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Experiment on Heat Transfer Enhancement for a Double Pipe Heat Exchanger with Air Injection of Perforated Turbulator

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Abstract - In this study, an attempt is made to evaluate the performance of a double pipe heat exchanger (DPHE) through an experiment. The outside surface of the inner pipe of the DPHE is surrounded with perforated helical tube, where the perforations allow air to enter the annulus. The air injection and the turbulator itself are expected to increase the number of vortices in the boundary layer of the inner pipe (annulus side) and expand the heat transfer regions. Due to the bouncy effect, the small air bubbles within the cold water promote flow eddies and hence increase the thermal performance. The Reynolds number based on the hydraulic diameter of the annulus is ranged from 5,000 to 17,000. The findings demonstrate that for a flow affected by an air injector, performance varies with Reynolds number. Both the Nusselt number and efficiency increase percentages are approximately 27 for each. However, the friction factor raises to 43% compared to the plain case, which only increases the performance evaluation criteria (PEC) by 8%.

***Keywords*:** Double-pipe heat exchanger (DPHE), Perforated turbulator, Air injection, Two-phase flow, Thermal-hydraulic performance.

1. Introduction

For efficient energy utilization, conversion, and recovery, heat exchangers in general are the main components of many machines, engines, and plants. In other words, many industries that include thermal exchange make use of heat exchangers, such as steam power plants, techniques for heating and/or cooling products in the chemical, agricultural, and many other fields. In specific, double pipe heat exchangers, or DPHEs, are typically utilized for two liquids with high heat capacities and sensible heat requirements. This particular type of assembled device has an inner pipe that contains hot liquid and a shell portion, or annulus, that contains cold liquid. Moreover, the annulus is frequently fitted with obstacles to slow the flow and produce swirling motions. Many studies, including the ones cited in [1], have demonstrated that heat exchange augmentation by modifying the geometry increases the thermal efficiency of such devices while also lowering their operating costs. This approach, which is by far the most popular one among all other enhancement techniques, is known as a passive method. However, fluid-stream perturbations create frictional resistance, which results in a pressure drop that must be controlled to ensure optimal heat-exchanger performance [2]. The role of a square-cut twisted tape in DPHE was investigated by the researchers [3]. The result revealed a considerable improvement in heat transfer rate, friction factor, and thermal performance factor as compared to a device using conventionally twisted tape. This beneficial factor was caused by the wide heat transfer region within the boundary layer thickness close to the wall. The authors [4] conducted an experimental study to evaluate the effect of helical tapes on heat transfer enhancement with and without a rod placed in the tube. A wide range of Reynolds numbers were examined in order to determine the optimal tube flow rate. They observed that the pressure drop was greater for the helical twisted tape case with rod, but the heat exchanger had the highest heat transfer rate there. In a shell and coiled tube heat exchanger, air bubbles were introduced by [5]. In order to spread the bubbles throughout the heat exchanger, they developed the concept of employing spiral tubes. The improved thermal performance of their experiments indicated the advantage of pumping air through the annulus.

The study of [6] involved an experimental research into the impact of air bubble injection on a horizontal heat exchanger's efficiency. It was evaluated how the injection of air bubbles at various air flow rates affected the number of thermal units (NTU), energy loss, and efficiency. The heat exchanger's efficiency and NTU greatly increased as the air bubbles were injected. It is therefore thought that the motion of air bubbles creates disruptions, which ultimately causes the amount of turbulence in the shell side flow to increase, raising the value of NTU and energy loss. Within the experimental setup of [7], a shell and multi-tube heat exchanger with parallel (upper and lower nozzles) and transverse (side nozzles) air injection methods were fitted. Cross injection had a bigger effect on the heat exchanger efficiency, number of transfer units (NTU), and overall heat transfer coefficient (U) in all test conditions than the first method (parallel injection). Cross injection produced a greater shell side pressure loss than parallel injection. Air was blown into the annulus side of the heat exchanger by a variety of injector types, according to [8]. Several annulus side and inner tube flow rates were studied experimentally. It was decided to conduct an energetic analysis to see how the air flow rate and heat exchanger positioning angle affect the performance as a whole. According to the information gathered, adding air bubbles could increase U by 10.3 to 149.5%. In the study, [9] supplied air to the coil side of a horizontal shell and coiled tube heat exchanger. The results demonstrated that adding air bubbles to the heat exchanger improved its thermal efficiency. Hot water and air were combined outside the heat exchanger in a T-junction of [10]'s device before going inside the inner tube of the heat exchanger. Hot water was kept flowing at a constant rate of 2 (lit/min). For the hot water side, 3, 4, 5, and 6 (lit/min) flow rates were considered. According to the findings, there was a 33% rise in the heat transfer coefficient and a 38% increase in the number of transfer units. The experimental and numerical findings for DPHE including helical turbulator and air-bubble injection as passive and active enhancement approaches were obtained by [11, 12]. The maximum levels of energy transmission, pressure drop, and efficiency were achieved by the turbulator configurations alone. Copper pipes with the following dimensions were used to construct the heat exchangers: d = 10, e = 2.4, d c = 13, D = 18 (mm), and wall thickness of 1 mm. The heat exchanger's total length was constant at 530 (mm). Also, during the studies, the shell-side Reynolds number was altered from roughly 1,000 to 4,000 (depending on the water inlet flow rate and shell-side hydraulic diameter).

Researchers [13] investigated the role of air bubble injection on the thermal performance of a counter-flow heat exchanger with a vertical, helical coil tube. Air bubbles with hot water improved effectiveness, and at a flow rate of 3.5 (lit/min), the maximum effectiveness value was 0.22. The friction factor increased dramatically as additional air bubbles were compressed at a rate of 0.1 to 0.31 (lit/min) at Reynolds number of 9823. With an air flow rate of 3.5 (lit/min) and Reynolds numbers ranging from 9823 to 48028, the maximum enhancement ratio of the Nusselt number was between 2.4 and 3.1. The maximum thermal performance factor was 1.33 with an air bubble injection rate of 3.5 (lit/min); however the energy loss increased from 74 to 88%. The combined effects of air injection and twisted tape turbulators on improving heat exchange were tested in the study of [14]. According to that study, adding air to the system can raise the pressure drop and heat transfer coefficient by a maximum of 62 and 30.3%, respectively. The cost-benefit ratio was determined to be acceptable for air and water flow rates of 1 and 5 (lit/min), with the greatest value coming in at around 0.93. The effects of rotating turbulators on forced convection, entropy, and heat transfer were studied by several researchers, including [15]. This supports the idea that the rotating turbulator is more energetic because the entropy generation was greatly decreased while the thermal performance was noticeably enhanced. Therefore, according to the previous papers, the majority of prior research has focused on the effect of an equipment inserted inside the inner pipe. In this work, however, an experiment was performed to test the feasibility of introducing air bubbles into the annulus to improve heat transfer in a horizontal, double-pipe heat exchanger with perforated, circular turbulator. The study analyzes the outcomes of two layouts: one in which an air injection was performed via holes in a turbulator and the other was a plain DPHE. Air bubbles were blown by amounts such that their mass flow rates are very small compared to those of water flow. It was found that the annulus with perforated turbulator has a high heat transfer coefficient and pressure drop higher than those of the conventional heat exchanger. After the review of the equations and instruments utilized in the next sections, the key findings of the thermal-hydraulic performance will be presented.

2. Mathematical Equations

The test section is a double pipe heat exchanger encompassing counterflow configurations. The working fluids (waters) have constant physical properties, and the flow is considered incompressible. Most correlations and formulae will be applied to the cold water within the shell side (annulus). The equations are all taken from textbooks like [16] and are presented below as needed. The hydraulic diameter equals to $D_{\mathrm{h}} = D_{\mathrm{o}} - D_{\mathrm{i},}$ where D_o is the inner diameter of outer pipe while D_i is the outer diameter of inner pipe. Reynolds number based on D_h and the mass flow rate ($\dot{m}_{\mathrm{h}}$) is $Re = (\rho\, u\, D_h)/\mu$, where u, μ, and ρ are annular fluid velocity, dynamic viscosity, and density, respectively. The experiments were performed for different mass flow rates, and $u = \dot{m}_{\mathrm{h}}/\rho A_{\mathrm{c}}$ was calculated for each test, where A_{c} is the cross section area of annulus. The dimensionless quantity representing the ratio of pressure drops (due to viscous and obstacles losses) to kinetic energy is the friction factor determined from

$$f = \frac{2\,\Delta P . D_{\mathrm{h}}}{\rho . u^2 . L}, \tag{1}$$

where ΔP is the pressure drop evaluated using a manometer between the two ends of the shell. The heat transfer rate $\dot{Q}_{\mathrm{C}}$ transferred to the cold water equals that released by the hot water, i.e. $\dot{Q} = \dot{Q}_{\mathrm{h}} = \dot{Q}_{\mathrm{c}}$, which is

$$\dot{Q} = \dot{m}_{\mathrm{h}} c_{\mathrm{p}} \left(T_{\mathrm{h,i}} - T_{\mathrm{h,o}}\right). \tag{2}$$

The Nusselt number was estimated by averaging the heat transfer coefficient of the cold water as follows:

$$Nu = \frac{\bar{h}\, D_h}{K}, \tag{3}$$

where $\bar{h}$ and K refer to the averaged heat transfer coefficient and thermal conductivity of cold fluid. For the current analysis, the average heat transfer coefficient ($\bar{h}$) was found via

$$\bar{h} \cong \frac{\dot{Q}}{A_{\mathrm{s}}(\bar{T}_{\mathrm{s}} - \bar{T}_{\mathrm{b}})}, \tag{4}$$

where the subscripts s and b denote for surface and bulk temperatures averaged for five probes mounted on the inner pipe surface and within the cold fluid. The total heat transfer surface area (A_s) includes the new surface extension due to the attached tube. The overall heat transfer coefficient can be calculated from

$$U = \frac{\dot{Q}/A_s}{\left[\dfrac{\left(T_{\mathrm{h,in}} - T_{\mathrm{c,out}}\right) - \left(T_{\mathrm{h,out}} - T_{\mathrm{c,in}}\right)}{ln \dfrac{\left(T_{\mathrm{h,in}} - T_{\mathrm{c,out}}\right)}{\left(T_{\mathrm{h,out}} - T_{\mathrm{c,in}}\right)}}\right]}. \tag{5}$$

The effectiveness (ε) is the ratio of actual ($\dot{Q}_{\mathrm{act}}$) to maximum ($\dot{Q}_{\mathrm{max}}$) heat transfer rates. The maximum heat was achieved at the lowest specific heat capacity and maximum temperature difference as follows:

$$\varepsilon = \frac{\dot{Q}_{\mathrm{act}}}{\dot{Q}_{\mathrm{max}}}, \tag{6}$$

where $\dot{Q}_{\max} = C_{\min}.\Delta T_{\max}$, $C_{\min} = \mathrm{f}(c_{\mathrm{c}}, c_{\mathrm{h}})$ and $\Delta T_{\max} = T_{\mathrm{h,i}} - T_{\mathrm{c,i}}$. The number of transfer units is

$$NTU = \frac{UA_s}{C_{\min}}. \tag{7}$$

For a counter flow where, $C = C_{min}/C_{max}$, the effectiveness can be sought from

$$\varepsilon = \frac{1 - exp[-NTU.(1 - C)]}{1 - C.exp[-NTU.(1 - C)]}. \tag{8}$$

Another factor called performance evaluation criterion, or (PEC), can be used to evidently show how this new modification of adding perforated turbulator is useful. The first and third authors used this factor in [17]'s study for other kind of heat transfer improvement as follows:

$$\mathrm{PEC} = \frac{Nu_{\mathrm{t}}/Nu_{\mathrm{p}}}{(f_{\mathrm{t}}/f_{\mathrm{p}})^{1/3}}, \tag{9}$$

where the symbols t and p refer to turbulator and plain (conventional DPHE) names respectively.

3. Experimental Setup

The perforated helical tube was fitted entirely around the outer surface of inner pipe inside the heat exchanger's core. The experimental parameters of the current work are listed in Table 1. Two test arrangements were used: one used a standard heat exchanger without turbulator, and the other was a modified heat exchanger that injects air by a turbulator. In the present study, air was pumped through the holes of tabulator into the annulus. There are 30 of these perforations evenly spaced across the helical tube. The heat exchanger was completely insulated from the outside environment, and the turbulator was fabricated of copper. The system for distributing cold and hot water was constructed of tanks, pipes, and pumps. In the test section, the supplied water was pumped at the ambient temperature, and it was heated with an electrical water heater (3kW Geyser**)** until it reached the appropriate temperature.

Table1: Double pipes and perforated, helical turbulator characteristics.

Parameter	**Inner pipe**	**Outer pipe**
Inner pipe diameter (mm)	19	52
Outer pipe diameter (mm)	22	63
Length (m)	2	2
Turbulator diameter (mm)	5	6
Material type	Copper (401 W/m.°C)	Poly vinyl Chloride (0.1 W/m.°C)
Mass flow rate (kg/s)	0.15	0.1 to 0.3
Inlet temperature (°C)	71	29
Turbulator holes diameter (mm)	1	
Turbulator perforation pitch (mm)	66.6	
Helical turbulator pitch (mm)	100	

The measurement tools included type K thermocouples, a 12-channel temperature recorder data logger (Model BTM4208SD), a pressure drop device (TAAM - 220 V - 50 Hz - 0.37 kW), and water flow meters (2 – 18 L/min). The inner pipe of the double pipe heat exchanger has five thermocouples fitted to the wall surface. The cold fluid was controlled by five additional thermocouples as sketched in Fig. 1. A pair of thermocouples were connected to the inlet and outlet of the hot

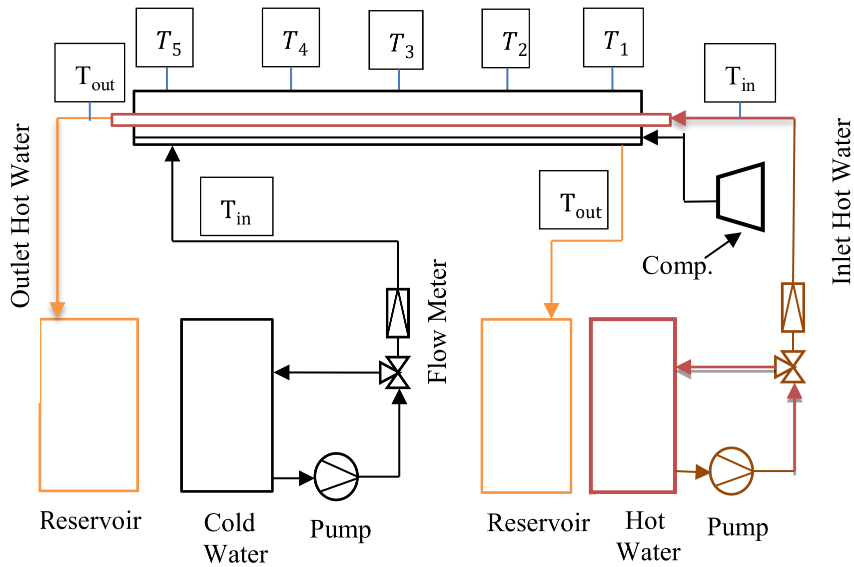

Fig. 1: Schematic diagram outlining the main experiment components of this work

and cold water pipes, respectively. The studies were conducted at Reynolds numbers between 5,000 and 17,000. All data were monitored by linking a computer to the test system. Uncertainty analyses were accomplished before starting any experiment. The following table illustrates the amounts of these uncertainties in the measurements:

Table 2 Experimental data uncertainties encountered in this work.

Measurement type	**Uncertainty Range**
Flow rate	±0.033 L/min.
Temperature	95%
Pressure drop	99%

Figure 2 displays the built apparatus that took place in one of the author's university facilities. All connections and reservoirs were thermally insulated. The measurements was begun by setting the flow rate of hot water at a constant value while modifying the cold water flow rates. In the annulus, the helical turbulator depicted in Fig. 3 was placed after being wrapped around the inner pipe wall. With the present distances, it has 20 turns, and each turn contains one and a half holes around it.

Fig. 2: The experimental facility utilized in the present study.

Fig. 3: The helical turbulator around the inner pipe.

4. Results and discussions

The tests were carried out for two arrangements, and results were divided into two groups accordingly: with and without turbulator. Figure 4 shows the impact of the perforated turbulator on the Nusselt number (Nu) in relation to the annulus' flow velocities. At all Reynolds numbers (Re), the amount of air that was injected into the specified region remained constant. The Nusselt number grows with Re as expected; however there is an expanding difference between the behaviour of the two lines shown in the figure. This provides a strong evidence that air bubble turbulators are advantageous, especially under high flow rates. The augment in thermal performance is expected to be even stronger at higher Re. This is because as the Reynolds number is elevated, the intensity of the turbulence will be increased, leading to a larger rate of convective heat transfer. The

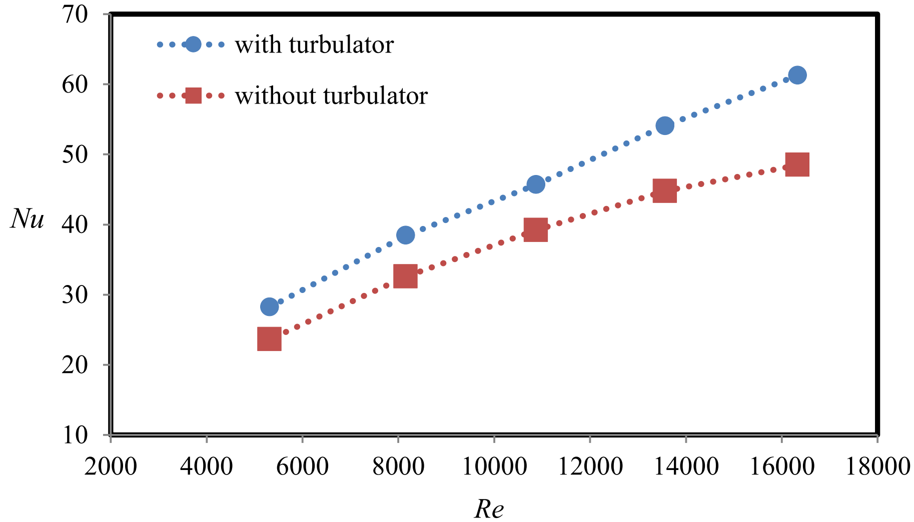

Fig. 4: The Nusselt number variations with Reynolds number with and without turbulator.

air bubble method improved the Nusselt number by around 16% to 21% depending on the Re and the volume of air injected. The turbulence of the fluid flow was increased to reach the largest Nusselt number augmentation at Re of 16,300 as a result of turbulator disturbances and air bubbles entering the annular side of the flow. The friction factor related to the pressure drop is the second key factor in this analysis. Using Eq. (1) with Re, the friction factors in Fig. 5 were determined. Both turbulated and plain double pipe heat exchangers exhibit roughly similar decays in terms of flow friction.

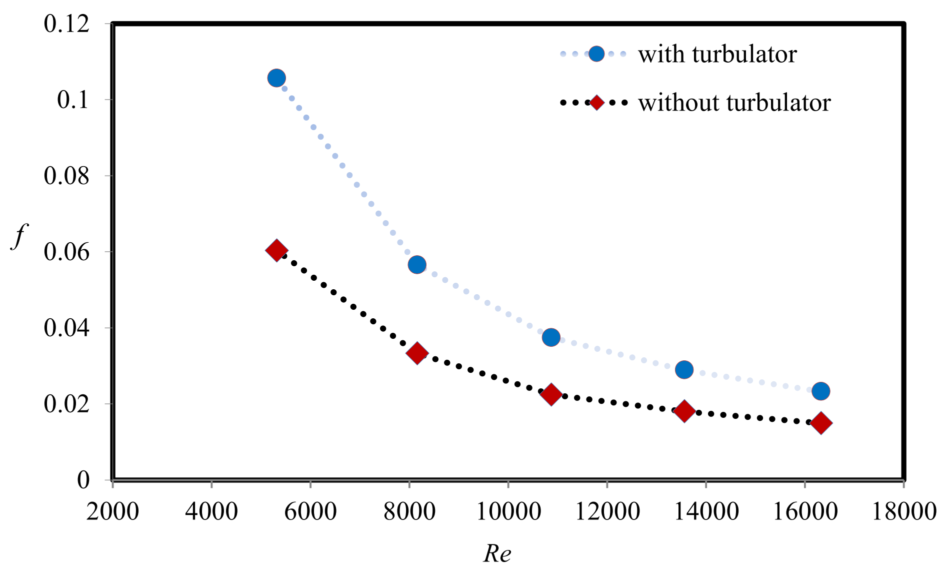

Fig. 5: The friction factor variations with Reynolds number with and without turbulator.

High viscous forces are prevailing in the flow at low velocity. As the momentum energy begins to dominate, this wall effect starts to diminish. As can be seen, the variances in the friction factors between the two geometries grow as the Reynolds number raises. This is because the flow experiences reduced resistance as the thickness of the boundary layer decreases close to the walls. That means the action of turbulator becomes less on the flow shear stress as the flow velocity increases. Figure 6 provides another proof of the advantage of adding a turbulator to improve the thermal performance of DPHE. If the turbulator is utilized at the low flow velocity exhibited in the figure, as determined by comparing the values of the two geometries, the effectiveness will climb to roughly 14.5%. When the annular flow accelerates, this benefit grows to 27%. As the cold flow velocity increases, it can be predicted that the effectiveness of modified DPHE will be boosted noticeably. In relation to the effectiveness of heat exchanger manufactured with perforated helical pipe and air injection, Fig. 7 exhibits the Nusselt number ratio (Nu_t/Nu_p), friction factor ratio (f_t/f_p), and Performance evaluation criterion (PEC). It is evident that when the mass flow rate increases, the friction factor ratio tends to decrease to be between 1.75 and 1.55.

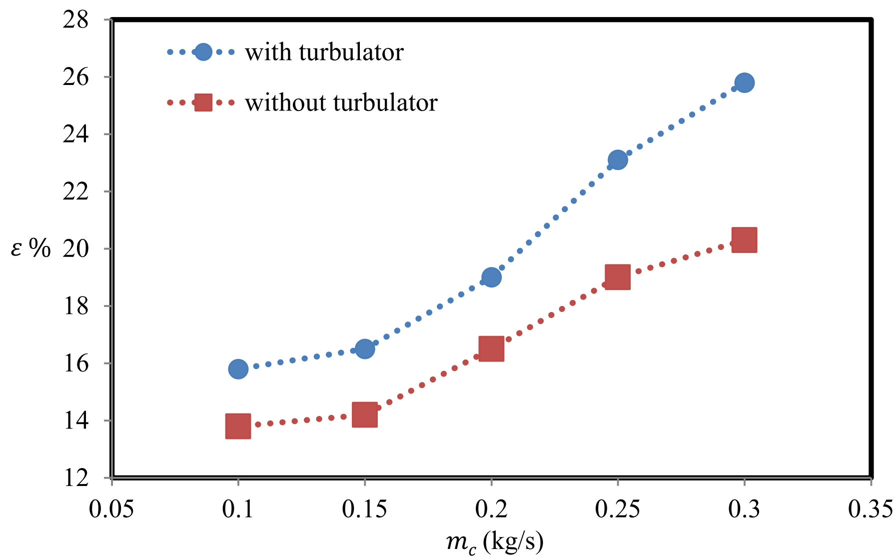

Fig. 6: The effectiveness of heat exchanger coupled with perforated turbulator.

This is because when water velocity climbs, the flow steadily stirs from laminar to transitional and turbulent regimes with little viscous effect, demonstrating that the unit performs best in high-flow conditions. It is well known that in the turbulent regime compared to the laminar one, the friction factor is usually smaller. But still, in this instance, more power is required to pump the flow inside the annulus and increase the flow velocity. The Nusselt number changes from 1.19 to 1.26 owing to the turbulator as the flow moves more quickly through the annulus. Air is injected through the device to achieve also the desired thermal operation. The PCE profile shows the most significant improvement in the heat exchanger as a whole provided by the turbulator. As comparison to the ordinary case, PCE increases by around 8% when the turbulator and air injection are added to the flow.

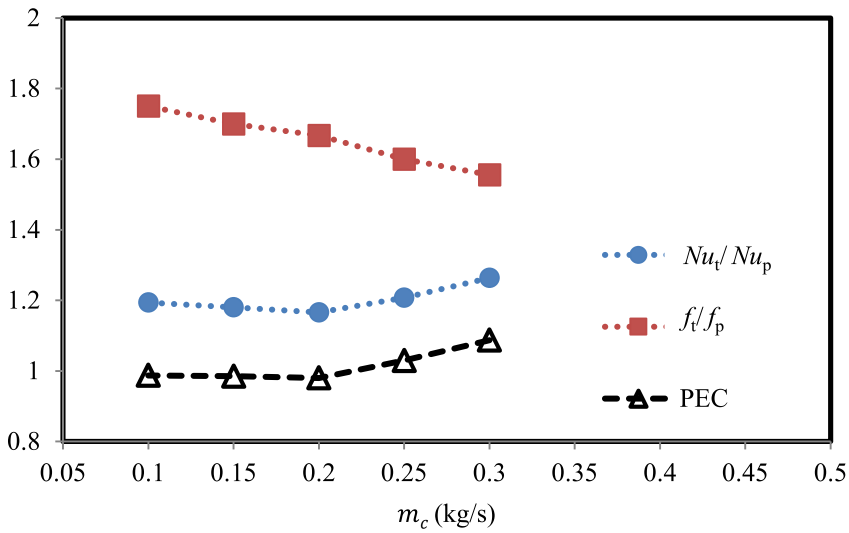

Fig. 7: Parameters of the thermal-hydraulic performance using perforted turbulator with air injection.

4. Conclusions

The strategy of improving heat transfer by turbulator perturbations and air injection babbles is hoped to be the main aspect of the double pipe heat exchanger in future. The experimental results of this research show that these enhancements have a significant effect on the Nusselt number, thermal efficiency, and other parameters. The influence is shown to grow with the Reynolds number (Re). An overview of the current findings is provided below:

- The highest increase in Nusselt number depends on the liquid flow rates, where it is shown to increase from 19% to 26% for Re changing between 5,000 and 17,000.
- The biggest boost in thermal overall effectiveness depends on the liquid flow rates, where it has been demonstrated to grow from 19% to 26% for Re varying between 5,000 and 17,000.
- When the turbulator is used and the Re ranges from 5,000 to 17,000, the percentage increase in friction factor likewise jumps from 36% to 43%.
- The increased heat transfer effect brought on by the enhancing approach is more substantial than the effects of the friction factor ratio, as seen in the PEC profile having more than one at the highest, presented Re.

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