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Enhancement of Double-Pipe Heat Exchanger Effectiveness by Using Porous Media and TiO_2 Water

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ABSTRACT

In this paper, the rate of heat transfer by forced convection in a counterflow heat exchanger, at turbulent flow conditions were investigated experimentally, using porous media and TiO2 Nanofluid to observe the behaviour of heat transfer with flow rate and volume concentration of nanoparticles t enhance heat transfer through it. 3 mm Steel balls (ε=39.12%) as a porous media completely filled to the inner pipe (core pipe). The cold and hot water are used as working fluids through the inner and outer pipes. Then using, the TiO2 nanofluid instead of cold water flowing through the porous pipe to enhance heat characteristics. The effects of operating parameters include flow rate (4 LPM, 6 LPM, and 8 LPM), Reynolds number between (3000 – 7000), and nanoparticle volume fraction (0.001, 0.002 and 0.003) on Convective heat transfer co-efficient and Nusselt number. Effective thermal conductivity is increased when the nanoparticle volume fraction is increased. The heat transfer coefficient increases with decreasing nanofluid temperature, but the heating fluid's temperature has no significant effect on the nanofluid's heat transfer coefficient. The results show that porous media and TiO2-based nanofluid's improve heat transfer at flow rate of 4 LPM by 35.4% and improve NTU and effectiveness at flow rate of 4LPM by 12.4%, and 24%, respectively, when compared to pure water without porous media. This improvement in thermophysical properties yielded high heat transfer of heat exchangers used in process industries.

1. Introduction

A heat exchanger's thermal resistance can be reduced to increase heat transfer and reduce wasted heat dissipation. Increasing heat flow in heat exchangers using porous media of different porosity per cent, arrangement, materials, and geometric structure along the flow path. Porous media leads to smaller heat transfer systems that are less expensive and more efficient [1]. As a result, porous medium problems are essential to the design and estimation of heat exchangers. Parameters of the porous media need to be optimized during the design to minimize fluid pressure loss and pumping power [2-6]. Studies on the use of porous media in heat exchangers have been conducted recently.

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Shirvan *et al.,* [7] a porous media-packed double-pipe heat exchanger was studied using a 2D numerical simulation with sensitivity analysis. In porous regions, the Darcy-Brinkman-Forchheimer equations are applied to simulate fluid flow. With increasing Re number, heat exchanger efficiency increased, while decreasing with increasing Da number. In a double-pipe exchanger, Shriven *et al.,* [8] conducted a similar study on turbulent flow. In a laminar flow double-pipe, counter-flow heat exchanger with open-cell porous media in both paths, Chen *et al.,* [9] examined the Heat transfer and pressure loss behaviour are affected by heat exchanger capacity, radiation, and foam properties. Increasing pore density improved heat exchanger performance while applying porous material increased pressure drop. A porous media is partially inserted into a double counterflow heat exchanger, Jamarani *et al.,* [10] examined the effects of turbulent flow. For the purpose of determining the best characteristics of the porous layer, their data was examined for two different porous media configurations using the performance evaluation criterion (PEC). The circular heat exchanger packed with porous media has been studied by Lochan *et al.,* [11] Different mass flow rates were examined to see how the porous material affected heat transfer enhancement. It was determined that a decrease in porosity caused a rise in heat transfer rate, which was confirmed by the CFD program. Bargh *et al.,* [2] examined the effects of porous media on hydrodynamic characteristics, inside a heat exchanger with a differentially configured porous medium, heat transfer and pressure drop are enhanced. The results of this study, a pipe filled with a porous material show a significant enhancement in heat transfer (in both laminar and turbulent flow).

As a heat transfer fluid, nanofluids were utilized in a variety of fields due to their high thermal conductivity [12]. A nanofluid is a suspension of solid nanoparticles (1-100 nm) in a base fluid. [11] Numerous researchers have published studies on Nanofluids, which have a better coefficient of heat transfer than conventional fluids due to their enhanced thermal conductivity and flow behavior [13]. The coefficient of heat transfer and friction factor of nanofluid TiO_2/H_2O in a counter-flow (double tube) heat exchanger were studied by Duangthongsuk and Wong wises [14] at Reynolds number (3000 - 4000), TiO_2 nanoparticles (average size 21 nm) dispersed in water (0.2 to 2 %). The coefficient of heat transfer of nanofluids1% vol. was 26% higher than that of base fluids, while 2.0% vol. nanofluids were 14% lower. With increasing nanofluid concentrations in a given volume, the pressure loss of the water nanofluid is a little larger than the basic working fluid. Farajollahi *et al.,* [15] studied Al_2O_3 (25nm)-water and TiO_2 (10 nm)-water nanofluids. Al_2O_3 had a 2% nanoparticle volume, while TiO2 had a 0.5 % volume. Nanoparticles improve heat transfer, according to their results. Two nanoparticle concentrations are most effective. Qi *et al.,* [12] thermal performance and pressure decrease in heat exchangers Hot water flow (1-5 L/m), TiO_2-H_2O nanoparticle frictions (ω= 0, 0.1, 0.3, and 0.5%), nanofluid positions (shell and tube sides), nanofluid Reynolds number range (3000-12000), and inner tube geometries (smooth and corrugated tubes)were all investigated. Compared to water, nanofluids increase heat transfer, NTU, and effectiveness by 10.8 %, 13.4 %, and 14.8%. For nanofluids with corrugated inner pipe and outer pipe side, the pressure drop increases by 51.9 % and 40.7 %, respectively. Corrugated double-tube heat exchangers perform smooth ones when using nanofluids.

Karimi *et al.,* [16] used MgO MWCNTs/EG nanofluids in a heat exchanger. The result showed that nanofluids with a concentration of 1 % could improve thermal performance by 20%, but also increase pressure drop. In a double-pipe heat exchanger using nanofluid TiO_2/CuO, Singh *et al.,* [13] investigated the relationship between heat transfer and flow rate as well as nanoparticle concentration. The correlations of Gnielinski, Duanghongsuk, Wongwises, and Petukov were utilized to validate the experimental data. At a concentration of 0.3% and a flow rate of 4 LPM, a nanofluid consisting of TiO_2/CuO improves heat transfer by 5%. The results of the experiment verified the

Duangthongsuk and Wongwises correlations. TiO_2/CuO nanofluid improves the thermophysical properties of heat exchangers used in process industries.

Many studies have investigated the effects of Nanofluids and porous media on the increase of heat transfer in a heat exchanger with two counter-flowing pipes [17]. Both of these methods have been shown to increase the heat transfer rate significantly, with porous media advancing the fluid's contact surface and the addition of solid nanoparticles enhancing the fluid's characteristics [18]. Siavashi *et al.,* [19] studied nanofluid forced convection within an annular tube with porous media. Filling the inner tube provides the maximum rate of heat transfer. However, an optimal porous layer thickness for optimizing performance, the ratio of improved heat transfer to increased pressure drop is given. Additionally, Siavashi *et al.,* [20] the optimal parameters of the layers of porous medium for Enhancing the heat transfer and power required for only one tube heat exchanger pump with nanofluids gradients and multilayer porous media were determined. This study investigates the heat transfer of TiO_2–water nanofluid in a copper double-pipe heat exchanger. Also discussed how flow and heat transfer parameters are affected by the presence of porous media at different flow rates and nanoparticle volume percentages.

2. Characteristics of a Porous Medium

2.1 Porosity (ε)

The ratio of interconnected empty spaces to total volume is effective porosity [21]. The porosity of porous materials does not exceed 0.6%. The porosity of similar spheres particle in a closed chamber can be from (0.2595 to 0.4644). Porosity will be reduced if the particles do not fit. Some porous human materials, like metal foams, have porosity close to 1. If the pipes pass through the porous media and the fluid fills them, the porous medium is said to be saturated. Since the fluid only fills a part of the overall pore space, the media is said to be unsaturated [22]. The information on porous media is presented in Table 1.

$$\varepsilon = \frac{V_t - V_p}{V_t} \tag{1}$$

2.2 Permeability (K)

The material's ability to transfer fluid. The porous material's size and the flow specification both have an effect on K. The researchers noted that permeability was affected by characteristics such as porous media structure, porosity coefficients, particle geometry, and surface homogeneity or non-homogeneity. Estimated permeability of the porous material using the Kozeny-Carman relation [23].

$$K = \frac{d^2 \varepsilon^3}{175(1-\varepsilon)^2} \tag{2}$$

Table 1
Characteristics of a Porous Medium

Balls Average diameter (d) mm	Mass (g)	Porosity (ε)	Permeability (K) (m^2)
3	2394	0.3912	8.30714E^{-09}

Table 2
Thermophysical properties of stainless steel (304)

Porous material	Specific heat J/kg K	Thermal conductivity W/m K	Density kg/m^3
Stainless steel 304	500	16.3	8000

3. Nanofluid Physical Properties

At an average temperature, Eq. (2), the thermal-physical properties of TiO_2 and water are listed in Table 2. The thermophysical properties of nanofluid were determined. Density, thermal conductivity, heat capacity, and Effective viscosity, Table 3 display the results of Eqs. (2) through (6).

$$T_{b,h} = \frac{T_{h_i}+T_{h_o}}{2} \;,\; T_{b,c} = \frac{T_{c_i}+T_{c_o}}{2} \tag{3}$$

Choi [24] Correlation predicts the density of nanofluids:

$$\rho_{nf} = \phi\rho_{np} + (1-\phi)\rho_w \tag{4}$$

Several earlier researchers have used Xuan and Roetzel [25] correlation to determine the specific heat of nanofluids.

$$cp_{nf} = \frac{\rho_{np}C_{p,np}(\phi)+C_{p,w}\rho_w(1-\phi)}{\rho_{nf}} \tag{5}$$

Of the several thermal conductivities for nanofluids formulas published in the literature, we focus on the Yu and Choi [26] model, which may predict the improvement in effective thermal conductivity.

$$k_{nf} = \frac{k_{np} + 2k_w + 2(k_{np} - k_w)(1+\beta)^3\phi}{k_{np} + 2k_w - (k_{np} - k_w)(1+\beta)^3\phi} k_w \tag{6}$$

β is the relation between the thickness of the nanolayer and the actual radius of the particle. Typically, $\beta = 0.1$ is applied to compute the thermal conductivity of nanofluids [20]. Drew and Passman's [27] relation establish a common and reliable way to estimate the efficient dynamic viscosity of a nanofluid.

$$\mu_{nf} = (1+2.5\phi)\mu_w \tag{7}$$

Table 3
The thermo-physical properties considered for TiO_2 and base fluid Properties

Properties		TiO_2	water
Density kg/m^3	ρ	3900	994.7
Specific heat J/kg	cp	697	0.615
Thermal conductivity W/m.K	k	11.8	4170.7
Viscosity kg/m.s	μ	-	0.000547

Table 4
Thermophysical properties of TiO_2 water-based Nanofluids

ϕ	ρ_{nf}	cp_{nf}	k_{nf}	μ_{nf}
0.001	997.94	4177.405	0.617	0.000548
0.002	1001.171	4183.962	0.619	0.000549
0.003	1010.6453	4103.643	0.62222	0.00055

4. Preparation of Nanofluid

There are three well-established ways to maintain the suspension and prevent nanoparticle deposition. Surfactants, adjusting nanofluid pH and ultrasonic vibration because of the very small amounts of surfactant utilized in the experiment, it did not affect the thermophysical properties of the nanofluids; therefore, an ultrasonic vibrator with a magnetic stirrer with a diameter of 15 nm of TiO_2 was employed in the current work. For about four hours after pH measurement, the nanofluid flowed at its highest flow rate. Even at the fluid's low flow rates, no deposition was seen in the experiments. The ultrasonic vibration process must be mechanically repeated. After 24 hours, there was no deposition in any of the nanofluid samples.

5. Experimental Setup

Figure 1 shows both the schematic and experimental setup of the counterflow double pipe heat exchanger. The experimental setup includes of a test section, a tank of hot water, and a tank for nanofluid collection. Two pumps were used: a hot water pump and a pump to re-circulate nanofluid, valves to control the counter flow, and two flow meters. The test section is a 1.04-meter-long horizontal double-tube heat exchanger with 3mm stainless steel balls packed in the inner pipe, nanofluid flowing inside the inner pipe (core), and hot water flowing in the outside pipe (annular).

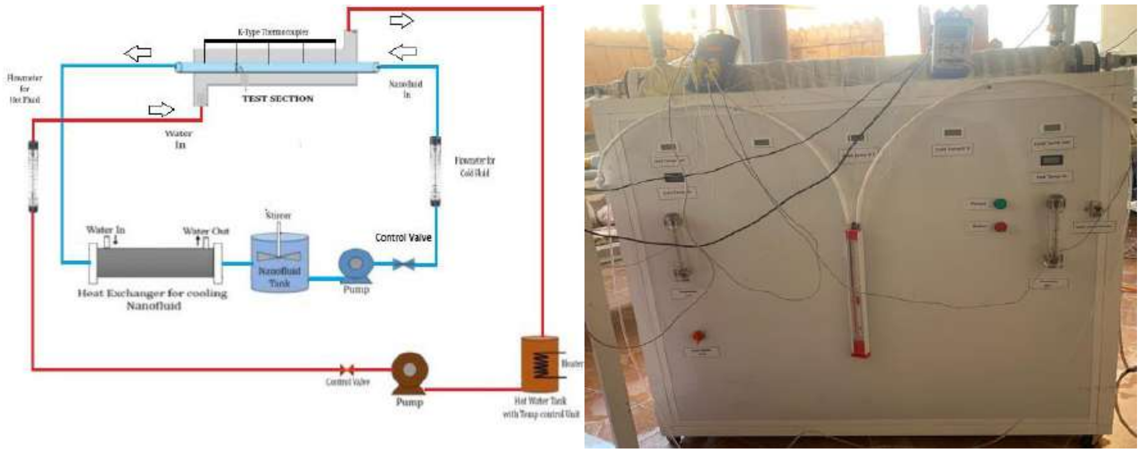

Fig. 1. Shows a schematic design and an experimental rig for a counter flow double pipe heat exchanger

The inner copper pipe has a diameter of 26.02 mm and a thickness of 1.74 mm; while the outer galvanizes steel shell has a diameter of 52 mm. Fibreglass (50 mm thick) was used to insulate the outer pipe's surface to reduce heat loss to the surrounding. In the outer pipe, the fluid flow at 6 LPM. And the inner pipe's fluid flow ranged from 4 to 6 LPM. To maintain a constant temperature at the test section's inlet, heated fluid was re-circulated. temperature is measured using a K-type thermocouple. The inlet and outlet temperatures of hot water and working liquid are measured with thermocouples of K- type put directly into the flowing current. Thermocouples are placed longitudinally on the inner copper pipe wall surface, with five thermocouples uniformly spaced around the perimeter. The nanofluid and hot water is leaving the test section in 40L galvanized steel receiver tanks. In a tank with a cooling capacity of 3.5 kW and a thermostat using the nanofluid temperature is maintained. Similarly, the cold-water tank, a 2.8 kW electric heater with a temperature control is applied to remain the hot water at an inlet temperature. The flow rate of working liquid and hot fluid is controlled and measured by valves and a flow meter. All T-type thermocouples are calibrated with excellent precision of 0.1C. The inlet and outlet temperatures of the hot water and working liquid, the inner wall temperatures of the inner pipe, the volume flow rate of the hot water and working liquid, and the porous medium with or without nanofluid will be measured during the test section.

6. Experimental Data Processing

The average heat exchange capacity from hot fluid (water) to cold fluid (cold water with or without nanofluid) flowing through porous is defined as:

$$Q_{ave} = \frac{Q_h + Q_c}{2} \tag{8}$$

Where Q_h, Q_c calculated by applying the energy conservation equation, which is expressed in Eqs. (9) and (10):

$$Q_h = \dot{m}_h c_{ph} \left(T_{h,i} - T_{h,o}\right) \tag{9}$$

$$Q_c = \dot{m}_c c_{pc} \left(T_{c,o} - T_{c,i}\right) \tag{10}$$

Calculation of the mass flow rate of hot and cold water:

$$\dot{m}_h = \frac{\dot{V}}{60000} \times \rho_h \,, \quad \dot{m}_c = \frac{\dot{V}}{60000} \times \rho_c \tag{11}$$

The convective heat transfer coefficient (h_c) can be determined using the internal surface temperatures of the inner tube:

$$h_c = \frac{Q_c}{A_i \times \left(T_s - T_{avg,h}\right)} \tag{12}$$

Where A_i is the inner tube's internal surface area, Ts is the local surface temperature at the inner tube's wall, and T avg, c is the cold fluid's average temperature (water):

$$A_i = \pi \times d_i \times L$$

$$T_s = \frac{T1 + T2 + T3 + T4 + T5}{5}$$

$$T_{avg,c} = \frac{Tc_i + Tc_o}{2}$$

The following formula determines the Nusselt number for cold water:

$$\mathrm{Nu_c} = \frac{\mathrm{h_i\ di}}{\mathrm{k_{eff}}} \qquad \text{For case porous media.} \tag{13}$$

$$\mathrm{Nu_c} = \frac{\mathrm{h_i\ di}}{\mathrm{k_{eff,nf}}} \qquad \text{For case porous media with nanofluid.} \tag{14}$$

$$\mathrm{k_{eff}} = \mathrm{k_f^{\epsilon}\, k_s^{(1-\epsilon)}} \ \text{ Or } \ \mathrm{k_{eff}} = (1-\varepsilon)\mathrm{k_f} + \varepsilon \mathrm{k_s} \tag{15}$$

$$\mathrm{k_{eff,nf}} = (1-\varepsilon)\mathrm{k_{eff}} + \varepsilon \mathrm{k_{nf}} \tag{16}$$

Where $\mathrm{k_f}$ is the thermal conductivity of the fluid (water), k_s is thermal conductivity of the solid (stainless steel balls (AISI 304)), $\mathrm{k_{nf}}$ is thermal conductivity of the nanofluid (TiO_2-water), $\mathrm{k_{eff,nf}}$ is thermal conductivity of solid, and nanofluid, ε is porosity [28].

$$\mathrm{U_{i,exp}} = \frac{\mathrm{Q_{avg,}}}{\mathrm{A}_i \times \mathrm{F} \times \mathrm{LMTD}} \tag{17}$$

The logarithmic mean temperature difference LMTD is evaluated from:

$$\mathrm{LMTD} = \frac{\Delta \mathrm{T_{max}} - \Delta \mathrm{T_{min}}}{\ln \frac{\Delta \mathrm{T_{max}}}{\Delta \mathrm{T_{min}}}} \tag{18}$$

For case of counter flow

$\Delta \mathrm{T_{max}} = \mathrm{T_{h,i}} - \mathrm{T_{c,o}}$ and $\Delta \mathrm{T_{min}} = \mathrm{T_{h,o}} - \mathrm{T_{c,i}}$, assume F=1 for counter flow

The effectiveness and the NTU are given by:

$$\mathrm{E} = \frac{\mathrm{q}}{\mathrm{q_{max}}} \tag{19}$$

Where q actual energy transferred and $\mathrm{q_{max}}$ maximum energy transferred [29]

$$\mathrm{NTU} = \frac{\mathrm{UA}}{\mathrm{C_{min}}} \tag{20}$$

Flow Rate Heat Capacity

$$\mathrm{C} = \dot{\mathrm{m}} \times \mathrm{C_p} \tag{21}$$

$$\mathrm{C_{min}} = \begin{cases} \mathrm{C_h} \text{ if } \mathrm{C_h} < \mathrm{C_c} \\ \mathrm{C_c} \text{ if } \mathrm{C_h} < \mathrm{C_h} \end{cases}$$

Figure 2 represents the validation of experimental results at 0.3 volume% TiO_2 Nanofluid with Singh *et al.*, [13] and correlations by Gnielinski, Duanghongsuk Wongwises, and Petukov. The correlation between the Gnelinski model and the experimental data was the best one.

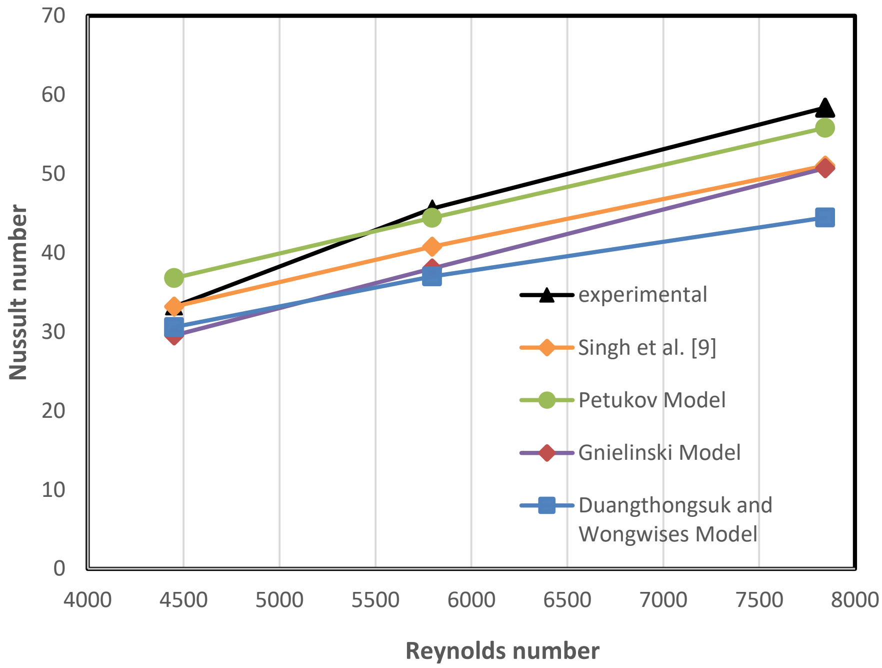

Fig. 2. Validation of result obtained using experimental and theoretical correlation (0.3% TiO_2)

7. Results and Discussion

The addition of nanoparticles to the cold water circulating inside the inner pipe filled with stainless-steel balls by varying the concentration of nanoparticles from 0.1% to 0.3% by volume and varying flow rates from (4-8 LPM) are studied experimentally. Figure (3) shows that the temperature difference decreases as the flow rate increases for all cases.

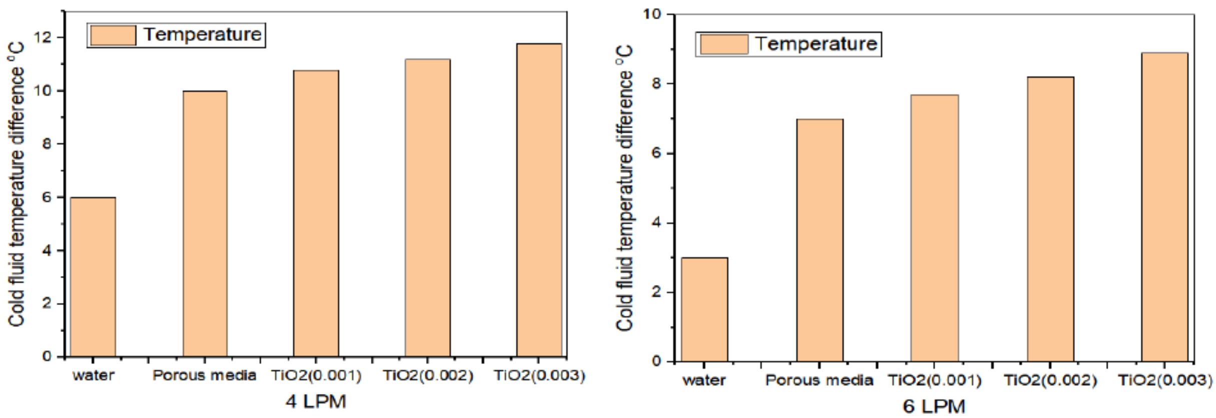

Fig. 3. Temperature difference along test section for all cases (a) 4LPM and (b) 6LPM

Figure 4 illustrates the Nusselt number as a function of flow rate for using porous media with various nanoparticle concentrations within inner pipe. Changing nanoparticle concentration, increase the flow rate. As shown in this figure, the Nusselt Number increased as a result. The presence of porous media and nanoparticles enhanced the mixture's thermo - physical properties. In addition, the heat transfer from the hot fluid to the cold fluid is significantly increased due to the porous media's enhanced conduction heat transfer.

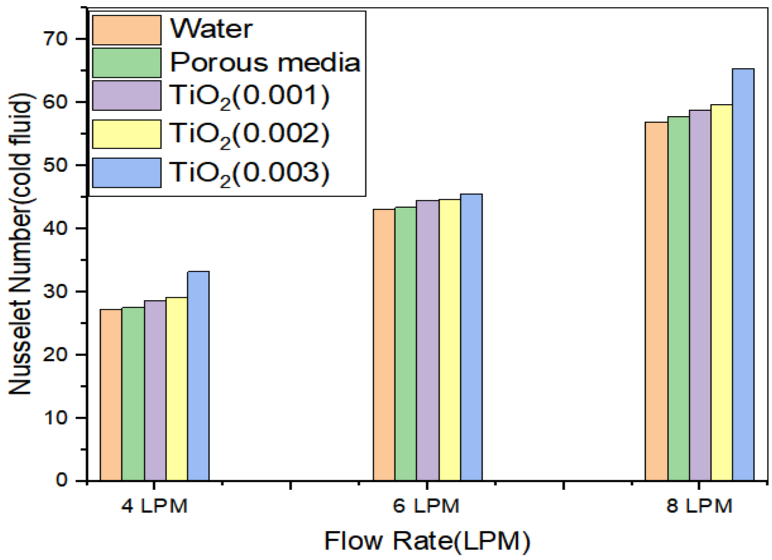

Fig. 4. Nusselt Number vs flow rates for all cases

Figure 5 shows the enhancement of the overall heat transfer coefficient because of an increase in flow rate and the temperature difference between the outer and inner pipes of the same flow.

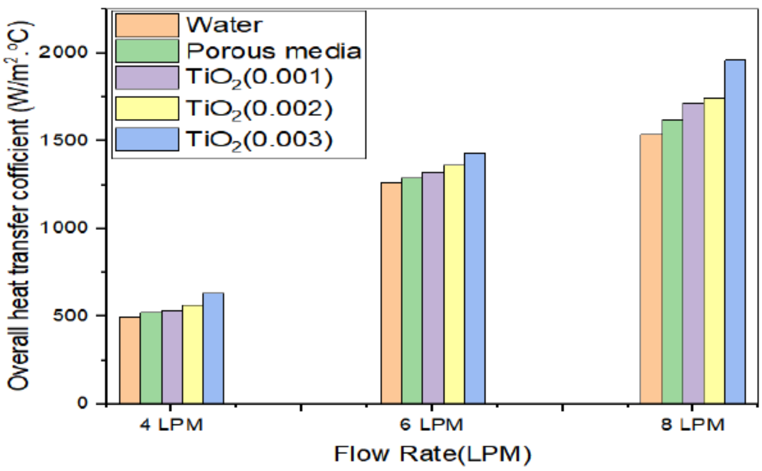

Fig. 5. Overall heat transfer coefficient vs flow rates for all cases

In Figure 6, as flow rates increase, the heat transfer rates increase as well.

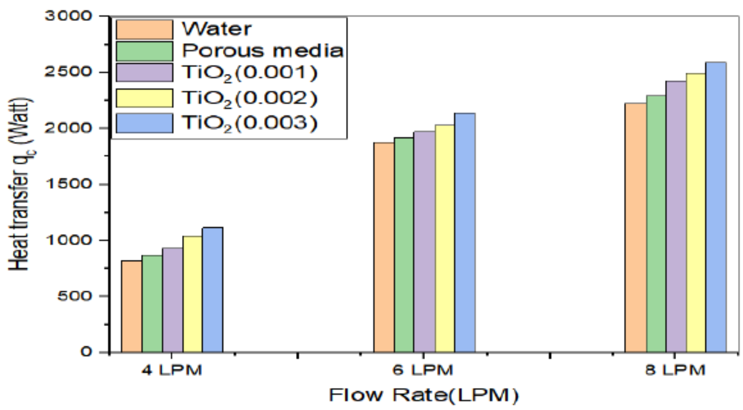

Fig. 6. Heat transfer rates vs flow rates for all cases

Figure 7 represents the effectiveness of each case. At lower flow rates (Reynold number) from, it is possible to achieve higher effectiveness. However, the effectiveness is affected by the presence of porous media and nanoparticle concentration. Applying porous media and nanofluids to the inner pipe could increase effectiveness. The comparison is to pure water.

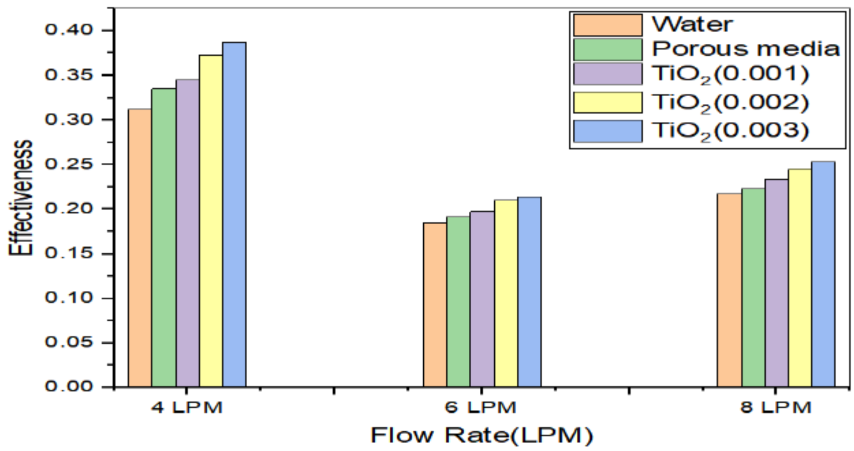

Fig. 7. Effectiveness vs flow rates for all cases

In Figure 8 NTU shows similar behaviour to effectiveness. the highest NTU values are achieved by filling the inner pipe with porous media when using nanofluids. The increase in NTU values and effectiveness is dependent on the overall heat transfer coefficient, which increases as the nanofluid volume fraction increases [30].

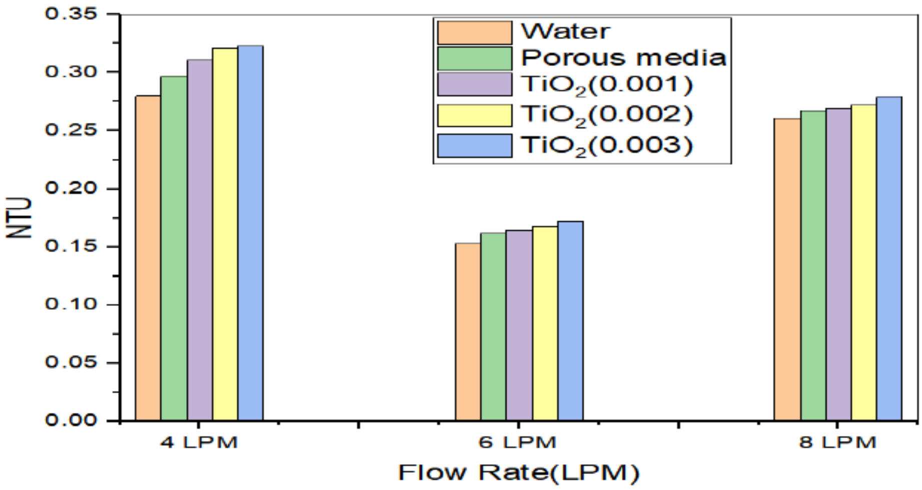

Fig. 8. NTU vs flow rates for all cases

8. Conclusions

For this experiment, double-pipe heat exchanger with counter current flow and a porous media consisting of stainless-steel balls 3 mm in diameter, with three different nanofluid volume fractions flowing through the inner pipe. Hot water at a constant inlet temperature flow through the heat exchanger's outer pipe. It can be concluded from the experimental tests:

i. In all cases, the temperature difference decreases as the flow rate increases.
ii. Both NTU and effectiveness improve with a decrease in flow rate and increase nanofluid fraction.
iii. In general, the heat transfer coefficient improves with both higher flow rates and a higher nanofluid fraction.
iv. Using porous media and nanofluid increases the surface temperature of the inner pipe compared to an empty pipe.
v. The heat transfer rate can be increased by 13.2%, 26.8%, and 35.4% for using porous media and TiO_2 nanofluids with φ = 0.1%, 0.2%, and 0.3% comparison to pure water.

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