Evaluation Of The Thermal And Hydraulic Performance Of A Double Pipe Heat Exchanger By Use Of Various Porous Structures

Ali Azeez Ali ALI, Ibrahim KOC, Ehsan Fadhil ABBAS

Abstract: The present study includes the evaluation of heat transfer enhancement of a double heat exchanger filled with five different porous materials, such as steel, ceramic, glass, plastic, and wood in the form of balls. The results are compared with the results of the heat exchanger without porous. Experiments were conducted on a double-core heat exchanger with dimensions of inner and outer diameter 20.7 and 22.2 mm and a length of 1.94 m of copper inner tube, inserted into the galvanized pipe with an inner diameter of 42.7 mm, wall thickness 2.15 mm, length 1.8 m, its outer surface insulated to prevent heat loss to the ambient. Experiments were performed under working conditions for hot and cold water flow ranging from 1 to 4 LPM and hot and cold water inlet temperature (60±2^{o}C and 28±3^{o}C), respectively, by two flow arrangements (parallel and counter flow). The results showed that steel balls give the highest effectiveness and number of transfer units, among other kinds of porous materials, where the ratio of improvement in their effectiveness relative to the non-porous state is about 60%. On the other hand, the use of porous materials leads to an increase in pressure drop, but the advantage of enhanced heat transfer was higher than the energy consumption by the pump. Where the highest pressure drop obtains in the ceramic case, which ranged from 20 to 178 mmHg.

Keywords : Double pipe heat exchanger, Concentric heat exchanger, thermal performance, pressure drop, porous medium, counter flow, parallel flow

1. INTRODUCTION

Heat exchangers are engineering equipment used to exchange energy between two or more liquids. Liquids can be separated by a solid material wall to prevent mixing. Heat exchangers are widely used in many applications, such as thermal power plants, cooling, and air-conditioning systems, chemical processing plants, nuclear power plants, dairy plants, etc. [1]. Thermal performance improved through many surface improvement ways; porous materials are one of these ways. The porous medium is using in many fields in industrial applications, such as solar energy storage systems, thermal insulation systems, and cooling operations for large electrical coils in high power generating devices, heat pipes, and heat exchangers in power plants [2]. Several studies conducted on the porous materials used as an enhancement surface in heat exchangers. Pavel and Mohamad [3], an experimental and numerical study investigated heat transfer enhancement for gas heat exchangers fitted with porous media under a condition of a constant and uniform heat flux on the pipe. The results compared with the non-porous material use situation. They have obtained the highest heat transfer rates using porous inserts but at the expense of reasonably low pressure. Lochan et al.[4], has investigated heat transfer improving in the cylindrical heat exchanger filled with porous media. The experiment carried out for different mass flow rates to examine the effect of the porous material on the heat transfer enhancement. They found a heat transfer rate increase with decreasing the porosity; on the other side, the result was compared with the CFD program and found agree between both results. Jamarani et al.[5], have studied numerically the investigation of the effect of high thermal conductivity of porous material filled into an annular gap of a double pipe heat exchanger. In their studies, simulation performed on the copper particle of diameter ranged from 1 to 6 mm and porosity from 0.8 to 0.95. The governing equations solved by the finite volume method using FORTRAN language. Results reveal that the overall heat transfer coefficient increases seven times, on another side, the heat transfer enhancement has no distinct difference with changing Reynolds from 10000 to 80000. Maid et al. [6], have experimentally conducted on the three cases used as a porous medium (inside the tube, outside the tube, and in both tubes). The results showed that increasing the rate of hot water flow to cold water flow in the case of the counter flow leads to reduced heat exchanger efficiency. Bhaskar and Choudhary [7], have experimentally investigated the effect of porous material on the enhancing of heat transfer and pressure drop through the porous material. The experiments were carried out on the heat exchanger using open-cell Aluminum foam material. The experiment results showed that the best heat transfer occurs at Reynolds number of 900, and the maximum effectiveness obtained about 0.3 when the flow rate is 0.2 m/s. Alwan et al. [8], experimentally investigated the unsteady natural convection heat transfer parameters through porous media. The experiments performed on the two phases material (solid and liquid) arranged as matrix balls insulted in two parallel sides and subjected to constant heat flux from the bottom. He found from the result, the heat transfer parameters (effectiveness and fluid parameters) are decreasing with increasing the time of heating. Shirvan et al. [9], have studied two-dimensional numerical simulations on heat transfer in a double pipe heat exchanger filled with porous material based on the Reynolds number, Darcy number, temperature difference between hot and cold fluids, and the porous thickness. Fluid flow is chosen based on the Darcy-Brinkman-Forchheimer model in the porous zone. Sensitivity analysis is performed to utilize the response surface systematically. They obtain that enhancement of the Nusselt number occurs due to an increase in Reynolds and Darcy numbers, while the effectiveness increased with Reynolds number, but it decreased with increase porous particle diameter. Sayehvand

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et al. [10], performed a numerical study on porous media use for enhancing heat transfer in three horizontal cylinders immersed in three kinds of porous media (rocks, sponges, and woods), which has irregular shapes and sizes. Three isothermal cylinders in a staggered configuration in isotropic packed bed investigated, the experiments conducted under laminar forced convection. The result of porous cases compared with an empty channel. The results showed that the porous cases give a good enhancing in Nusselt number. Bargh et al. [11], have studied experimentally change in hydrodynamic parameters, improvement of heat transfer by porous media in the channel, as well as pressure drop, resulting from porous media in a double pipe heat exchanger filled with a different arrangement porous media. The results of this study show that the fully filled duct of the porous media has the best improvement for heat transfer (in both laminar flow and turbulence). The aim of this study is to investigate the promotion of heat transfer and pressure drop in the heat exchanger using porous media. For this purpose, five different types of porous materials used in the annular space between the inner and outer tubes, such as balls (ceramic, steel, glass, plastic, wood). Experiments conducted in both parallel counter flow arrangements under similar working conditions. The results obtained from the porous mediums compared with the results of the normal state of the heat exchanger (without porous material).

2. MATERIAL AND METHODS

2.1 Experimental apparatus and procedures

A concentric double-tube heat exchanger was used in this study illustrated in Fig.1 and Fig.2 shows a schematic diagram of connecting the heat exchanger components indicated in Table 1. It consists of two different tubes, the inner pipe is copper of 20.7 and 22.2 mm diameter and 1.94 m length and the outer pipe made of galvanized pipe with dimensions of inner and outer diameter 42.7 and 47 mm and a length of 1.8 m. Both pipes are tightly assembled at two ends to prevent water leakage, and the outer surface of the galvanized pipe insulated with the glass wool to avoid heat loss to the surrounding.

***Fig.1** A Photo of the heat exchanger used in the present*

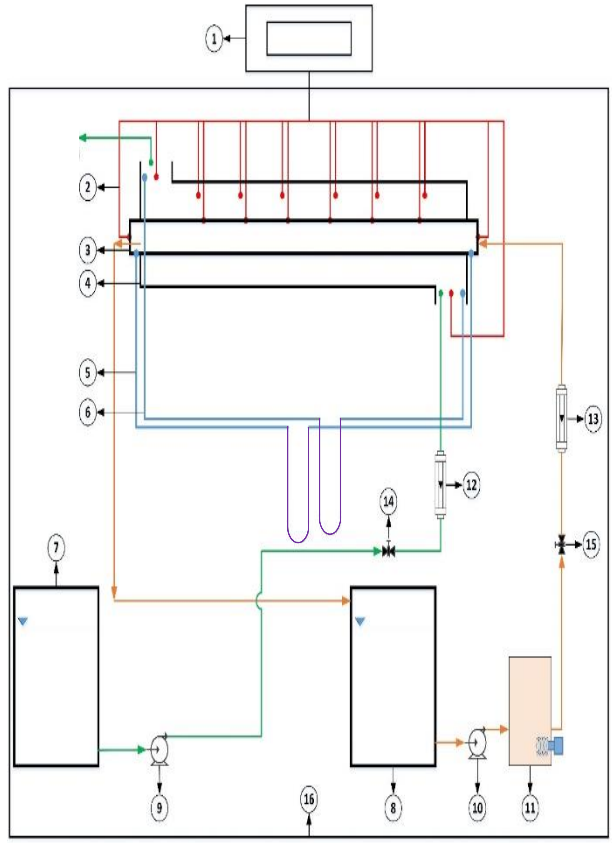


***Fig. 2** Schematic diagram of connecting heat exchanger components*

No.	Part name	No.	Part name
1	Temperature data logger	2	Thermocouple sensors type K
3	Inner tube	4	Outer tube
5	Hot side U type manometer	6	Cold side U type manometer
7	Coldwater storage tank	8	Hot water storage tank
9	Coldwater pump	10	Hot water pump
11	Boiler	12	Cold water rotameter
13	Hot water rotameter	14	Cold water gate valve
15	Hot water gate valve	16	Body frame

Two pumps of (327.85W) are used to circulating water in the heat exchanger, one used to circulate hot water in a closed cycle and the other cold water that acts as an open cycle. Two rotameters used to adjust cold and hot water flow rates with accuracy (±1%). In order to measure the temperature, a multichannel data logger type (Applent AT4516) with accuracy (0.2%±2^{o}C) was used to record the temperature at 16 locations inside the heat exchanger using K type thermocouples, and the pressure drop on either side of the heat exchanger has measured by two manometers type U as show in in Fig .2. The required experiments performed according to the following procedures based on the working conditions shown in Table 2:

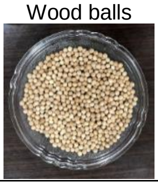
***Table 2.** Working conditions*

The name of the variable	Allowed range
Inlet cold water	28^{o}C ± 3^{o}C
Inlet hot water	60^{o}C ± 2^{o}C

Coldwater flow rate	1 to 4 LPM
Hot water flow rate	1 to 4 LPM

- The heat exchanger prepared according to the parallel-flow / counter-flow.
- The experiment begins in each case after the hot water temperature reaches 60 ° C, and it flows into the inner tube in order from 1 to 4 LPM in four steps.
- Coldwater flows into the annular space or on porous materials at a flow rate of 1 to 4 LPM in four steps at each hot water flow rate.
- The data recording time in the data logger is set every five minutes at a cold water flow rate to ensure the heat exchanger reaches a stable state, and the data collected is as accurate as possible when used in mathematical equations.
- When the experiment is complete, the cold flow path is only reversed to change the flow arrangement
- Upon completion of the above procedures, the annular space of the heat exchanger is filled with steel balls, and the above procedures have been followed to the execution experiment in this case.
- With the same procedures, experiments on other porous media (ceramic, glass, plastic, wood) have carried out sequentially. Table 3 shows information about these porous media used.

Table 3. *Specifications of porous material used in experiments*

Type and photo	Average diameter (mm)	Mass (g)	Porosity (-)
Ceramic balls 	5.303	1630	0.397
Steel balls 	6.296	8505	0.394
Wood balls 	5.892	670	0.375
Plastic balls 	8.078	2065	0.409
Glass balls 	5.92	2840	0.374

2.2 Data reduction

The heat transfer in the heat exchanger may be calculated based on two methods; the first one is by applying the energy conservation equation, which is expressed in Eq. (1), and the other method is by using a heat transfer rate, which is expressed in Eq. (3) [12].

$$q = \dot{m}_h Cp_h (T_{h,i} - T_{h,o}) \quad \text{(1-a)}$$

or

$$q = \dot{m}_c Cp_c (T_{c,o} - T_{c,i}) \quad \text{(1-b)}$$

where

$$\dot{m} = \rho \dot{V} \quad (2)$$

$$q = UA\, \Delta T_{lm} \quad (3)$$

where

$$\Delta T_{lm} = \frac{\Delta T_R - \Delta T_L}{\ln \frac{\Delta T_R}{\Delta T_L}} \quad (4)$$

for case of counter flow

$$\Delta T_R = T_{h,i} - T_{c,o} \text{ and } \Delta T_L = T_{h,o} - T_{c,i}$$

while, for case of parallel flow

$$\Delta T_R = T_{h,i} - T_{c,i} \text{ and } \Delta T_L = T_{h,o} - T_{c,o}$$

$$A = \pi\, d\, L \quad (5)$$

In order to estimate the thermal performance of the use of porous materials, the effectiveness should be determined and determined as the ratio between the actual energy transferred to the maximum energy transferred [12]:

$$\varepsilon = \frac{q}{q_{max}} \quad (6)$$

where

$$q_{max} = (\dot{m}C_p)_{min} (T_{h.i} - T_{c,i}) \quad (7)$$

$$(\dot{m}C_p)_{min} = \begin{cases} (\dot{m}C_p)_h & if\ (\dot{m}\,Cp)_h < (\dot{m}\,Cp\ )_c \\ (\dot{m}\,Cp)_c & if\ (\dot{m}\,Cp)_c < (\dot{m}\,Cp)_h \end{cases} \quad (8)$$

In general, effectiveness is a function of three parameters as described below:

$$\varepsilon = f\left(NTU, \frac{(\dot{m}\,Cp)_{min}}{(\dot{m}\,Cp)_{max}}, flow\ arrangement\right) \quad (9)$$

where

$$NTU = \frac{UA}{(\dot{m}\,Cp)_{min}} \quad (10)$$

The physical properties of both fluids, such as density and specific heat, were calculated at a bulk temperature (T_b). Expressed as

$$T_{b,h} = \frac{T_{h,i} + T_{h,o}}{2} \quad \text{and} \quad T_{b,c} = \frac{T_{c,i} + T_{c,o}}{2} \quad (11)$$

2.3 Experimental error analysis

The results should be tested to confirm uncertainty using the experimental error method because it contains three types of errors, such as instrument calibration, bias errors, and random errors [13][14]. Table 4 shows the accuracy and resolution of the measuring devices used in this study.

Table 4. *Measuring devices uncertainty*

Devices	Resolution	Accuracy
Caliper	0.01mm	±0.02mm
Flow meter	-	±0.5%LPM
Temperature data logger	0.1^oC	0.2^oC

Experimental error simulation was performed based on the following procedures (Figliola and Beasley, 2011):

i. Measurements uncertainty

The bias error (B) is calculated as:

$$B = \pm\sqrt{\left(\frac{1}{2} \times Resolution\right)^2 + (Accuracy)^2} \quad (13)$$

The average of measuring values (x_i) evaluated as:

$$\bar{x} = \frac{i}{N}\sum_i^n x_i \quad (14)$$

and standard deviation (σ_x)of sample distribution

$$\sigma_x = \left[\frac{1}{N-1}\sum_{i=1}^{n}(x_i - \bar{x})\right]^{1/2} \quad (15)$$

The mean standard deviations ($\bar{\sigma}_x$)for the sample was deduced from

$$\bar{\sigma}_x = \frac{\sigma_x}{\sqrt{N}} \quad (16)$$

The overall precision error(Px) was calculated using the student-t distribution on a 95% confidence interval with a score (N-1) of freedom and can be written as follows:

$$Px = \bar{\sigma}_x \times t_{(N-1),95\%} \quad (17)$$

The 95% confidence uncertainty(U_x) calculated from:

$$U_x = \pm|B^2 + Px^2| \quad (18)$$

and relative uncertainty is calculated by:

$$\frac{U_x}{x}\% = \pm\left(\frac{U_x}{x}\right) \times 100\% \quad (19)$$

The relative uncertainty for geometry was evaluated based on the relations above as:
Inner hydraulic diameter= ±0.106%
Annular hydraulic diameter = ±0.17%
Surface heat transfer area = ±0.116%
and relative uncertainty of the measuring devices was evaluated as:
flow meter = ±0.68%, and
temperature data logger =±2.67%

ii. Uncertainty estimation for water properties

In order to evaluate the uncertainty error of the physical properties of water in this study, we randomly selected a trial of 4 LPM of hot water flow rate based on the relation expressed below [15].

$$U_\varphi = \pm\frac{1}{2}\left|\varphi_{Tb,max} - \varphi_{Tb,min}\right| \quad (20)$$

where φ is water property
Calculation results are shown in Table 5.

***Table 5**. Uncertainly and relative uncertainty for water properties*

Properties of water	U_φ	$\frac{U_\varphi}{\varphi}\%$
Density, kg/m^3	±1.454	±0.15
Specific heat, J/kg.^{o}C	± 0.787	±%0.01
Dynamic viscosity, Pa.s	±2.38×10^{-5}	±4.6
Thermal conductivity, W/m.^{o}C	±0.003	±0.64
Prandtl number	±0.167	±5

iii. The overall relative uncertainty in the heat rate.

It is calculated from [13],

$$\left(\frac{U_Q}{Q}\right) = \pm\frac{1}{2}\left[\left(\frac{U_V}{V}\right)^2 + \left(\frac{U_\rho}{\rho}\right)^2 + \left(\frac{U_{cp}}{cp}\right)^2 + \left(\frac{U_{\Delta T}}{\Delta T}\right)^2\right]^{1/2} \quad (21)$$

and its result is 1.5%.

3. RESULTS AND DISCUSSION

The current analysis is limited to the study of the allocation of porous material types on the thermal performance and pressure drop of a concentric double-tube heat exchanger in two kinds of flow arrangements, such as parallel and counter flow, and the results compared with a normal heat exchanger (without porosity of materials). In order to investigate the objective of this study, a series of (192) sequential experiments were conducted to cover five types of porous materials and the normal state of the heat exchanger (without porous media), as shown in Table 3 and operating conditions in Table 2.

3.1 Thermal performance

Experimental data are simulated for each flow arrangement (parallel and counter), and results indicate that increased (ε) when NTU increasing. Figs. 5 to 8 shows the results of four volumetric flow rates for hot water in parallel flow tests, which ranged from (1 to 4) LPM. They noted that the relationship between (ε) and (NTU) in most tests was linear. The effect of porous materials has been observed, as the normal state of the heat exchanger in four tests gives a lower range of (ε) corresponding to NTU than that porous medium, their (ε) and NTU does not exceed 0.44 and 0.0114, respectively, at both V_h and V_c is 4 LPM as shown in Fig. 8, whereas porous mediums have a clear effect on improving (ε) and NTU, they are about (0.683 and 0.0198), respectively, which were obtained in plastic balls with the same volumetric flow as shown in Fig. 8, where the improvement was about 55% in (ε) and 73.7% for NTU. As for the thermal performance of porous materials in these tests, found with increasing hot water flow rate leads to convergence of values (ε) and NTU, after 3 LPM of hot water shown in Fig. 7, the expression curve (NTU and ε) changes to an almost linear relationship. From these four tests, it observed that the (ε) and NTU values strongly depend on the cold water flow rate, Plastic balls can be considered the best porous material among other material, it gives the highest amounts of (ε - NTU) as shown in Fig. 8, because of its large diameter and high porosity. In general, steel balls can be considered the best media, where their effectiveness ranges from 0.45 to 0.647. Because it has the smallest size and highest thermal conductivity compared to other types, it makes up a higher heat exchange area and heat flow.

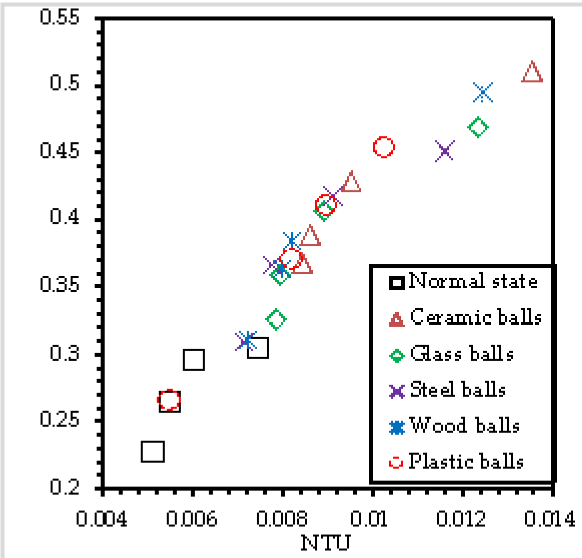


***Fig. 6** Relation between e and NTU for parallel flow when $_h$ is 2 LPM*

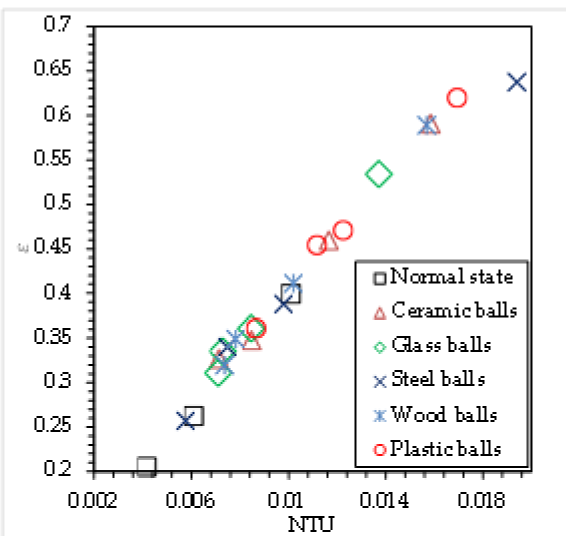


Fig. 7 Relation between ε and NTU for parallel flow when $\dot{V}_h$ is 3 LPM

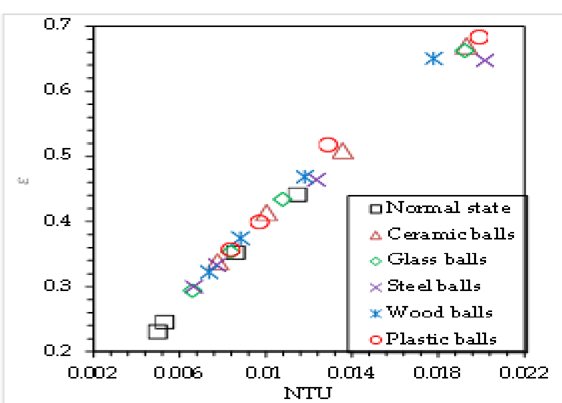


Fig. 8 Relation between ε and NTU for parallel flow when $\dot{V}_h$ is 4 LPM.

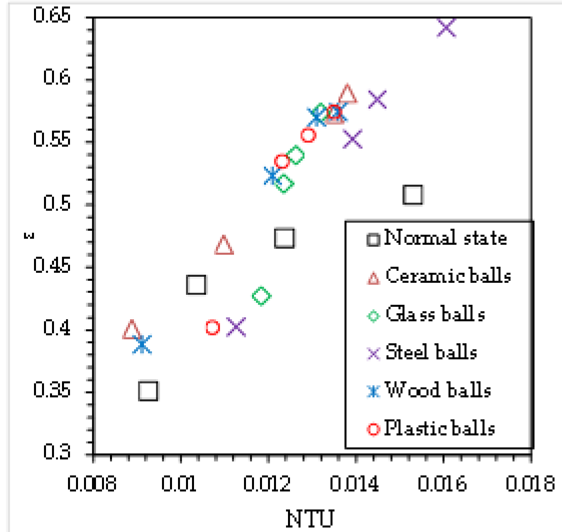


Fig. 9 Relation between ε and NTU for counter flow when $\dot{V}_h$ is 1 LPM.

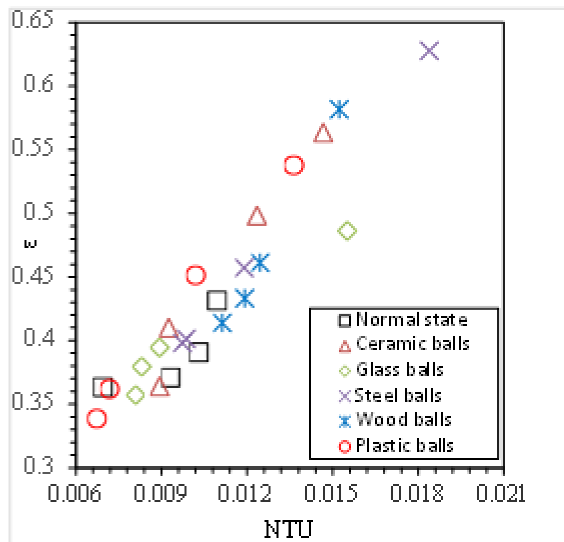


Fig. 10 Relation between ε and NTU for counter flow when $\dot{V}_h$ is 2 LPM

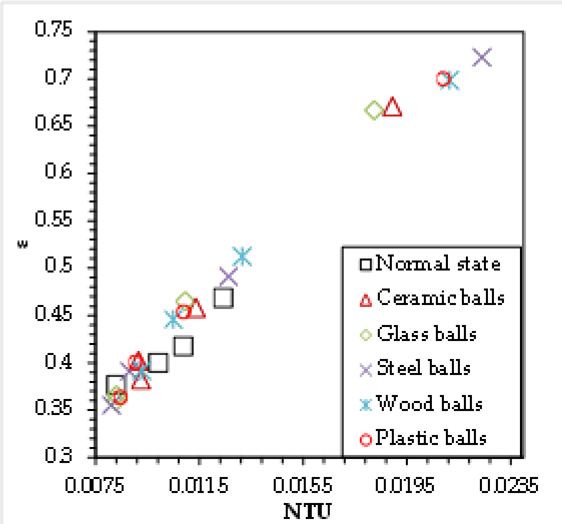


Fig. 11 Relation between ε and NTU for counter flow when $\dot{V}h$ is 3 LPM

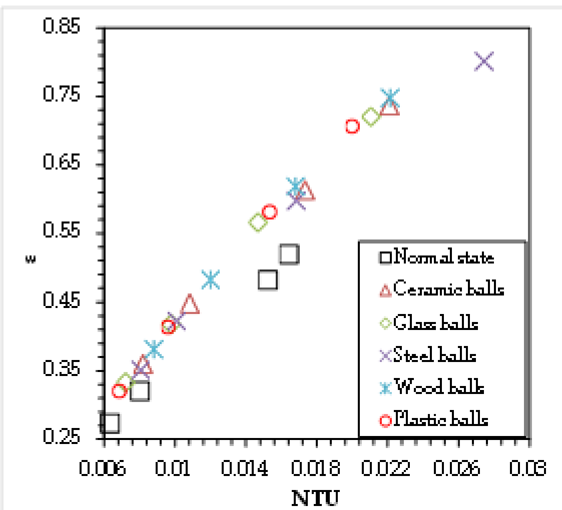


Fig. 12 Relation between ε and NTU for counter flow when $\dot{V}_h$ is 4 LPM

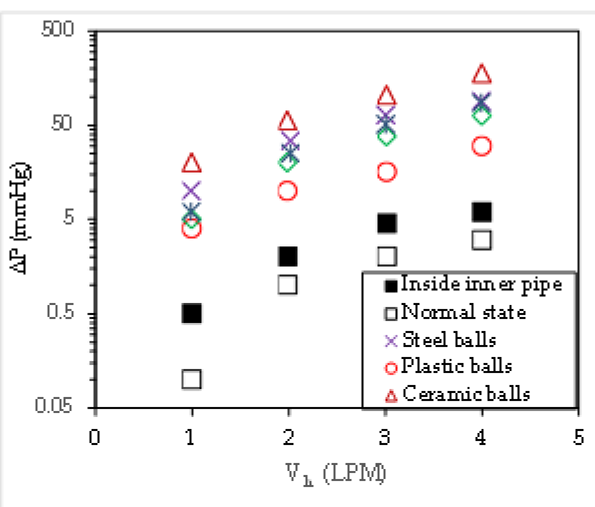


Fig. 13 Variation of pressure drop relative to the fluid flowrate

The same procedures applied to the counter flow during the tests; there were similarities between the results of the corresponding tests between the two flow arrangements as shown in Figures 9 to 12. From these figures observed that the increase of hot water flow rate leads to the relationship between (NTU and ε) becoming almost linear and convergence with porous medium results obtained when the hot water flow rate reached 4 LPM. On the other hand, the heat exchange performance was not significantly affected in the normal state with the change of hot water flow rate when compared to the cases in which porous materials used, where it did not exceed the max. ε and NTU (0.52 and 0.016) respectively. There is a good improvement in the thermal performance of the heat exchanger that occurs with a porous media in counter flow arrangements when compared with the parallel flow. In these tests, when the hot water flow rate increases to 4 LPM, steel balls give a higher ε and NTU among other types of porous materials, which ranged from 0.63 to 0.80 and 0.018 to 0.027 for ε and NTU respectively. Because it has the smallest size and highest thermal conductivity compared to other types, it makes up a higher heat exchange area and heat flow. Each of wood, ceramic, and glass balls gave efficiency in the range of 0.75, except for plastic balls that have been given the lowest values of effectiveness, reverse to the steel balls; it has the largest size and higher porosity, which leads to less of the heat transfer area and flow resistance. The results were compared to a similar experimental work that was used in ceramics, steel, and glass in their work, but hot water was flowing on porous media [16]. The result gave a good agreement between the two studies.

3.2 Pressure drop

The purpose of adding a porous medium to a heat exchanger is to enhance the rate of heat transfer, but on the other side, this process actively contributes to increasing the pressure drop in an annular side, which in turn increases the pumping power. Fig. 13 shows the pressure drop on both sides of the heat exchanger concerning the flow rate for all cases. For the hot side or inner tube, the variation in pressure drop in all cases was the same. Since the operating conditions have not changed significantly, so the physical properties of hot water remain the same concerning the working condition. The ceramic test gives the highest pressure among other types of porous materials, ranging from 20 to 178 mmHg, due to two factors, the first one is the porosity property because it has the minimum porosity and the second is the surface roughness because it has a high roughness, for this reasons the pressure drop decreases in porous materials gradually when porosity increases, as shown in plastic case, which gives the lowest pressure drop levels, that did not exceed 30 mm Hg, which did not exceed 30 mmHg. The result of the pressure drop compared to similar work (Abbas and Mohammed, 2019) gives a good match between them.

4. CONCLUSIONS

Based on the results obtained from 192 tests carried out on a double-tube heat exchanger using five different porous materials in order to compare these results with the result of the normal sate, the maximum values of ε, NTU, and ΔP have been selected at V_h is equal to 4 LPM, as shown in Table 6.

***Table 6**. Show the maximum values obtained from the experiments.*

Porous type	Parallel flow		Counter flow		Pressure drop
	ε	NTU	ε	NTU	
Normal state	0.441	0.0114	0.519	0.0164	3
Plastic	0.683	0.0198	0.705	0.012	30
Steel	0.647	0.0201	0.8	0.0273	86
Ceramic	0.662	0.0192	0.736	0.022	173
Glass	0.66	0.019	0.72	0.021	63
Wood	0.65	0.0178	0.747	0.0221	86

The result is shown in Table 4, indicated that the counter flow arrangement was an efficient than parallel flow, and it has a higher ε and NTU than the parallel flow in all cases. The steel ball is the best porous material in both flow arrangements, as it gives acceptable effectiveness of not less than 50% and up to 80% with a cold water flow rate of 4 LPM. Furthermore, the use of porous materials leads to increased pressure drop, but the advantage of enhanced heat transfer is higher than the power consumption of the pump operation, where the highest pressure drop obtained at ceramic balls. Although the steel is a best porous, but the plastic has higher effectiveness than steel in parallel flow around 5.6%.

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Nomenclature

A: Area, [m^2]
Cp: Specific heat, [J/kg.°C]
d: Hydraulic diameter, [m]
L: Lenth of the pipe, [m]
LPM: Liter per minute.
$\dot{m}$: Mass flow rate, [kg/s]
NTU: Number of transfer units
ΔP: Pressure drop [mmHg]
q: Heat transfer rate, [W]
T: Temperature, [°C]
ΔT: Temperature difference, [°C]
U: Overall heat transfer coefficient, [W/m^2. °C]
$\dot{V}$: Volumetric flow rate, [m^3/s]
$\dot{V}_h$: Volumetric flow rate for hot water,[Liter/minute]
ρ: Density, [kg/m^3]
ε: Thermal effectiveness

Subscript:

c: Cold water
h: Hot water
i: Inlet
L: Left side
Lm: Logarithmic mean
max: Maximum
min: Minimum
R: Right side
o: Outlet