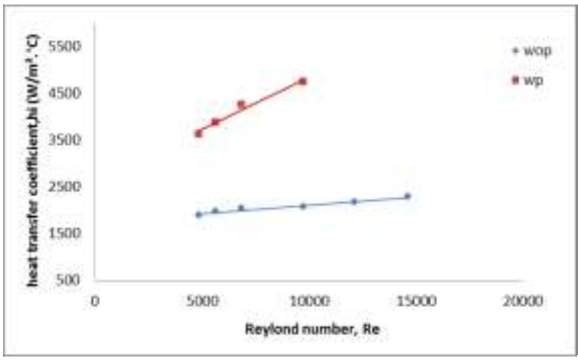


Figure(1) surface temperature distribution along outer surface of inner tube for plain tube

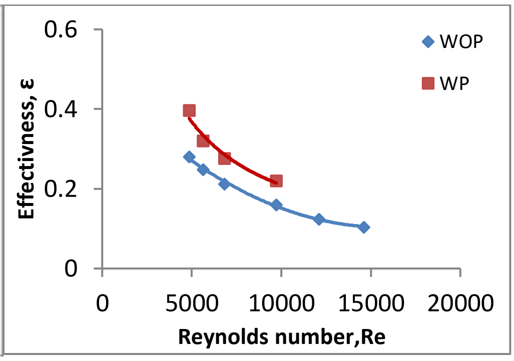


Figure(2) surface temperature distribution along outer surface of inner tube at WP case

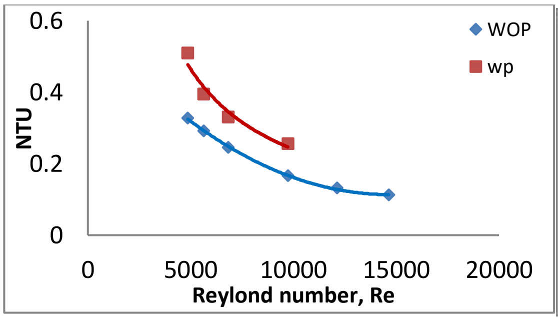
RESEARCHERID
THOMSON REUTERS
[Maid* 4(9): September, 2017] ISSN 2349-4506
Impact Factor: 2.785
Global Journal of Engineering Science and Research Management



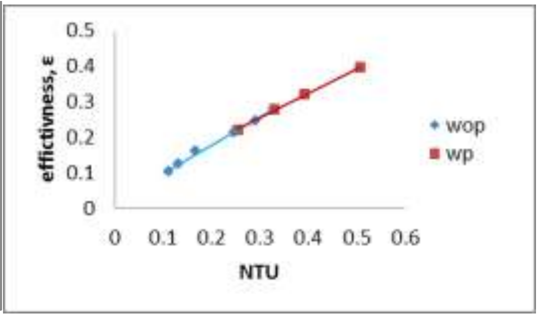
Figure(3) Experimental heat transfer coefficient as a function hot water Reynolds number



Figure(4) effect hot water Reynolds number and porous media on effectiveness



Figure(5) relation between hot water Reynolds number with number of transfer unit



Figure(6) variations of effectiveness with number of transfer unit

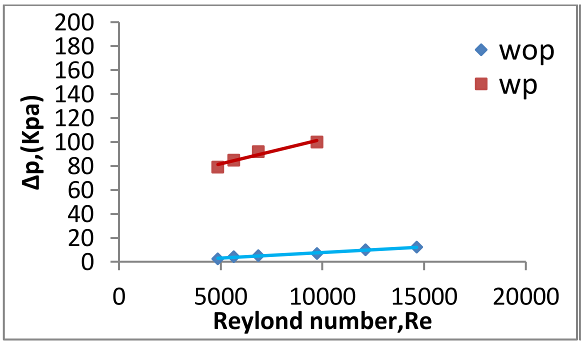


Figure (7) relation between Pressure drop and hot water Reynolds number

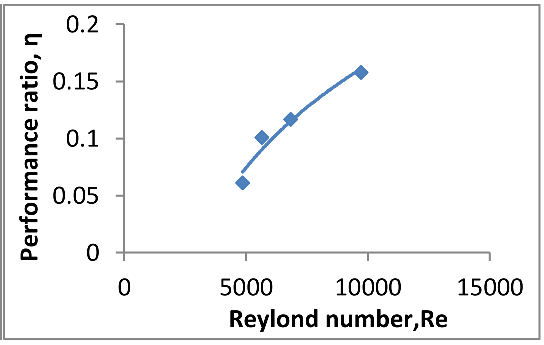


Figure (8) relations between performance and hot water Reynolds number

CONCLUSION

In this study, an experimental investigation into heat transfer parameter enhanced in counter flow heat exchanger by adding metallic pad at inside inner tube.

From the discussion of experimental results has arrived at following conclusions:

1. Decrease in surface temperature with axial distance due to heat transfer from hot water to the cold water. The surface temperature increased with increase in mass flow rate of hot water, in the same form for all cases, with (2 %) for maximum flow in comparison of the minimum one. Also the surface temperature increase by (10.5%) in cases that use metallic pad as a porous media when compared with WOP case.
2. The inner heat transfer coefficient of heat exchanger increased with increase in Reynold number, heat transfer coefficient when add metallic pad the inner tube of heat exchanger higher of plain pipe, the enhancement factor of heat transfer coefficient in metallic pad comparison with plain case are (2.074).

3. The effectiveness decreases with increase hot water Reynold number, and increase by 26.5% when use of metallic pad compared with (WOP).
4. NTU decreases with increase hot water Reynold number
5. The effectiveness increases with increase in NTU.
6. The pressure drop increases with increase in Reynold number along test section and increases with adding pad through inner tube metallic pad inner tube comparison with plain clean, are 94 %
7. Performance ratio increase with increase Reynold number the ranges of 0.1.

ACKNOWLEDGEMENTS
I would like to express my great thanks to all who supporting and given me most of their scientific, efforts, and

REFERENCES

1. Boomsma, K., Poulikakos, D.and Zwick ,F. " Metal foams as compact high performance heat exchangers" , Mechanics of Materials vol. 35, No. 1 , pp.1161–1176,2003.
2. Andrew M. Hayes , Jamil A. Khan , Aly H. Shaaban and Ian G. Spearing "The thermal modeling of a matrix heat exchanger using a porous medium and the thermal non-equilibrium model" , International Journal of Thermal Sciences,2007.
3. Narasimhan, A. , Sumithra, K. R., " Effect of Variable Permeability Porous Media inter-connectors on the thermo-hydraulics of heat exchanger modeled as porous media", Int. J. of Heat and Mass Transfer, vol.50,pp. 4052-4062,2007.
4. Odabaee ,M., Hooman, K. and Gurgenci, H. "Metal Foam Heat Exchangers for Heat Transfer Augmentation from a Cylinder in Cross-Flow" , Transp Porous Med ,vol.86,No.1 ,pp.911–923,2011.
5. Reza Rezaey," High Temperature Gas to Liquid Metal Foam and Wire Mesh Heat Exchangers" M.sc. Thesis, University of Toronto, Mechanical and Industrial Engineering,2012.
6. Kifah H. Hilal, Layth T. Fadhl and Sabah N. Faraj "Experimental Investigation of Heat Transfer and Pressure Drop in Square Metal Packed Duct with Different Boundary Heating" , Engineering and Technology Journal ,vol.30,No.6,2012 .
7. Kifah H .Hilal ,Adel M.Saleh and Muntaha H.Ebraheem "experimental study on heat transfer enhancement for porous heat exchange (Rectangular duct)" Engineering and Technology Journal, vol.32,No.11,2014 .
8. Yunus A. Cegel, "Heat Transfer a Practical Approach", second edition, McGraw-Hill, New York,pp.486-488,2004.
9. Veeresh Fuskele and Sarviya, "Experimental Investigation of Heat Transfer Enhancement Double Pipe Heat Exchanger Using Twisted Tape with Wire Mesh Screen", International Journal of Advanced Engineering Research and Studies, pp. 5-9, 2012.