

EXPERIMENTAL IMPLEMENTATION OF THERMAL ENHANCEMENT PERFORMANCE OF AIR HEAT EXCHANGER'S PIPES UTILIZING UNCONVENTIONAL TURBULATOR

Ali Abdulwahab Ismaeel

*Department of Electromechanical Engineering¹
50106@uotechnology.edu.iq*

Nassr Fadhil Hussein

*Department of Electromechanical Engineering¹
50153@uotechnology.edu.iq*

Kadhim H. Suffer

*Department of Mechanical Engineering
Al-Nahrain University
Al-Judrayei neighborhood, Baghdad, Iraq, 64085
kadhim_askar@yahoo.com*

Zuradzman M Razlan

*Faculty of Mechanical Engineering Technology
Universiti Malaysia Perlis
Pauh Putra Campus, Arau, Perlis, Malaysia, 02600
zuradzman@unimap.edu.my*

*¹University of Technology – Iraq
Al-Wehda neighborhood, Baghdad, Iraq, 19006*

Abstract

Heat exchangers are widely used in industry, however, raising their performance are important for the variety of applications. Consequently, efficiency improvement associated with low production cost is considered in this experimental work. The current study aims to enhance the rate of heat transfer in pipe-type heat exchangers experimentally by using a novel nozzle as a turbulator. The cross-sectional shape of the nozzle is hexagonal, and the diameter ratio DR is equal to 0.5. Constant heat flux was maintained in the vicinity of the section of the test tube, while the working fluid was pumped into the open system at six discrete Reynolds number values ranging from 6000 to 19500. To investigate the effect of distance among the pieces, three turbulators with different numbers were assigned and named as ($N=4, 5$ and 6). The results indicated an increase of 172 %, 194 % and 216 % of the heat transfer rate for cases 4, 5 and 6 respectively comparing to the benchmark tube. On the other hand, the friction factor values increased remarkably due to the inserting of turbulators by about of 722.9 % for $N=4$, 823.9 % for $N=5$ and 886.7 % for $N=6$ compared to a plain tube case. Moreover, it has been established that with the insertion of 6 pieces two enhancements was observed; heat transfer rate and thermal performance, where, thermal performance of all cases exceeds unity (maximum thermal performance of 1.62 has been obtained by inserting 6 pieces of hexagonal nozzles turbulators). A comparison with another types of vortex generators shows the gap between the turbulator and heated surface offers a solution for problems occurred in the pipes of heat exchanger. The study therefore suggests a wider practical implementation of the turbulators.

Keywords: air heat exchanger, thermal performance, heat transfer, hexagonal turbulators, heat losses.

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1. Introduction

Heat exchangers have been used enormously in industrial applications. However, their performance plays an important role in the effective execution of myriads of industrial processes. For this reason, many researches have tended to improve the efficiency of heat exchangers, throughout simple approaches such as enhancement of the heat transfer rate. Whereby, the passive technique is the most eminent. By using this method, the turbulence can be developed in the working fluids

flow, in which it occurs through employing distinct augmentations such as nozzles, twisted tapes, coils, etc. However, the insertion of augmented turbulators requires an extra pumping power to circumvent the pressure drop generated. Due to this factor, a large number of researches have been carried out focused on the development and examination of various turbulators such that enhanced rate of heat transfer could be achieved with a minimal pressure drop. The performance of Conical and v-nozzles was evaluated by [1–3]. It was reported that 236–344 % increment of the heat transfer rate could be achieved with the insertion of conical nozzles. While the insertion of v-nozzles at pitch ratio=2 was reported to increased thermal effect. The influence of pitch ratio insertion with circular perforated rings was investigated by [4]. The study indicated that 0.92 is maximum thermal performance could be reached comparing to the plain tube. Different holes' shapes for the perforated nozzles such as (circular, triangular and square) studied by [5]. The study could achieve 1.7 of thermal performance and reported an increment in the friction factor values that ranging between 738–747 % higher than plain tube case. Inserted conical C-nozzles possessing different pitch and twisted ratios ($PR=2, 4 \& 7$ and Twisted ratio= $3.75 \& 7.5$). It was found that thermal performance could be reached to a level as high as 1.96 compared to a case having a plain tube [6]. Another investigated was tested the effect of conical cut-out turbulators with three different pitch ratios of 3, 4 and 5. They reported that inserting turbulators with pitch ratio of 3 offers maximum heat transfer by 315 % as compared to the plain tube case [7]. Performance of multiple divergent nozzles having various pitch ratios ($PR=2, 4, 6, 8, 10$). The pitch ratio of 2 was increased the rate of heat transfer to its maximum value of 215 % higher than the plain tube case [8]. While the friction factor decreases by about 52 % at the same pitch ratio. With twisted tape conical nozzles inserted [8] and combination of turbulators can produce highest rate of heat transfer [9, 10].

With an exhaustive literature review, it is clearly concluded that most of the researchers are focusing more on the manipulation of nozzles to enhance heat transfer. However, little effort has been made towards the structural reforms of the nozzles. In this paper, an unprecedented shape of nozzle has been proposed with a hypothesis of simultaneously increasing heat transfer rate and decreasing the pressure drop, thereby causing a significant reduction in the required pumping power of the working fluid.

Therefore, the objective of the study is toward into enhancement the thermal performance of heat exchanger by utilizing a novel turbulator design which an unprecedented shape of nozzle has been proposed with a hypothesis of simultaneously increasing heat transfer rate and decreasing the pressure drop, thereby causing a significant reduction in the required pumping power of the working fluid.

2. Materials and Methods

2.1. Instrumentations and Data Acquisition

Aluminum tube with a length (L) of 1350 mm having an inner and outer diameter of 45 & 50 mm respectively represents the main section of the heat exchanger experimental unit. To enable the production of uniform heat flux in the vicinity of the testing tube, an electrical wire was employed. Two layers of rubber and gypsum were mounted at the outside wall of the test tube in order to provide insulation and thereby reduction in the radial heat losses. Both ends of the tube were isolated by Teflon such that losses of heat could be circumvented. Settling chamber was used to ensure the working fluid flow is uniform when entering the tube test section. For obtaining the best temperature distribution on a stretch test tube wall, 18 thermocouples were used added to the existing 4 thermocouples. They are installed at the inlet and outlet of the system to measure working fluid temperature. In addition to that, two holes were being installed at the inlet and outlet of the test tube section to enable measurement of the pressure drop at these two points. The nozzles that used as turbulators in the present work are made from aluminum sheet having a thickness of 0.5 mm and a constant length (l) of 50 mm. The schematic of the open experimental setup and laboratory instruments components are shown in **Fig. 1, 2** respectively. Furthermore, the diameter ratio of the nozzles is $DR=D_n/d_n=0.5$ (where, D_n and d_n represent large and small nozzle diameters respectively) and their cross sectional shape is hexagonal that ranging in the tube between four to six pieces as shown in **Fig. 3** and for the distribution of them in the tube is shown in **Fig. 4, 5**.

The hexagonal nozzles were fixed inside the test tube by gluing them on a thin wire that has a diameter of 1 mm. The measurement devices and facilities that employed in this study are shown in Fig. 1 including; electric blower for pumping the working fluid through the open system, control valve to control the mass flow rate of the working fluid and therefore specifying the value of Reynolds number, U-tube manometer for measuring pressure drop between position 1 and position 2 of the tested tube, clamp meter & variac transformer for controlling heat flux. Finally, digital thermometer & selector switch for measuring temperatures in the whole system.

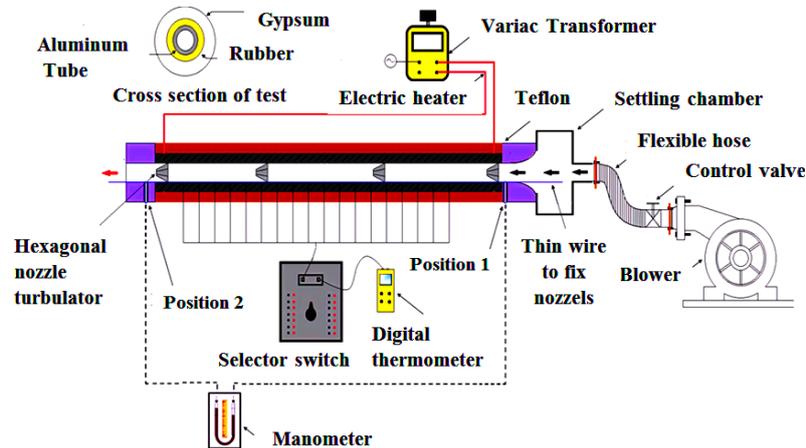


Fig. 1. Schematic diagram of heat exchanger experimental unit

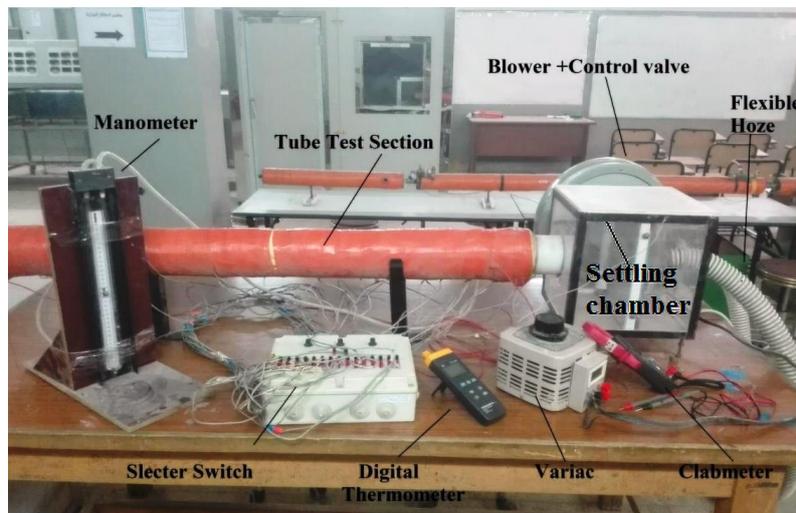


Fig. 2. Photographical diagram of heat exchanger experimental unit

At the initial start of the experiment, the electrical blower was switched on and the mass flow rate of the air was adjusted by the control valve through varying the flow cross sectional area. Consequently, this valve allows to adjust Reynolds number value at each run. It is important here to note that pumping of air inside the experimental unit is based on ten different inlet velocities corresponding to Reynolds number ranging between 6000 to 19500 at an interval of 1500 through the testing tube. Afterwards, adjustment of the variac transformer to the requisite heat flux that fed into the test tube. In addition to that, the initial pressure drop values, velocity of the working flow and open system temperatures were measured. Finally, the experimental unit reached the steady-state conditions by keeping continues running for about 2.5 to 3 hours. In this duration, the distribution of temperature was checked every 15 min. At the steady state of the system, the other constraints that include pressure drop and temperature along the system were checked simultaneously.

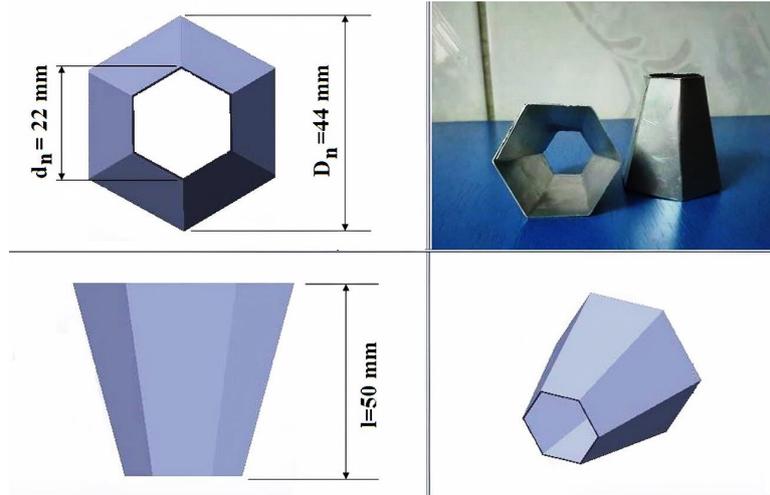


Fig. 3. Details of proposed nozzle turbulators

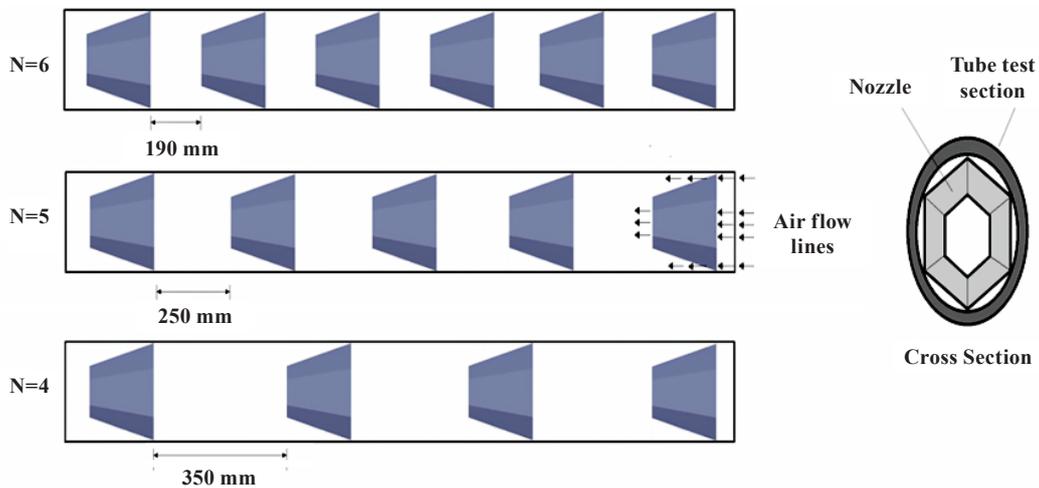


Fig. 4. Illustration figure of the distribution of the hexagonal nozzles at different numbers (N)

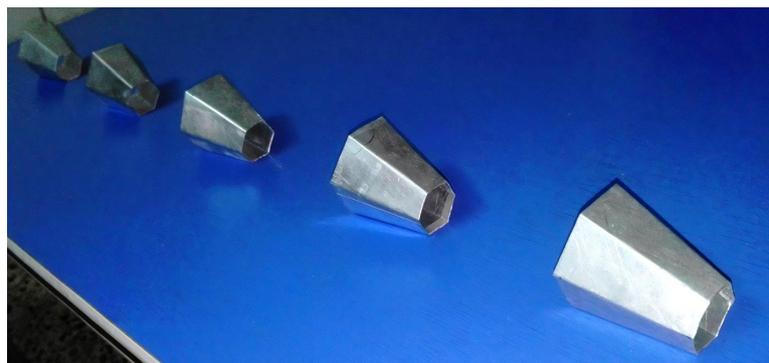


Fig. 5. Real picture of the hexagonal nozzles distribution

When turbulators inserted in the tube, the whole process of the experiment was similarly repeated for each turbulators number ($N=4, 5$ and 6).

2. 2. Data Processing

The governing equations for this study have been adopted form [9, 11]. However, all thermo-physical properties of the working fluid involving (thermal air conductivity (k_a), air density (ρ_a),

specific heat of air (C_{pa}) and air kinematic viscosity (ν_a) have been determined at the average bulk air temperature.

$$T_b = (T_{b,out} + T_{b,in}) / 2, \quad (1)$$

where $T_{b,in}$ and $T_{b,out}$ are entrance and exit air temperature ($^{\circ}\text{K}$) respectively.

In a steady state case, the net input heat (Q_n) to the working fluid is represented below.

$$Q_n = I_{in} \cdot V_{in} - H_{loss}, \quad (2)$$

where H_{loss} is the loss of heat from the tested tube to the surrounding environment by convection and radiation, and I_{in} and V_{in} are electrical current (A) and voltage (Volt) respectively.

In addition, the receiving of heat (Q_{air}) of the cold air (working fluid) in the test section is representing as follow.

$$Q_{air} = m_a \cdot C_{pa} (T_{b,out} - T_{b,in}), \quad (3)$$

where m_a is mass air flow (kg/s).

The net heat input to working fluid was higher than the heat absorbed by the working fluid approximately 4 % due to convection and radiation losses. Moreover, in [9] reported that the data that obtained from running the experiment could be accepted if the deference between the enthalpy rise and the net heat input is less than 5 percent.

Therefore, the average between Q_n and Q_{air} was adopted as the actual heat transfer (Q_{actual}) to test tube Section as follows.

$$Q_{actual} = \frac{Q_n + Q_{air}}{2} = h \cdot A \cdot (T_{wall} - T_b), \quad (4)$$

where A is the area of inner tube (m^2) and h is the coefficient of heat transfer, ($\text{W}/\text{m}^2 \cdot ^{\circ}\text{K}$).

The T_{wall} is the average surface temperature of the test tube, it can be determined using equation (5).

$$T_{wall} = \sum_{j=1}^{18} T_{wj} / 18, \quad (5)$$

where T_{wj} represents the local surface temperature value of the section of the test tube in $^{\circ}\text{K}$.

Numbers of Reynolds (Re), mean Nusselt (Nu) and friction loses (f) can be predicted from the below equations.

$$\text{Re} = \frac{U \cdot D}{\nu_a}, \quad (6)$$

$$\text{Nu} = \frac{h \cdot D}{k_a}, \quad (7)$$

$$f = \frac{\Delta P}{(L/D) \cdot (\rho_a U^2 / 2)}, \quad (8)$$

where ΔP is pressure loses in Pascal, D is the test tube inner diameter (m) and U is air velocity (m/s).

Thermal performance factor ($\eta_{thermal}$) is utilized to estimate the heat transfer improvement. However, the calculation is depending on the following expressions, as introduced by [4].

$$\eta_{thermal} = \frac{\text{Nu}_n / \text{Nu}_{plain}}{\left(f_n / f_{plain} \right)^{1/3}}, \quad (9)$$

where Nu_{plain} and f_{plain} are Nusselt number and friction factor of normal Tube Case, while, Nu_n and f_n are Nusselt number and friction factor of using unconventional nozzles case.

$\eta_{thermal}$ represents the ratio between the case of inserting turbulators to the smooth case at constant pumping power.

$$\text{Heat Transfer Enhancement} = \frac{Nu_n - Nu_{plain}}{Nu_n} \times 100\%. \quad (10)$$

Heat transfer enhancement is important for assessing the ability of proposed turbulators to endure in the working environment.

3. Experimental results and discussion

3.1. Validation of the plain tube

The empirical correlation that developed by the previous studies [4, 12] were employed for validating the results of the plain tube case. The results show a good agreement with the previous work as shown in Fig. 6, 7.

The results of the average Nusselt number obtained by the current study deviate by $\pm 7\%$ in comparison with the previous studies while the friction factor values deviate by $\pm 2.3\%$.

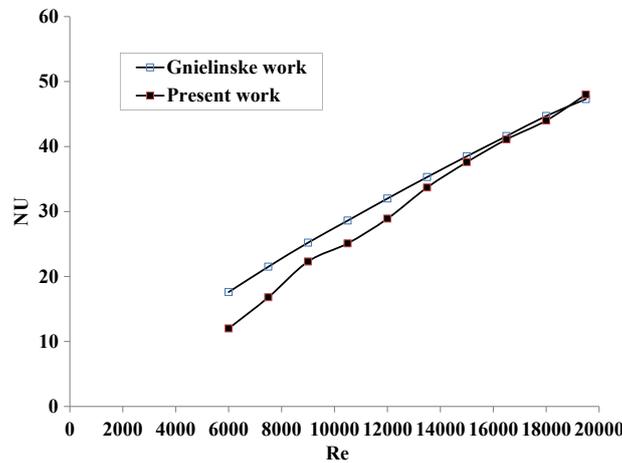


Fig. 6. Validation of heat transfer rate of plain tube case

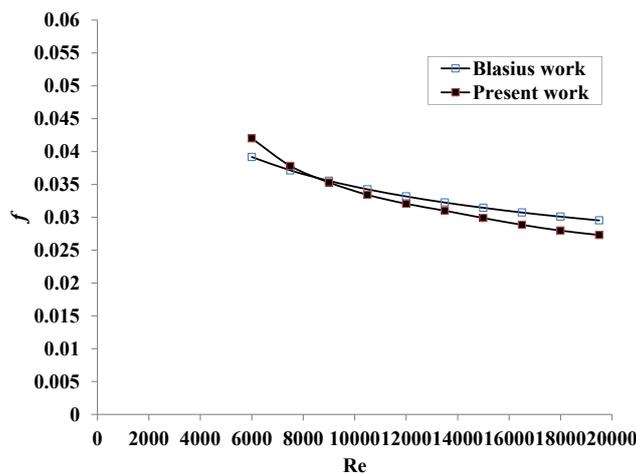


Fig. 7. Validation of friction factor of plain tube case

3.2. Analyses of Reynolds and Nusselt number

The relationship between Reynolds number and the average Nusselt number that calculated in (7) for the distinct turbulators is shown in Fig. 8. A direct proportional relationship was observed

between Reynolds number values, in which it attributed to the rising in velocity of the flow, thereby lead to increase turbulence that causing heat transfer due to convection [13]. In addition, the average Nusselt number for all cases of inserting turbulators are higher than plane tube case and this behavior ascribed the increasing of the contact between fluid particles and heated wall surface. This increment is representing the secondary flow (vortex and eddy motion) in vicinity near the hexagon nozzle in the downstream [14]. Furthermore, the insertion of hexagonal turbulators are also enhance the rate of heat transfer by 172 % for $N=4$ when compared to the plain tube. While 194 % enhancement for $N=5$ and 216 % for $N=6$.

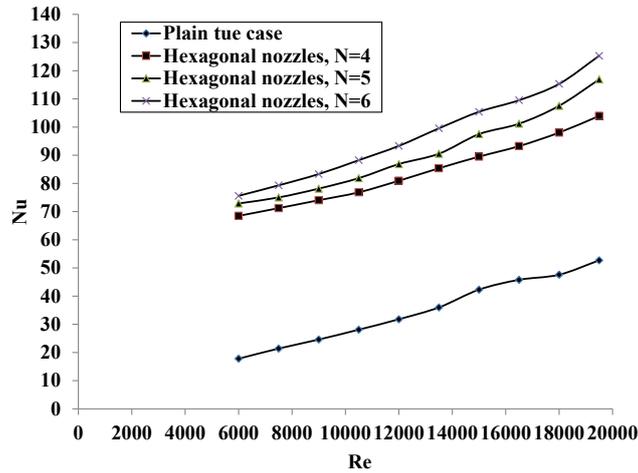


Fig. 8. Relation between the average Nusselt number and (Re) at different turbulators number

The extra numbers of turbulators that inserted above gave rise to the turbulence intensity of the flow, and thereby making thinner thermal boundary layer near the wall. So, these are considered as the two major reasons behind the increase in the heat convection. The average Nusselt number for all three cases of inserting turbulators was higher than the case of plane tube due to larger contact between fluid particles and the heated wall surface, that leading to generating vortices and eddy motion (secondary flow) in the vicinity of hexagon nozzle in the downstream.

3. 3. Analyses of friction factor

As depicted by Fig. 9, an inverse proportional relationship was observed between the friction factor's values (f) that are calculated by (8), and Reynolds number values for distinct quantities of turbulators.

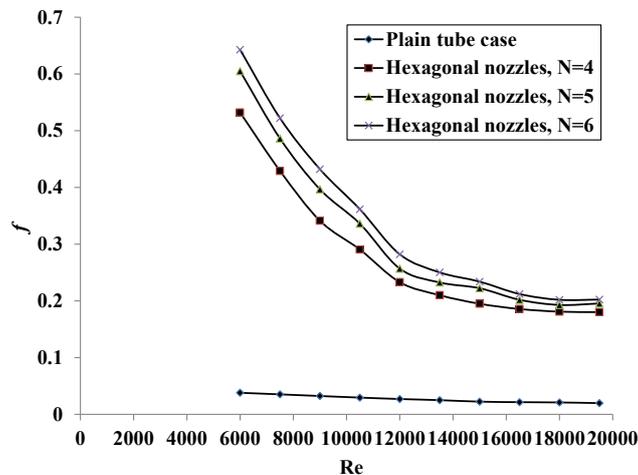


Fig. 9. The effect of different numbers of turbulators on the variation of friction factor (f) and Reynolds number

It is evident that, increase in turbulators' number leads to a significant impact on the friction factor value. Where increases by 722.9 % for $N=4$ compared to a plain tube case, and this goes up to 823.9 % and 886.7 % for $N=5$ and $N=6$ respectively. This could be mainly attributed to frictional losses that arise from increasing surface area and the higher intensity of the turbulent flow [15].

In addition, the figure shows that the friction factor reduces as Reynolds number increase due to the inverse relationship square velocity as shown in (8). Also, the results appeared that by inserting more turbulators increases the pressure drop, and this behavior could be attributed to losses which is generated due to friction that arises from increasing surface area and the higher intensity of the turbulent flow.

3. 4. Empirical correlations

Table 1 illustrates the empirical correlations that obtained from the current study of plain tube, as well as for the case of using turbulators.

Table 1

Empirical correlations of present work

| No. | Number of turbulators | Nusselt number empirical | Accuracy \pm % | Friction loses empirical | Accuracy \pm % |
|-----|-----------------------|---|------------------|--------------------------------|------------------|
| 1 | Normal case | $Nu=0.003Re^{0.994}Pr^{0.4}$ | 4.21 | $f=0.844Re^{-0.347}$ | 1.0 |
| 2 | 4 | $Nu=22.7Re^{0.353}Pr^{0.4}N^{-1.353}$ | 2.3 | $f=170.19Re^{-0.985}N^{1.985}$ | 5.2 |
| 3 | 5 | $Nu=23.845Re^{0.395}Pr^{0.4}N^{-1.395}$ | 3.16 | $f=186.76Re^{-1.039}N^{2.039}$ | 4.9 |
| 4 | 6 | $Nu=26.641Re^{0.423}Pr^{0.4}N^{-1.423}$ | 2.64 | $f=168Re^{-1.066}N^{2.066}$ | 4.4 |

It can be seen that the accuracy between the predicted and experimental values for both Nusselt number and friction factor are within ± 1 to 5.2 %.

3. 5. Enhancement of Thermal Performance

The assessment of thermal performance is utilized for the evaluation ability of the turbulators to endure in the working environment. (9) was used to obtain thermal performance that shown in **Fig. 10**. The figure illustrates a different thermal performance of the nozzle's numbers (4–6). It is noted that thermal performance in all nozzle cases exceeded the unity. This characteristic indicated that influence of friction loss creating with unconventional turbulators is below the heat transfer enhancement [16].

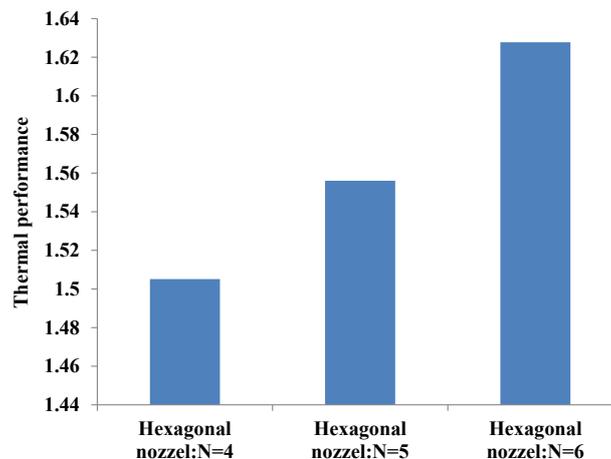


Fig. 10. Thermal performance Comparison with varies turbulator numbers

In addition to that, maximum thermal performance of 1.62 has been obtained by inserting 6 pieces of hexagonal nozzles turbulators. Further the proposed augmentations offer higher

thermal performance factor compared with some literatures [2, 14, 17] that proposed hexagonal ring, perforated conical ring and conical nozzle with snail entry respectively; and the enhancement in factor which is achieved ranged between 0.28–0.7.

The big disadvantage of proposed augmentations is the large pressure drop which rises with increasing the number of inserted turbulators, and to solve this problem, perforated hexagonal nozzles can be simulated by computation fluid dynamic or mathematical analyses in future work.

4. Conclusions

Reynolds number has a direct proportional relation with the rate of heat transfer while an inverse proportional relation with the friction factor.

The insertion of hexagonal nozzle turbulators causes a momentary increase in the rate of heat being transferred to the working fluid. Furthermore, insert additional turbulators enhance the heat transfer rate. However, the enhancement rate is 172 % higher than plain tube case for $N=4$, similarly for $N=5$ and 6 the enhancements are 192 % and 216 % respectively.

On the other hand, inserting turbulators causes a remarkable increase in the friction factor value to be about 722.9 % for $N=4$, 823.9 % for $N=5$ and 886.7 % for $N=6$ comparing to a plain tube case.

For all the three cases of $N=4$, 5 and 6, the value of the thermal performance exceeds unity. This well establishes the capability of hexagonal nozzle turbulators for being applied practically in industrial units.

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