Heat Transfer Enhancement in a Tube Fitted with Modified Nozzle-Turbulators and Wire-Coil

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Abstract--Experimental investigation has been conducted to study the enhancement of forced convection heat transfer by means of passive techniques for a turbulent air flow through an Aluminum test tube. The length to diameter ratio is 40. Reynolds number range from 6000 to 13500 with boundary conditions of constant heat flux of 357 W/m². The augmentation process is done by using six different arrangements which are; convergent nozzle CN, divergent nozzle DN, convergent-divergent nozzle CDN, divergent-convergent nozzle DCN, combined convergent-divergent nozzle CCDN and combined divergent-convergent nozzle CNDN. Then, the DN was modified by using three different modifications; perforation (triangle holes, square holes and circle holes), combining the Nozzle-Turbulator with Wire-Coil, and combining the Nozzle-Turbulator with drilled plate CNDP. The results at the same Reynolds number show that the CNDP provides higher heat transfer rates by 408% than that in a plain tube and friction factor 35 times that provided by the plain tube. On the other hand the perforated Nozzle-Turbulators with triangle holes give a thermal performance factor of 1.7 which is the highest thermal performance factor among all other augmentation devices used in the present study.

Index Term-- heat transfer; tube; conical ring; nozzle turbulator.

I. INTRODUCTION

Heat transfer techniques with a passive method have been developed and applied in many engineering applications such as heat exchangers, combustion chambers, gas turbine blades and electronic devices. Development of high-performance thermal systems stimulated the interest in methods to improve the heat transfer. In general the study of the improvement of the heat transfer performance is referred to as the heat transfer augmentation, enhancement, or intensification. The main thermo-hydraulic goals of the heat transfer enhancement include: - reducing the size of a heat exchanger required for a specific heat duty, increasing the heat duty of an existing heat exchanger, reducing the approach temperature difference for the process stream, or reducing the pumping power [1].

Many researchers studied the heat transfer augmentation techniques using modified twisted tape, modified conical ring, coil wire, etc. Promvonge and Eiamsa (2006) [2], showed that the using of the conical nozzle and the snail can help to increase considerably the heat transfer rate over that of plain tube by 278% and 206% respectively, and the use of conical nozzle in common with snail leads to maximum heat transfer rate that is up by 316% for range of Reynolds number was (8000-18000) at pitch ratios (2, 4 &7). They proved (2006) [3] for the same range of Reynolds number that the use of the V-nozzle can help to increase considerably the heat transfer rate at pitch ratio of (2, 4 and 7) to be about (270%, 235% and 216%), respectively; over the plain tube. Yongsiri, et al (2006) [4], used nozzles placed inside the inner test tube with the same values of Reynolds number and pitch ratio taken above. It was found that the heat transfer rate and friction factor increase with the decrease of pitch ratio. The maximum Nusselt numbers for both enhancement devices used with pitch ratios of (2, 4 & 7) were greater by (374%, 342% and 309%) respectively, than that the plain tube. Promvonge and Eiamsa (2007) [5], used DN and CN arrangements fitted inside tube at pitch ratios (2, 4 & 7) under Reynolds number range of 8000 to 18000. It was found that, the friction factor at a given Reynolds number increases with the reduction of pitch ratio and DN and CN arrangement gives higher heat transfer rate than the plain tube by 236 and 344%; respectively. Promvonge and Eiamsa (2007) [6], showed that the tube fitted with the conical-ring and twisted tape provided Nusselt number values higher than that with conical-ring alone with a Reynolds number range of 6000 to 2600. They submitted another study (2007) [7], presented the effect of a free-spacing entry together with conical-nozzle turbulators on heat transfer and friction characteristics in a uniformly heated tube. It was found that for the range of the Reynolds number of air extended from 8000 to 18000 that the Nusselt number values for employing both the enhancement devices with pitch ratio (2, 4 and 7) were higher than that for the plain tube around (315%, 300% and 285%) respectively. They extended this study (2007) [8] under the same values of Reynolds numbers and pitch ratios of [3] by using a set of converging-diverging nozzles like venturi structure (referred to as V-nozzle) which were placed inside the test tube while the snail was mounted at the tube entrance. The maximum improvement of heat transfer rate over the corresponding plain tube were found to be about 294%, 258% and 244%, for pitch ratio 2, 4 and 7, respectively. Promvonge (2008) [9], investigated the effect of the diameter ratios of the ring to tube diameter (0.5, 0.6 and 0.7) under three different arrangements (CN, DN & CDN) and Reynolds numbers ranging from 6000 to 26000. An augmentation of up to 197%, 333% & 237% in Nusselt number was obtained for the converging, diverging, and converging-diverging arrays respectively. Kongkait & Nanan (2010) [10], used three different numbers of perforated conical ring holes 4, 6 and 8 holes fitted inside tube with the range of Reynolds number between 4000 and 20000 under three different pitch ratios 4, 6 and 12. The results showed that the perforated conical-rings considerably diminishes the development of thermal boundary layer and leads to a heat transfer rate up to 137% over that in the plain tube. Kongkait & Nanan (2010) [11], used circular-ring turbulator for three different diameter ratios of 0.5, 0.6 and 0.7 with three different pitch ratios 6, 8 and 12 in a test tube under Reynolds
number of air ranged from 4000 to 20000. They concluded that the tube fitted with circular-ring turbulator enhances the heat transfer rate in around (57% to 195%) compared with that in the plain tube. Bankar and Pathare (2010) [12], examined V-nozzle inserts with pitch ratio 5.0 in a circular tube with L/D of 28 and Reynolds number ranged from 21500 to 48500. It was found that the use of the V-nozzle could help to increase considerably the heat transfer rate at about 140% over the plain tube with a maximum gain of enhancement efficiency of 1.19. Eiamsa and Promvonge (2010) [13], experimented the diamond-shaped turbulators in tandem arrangements inside tube with included cone angle of 15°, 30° and 45° and tail length ratios 1.0, 1.5 and 2.0 under Reynolds number ranged from 3500 to 16500. The results showed that the included cone angle decreases with the rise of the tail length ratio. For turbulator with 45° cone angle, the heat transfer was increased by 67%, 57% and 46% for tail length ratio 1.0, 1.5 and 2.0, respectively. Ibrahim and Kashif (2012) [14], used as trapezium-nozzle a passive technique to enhance heat transfer process inside tube with three different pitch ratios 2, 4 and 7, under Reynolds number ranged from 8000 to 16000. The authors concluded that the maximum gain of the Nusselt number was obtained for the smallest pitch ratio used (2) in the range of 202% to 257% compared with the plain tube. Karakaya and Durmus (2013) [15], devised the conical spring turbulators for three different conical arrangements (CN, DN and CDN) and three different cone angles 30°, 45° and 60° in Reynolds number range of 10000 to 34000. It was found that the best results in terms of heat transfer, are respectively DN, CDN and CN arrangements, while the turbulator best results were obtained, for cone angles 30°, 45° and 60°; respectively. Nishidha and, Basavaraj (2013) [16] found the heat transfer in tube with conical spring inserts is higher by 77.88% than that in smooth tube (i.e, without using any inserts). Durgesh et al [17] (2017) used copper and aluminum conical rings with two pitches of 3 cm and 5 cm respectively to enhancement heat transfer process in circular tube. The results show that the copper conical insert of 3 mm thick and 30 mm pitch has greater Nusselt number than aluminum conical ring insert in the range of 88% to 141%.

In order to achieve the goal of the present study; an appropriate testing unit was designed and constructed to investigate the turbulent forced convection heat transfer augmentation in a in a uniformly heated horizontal circular test tube for a specific range of Reynolds number from (6000 to 13500). The best transfer augmentation has been investigated by using the passive techniques of nozzle-turbulators, perforated nozzle-turbulator, combined nozzle-turbulators with wire coil and combined nozzle-turbulators with drilled plate. For all the previous augmentation procedures a performance evaluation has been made with comparison between the augmented arrangements and the plain tube to find the effect of the inserts on heat transfer and pressure drop, and to develop an empirical equation for the best tested augmentation procedures.

II. EXPERIMENTAL SETUP

The experimental rig is shown schematically in Fig.(1). The experimental setup (the open air flow loop) can be divided into two main sections, the air supply section and the heating section. The open air flow loop used in the system consists of an electrical centrifugal air blower with 2800 rpm and 764 W AC power motor. This blower is the source that provides air flow for the system. A flow control valve (ball valve) was installed on the pipe line located on the blower outlet to control the air flow. A scaled panel was fixed to the valve and used with the flow control valve handle to indicate the amount of air flow through the system.

The outlet of the control valve is connected with a cast iron pipe that supplies the orifice plate with the air flow. The dimensions of the cast iron pipe are 1510 mm length and 75 mm outer diameter and 71 mm inner diameter. The orifice plate has been installed on the pipeline to measure the flow rate of the air that enters the heat transfer testing unit. The orifice plate is then connected to a cast iron pipe with a length of 480 mm and outer diameter of 75 mm and inner diameter of 71 mm. The cast iron pipe is connected directly to a flexible hose which is attached firmly to the settling chamber to avoid any leakage. The settling chamber is constructed from a cubic iron frame with six Perspex walls (300 mm x 300 mm). It is used to reduce the fluctuations in the air flow before entering the test tube by using the straightener that is located inside the settling chamber.

The heating section was designed and constructed carefully to study the forced convection heat transfer process and the friction factor with recording the data of (temperature, volumetric flow rate and the pressure drop) of the bulk air flow at the steady state condition. It is composed of a 1350 mm length of aluminum pipe with 45 mm inner diameter and 2.5 mm of its wall thickness. To achieve the uniform heat flux boundary condition, the aluminum pipe was heated electrically by using a continuous and uniform winding of a flexible electric heater wire type KANTHAL made from (FeCrAl) with a pitch of 10 mm. The heating wire has an 0.6 mm diameter and a 26 cm length. It was carefully placed in ceramic beads to insulate it electrically from the aluminum pipe. To reduce the heat loss from the test unit a multilayer thermal insulation was added. The first layer is the asbestos rope with a thickness of 20 mm. The second insulation layer is the plaster with a thickness of 5 mm followed by a layer of glass wool with a thickness of 7 mm.

The temperature of the Aluminum pipe surface was measured by fixing 18 Chromel Alumel calibrated thermocouples (type K) along it, each one of thermocouples was fixed on the pipe by drilling a hole in the pipe, a non penetrative hole with 2 mm diameter and 2 mm depth. The thermocouple was inserted in the hole and pasted to it with a high temperature epoxy which was carefully refined with the sandpaper after drying is achieved. Also four thermocouples were attached on the multilayer thermal insulation surface to determine the heat losses by conduction to the surrounding. At the inlet of the heating section, a Teflon bell mouth has been carefully fabricated to form the entrance to the test tube. It was fixed firmly between the heating section entrance and the settling chamber outlet hole. Another Teflon piece has been attached at the exit heating section. Both the inlet bell mouth and the exit disk Teflon pieces have been used because of their higher thermal resistance to reduce the axial heat losses from the test tube. Pressure tapings were manufactured and fixed.
permanently on the inlet and outlet tube to find the pressure drop through the heating section by a water manometer connected to the tapping. A single thermocouple was attached inside the settling chamber to record the inlet air temperature to the heating section while there are two thermocouples located on the exit teflon disk to record the average outlet air temperature.

III. AUGMENTATION TECHNIQUES

The turbulaters are made from Aluminum with a truncated hollow cone shape. The nozzle-turbulators are constricted with the length of \( a=45 \) mm and with ends diameters of \( D=45 \) mm and \( d=22.5 \) mm. The insertion of the nozzle-turbulators were made thoroughly inside the heating section by press with pitch ratio of \( (PR = l/D = 5) \). The nozzle-turbulator was employed to generate the reverse flow from the separation and reattachment of the boundary layer occurring between the nozzle-turbulators.

**Nozzle-turbulators**

They were mounted over the testing unite with six different arrangements, as illustrated in Fig.2, as follows: Converging nozzle-turbulators array (CN), Diverging nozzle-turbulators array (DN), Converging-diverging nozzle-turbulators array (CDN), Diverging-converging nozzle-turbulators array (DCN), Combined converging-diverging nozzle-turbulators array (CCDN), and Combined diverging-converging nozzle-turbulators array (CDCN).

**B. Perforated Nozzle-turbulators**

Nozzle-turbulator was perforated with four holes which had three geometric shapes; circle, square, and triangle; as shown in Fig.3-a, b, &c. All the holes (regardless of its shape) were fabricated to have the same area of 100 mm².

**C. Nozzle- Turbulators Combined with Drilled Plate in the Outlet of the Nozzle**
The aluminum circular plate was designed to be fitted in the bigger end of nozzle-turbulators that arranged inside the heating section with two holes at diameter of 10 mm. These combined Nozzle-turbulator were arranged in the test tube with an array such that the holes in the plates to be crossed as shown in Fig.(3-d).

D. Nozzle-Turbulators Combined with Wire Coil.

The wire coil was designed to be fitted in the core of the nozzle-turbulators arranged inside the heating section. Two wire coils with different coil pitch were used in the present work. The wire coils are made of steel with wire diameter of $b=2$ mm and coil outer diameter of $B=21$ mm which is a little less than the small end diameter of the nozzle-turbulators to facilitate the insertion of the wire coil inside it. The length of wire-coil used is $L=1.35$ m and the coil pitches used are $e=8$ mm and $14$ mm, as shown in Fig. (4).

IV. Data Analysis

To analyze the process of heat transfer from the test tube to the air, a software program was applied using the (MATLAB-2009) to perform all calculations of the data analysis. By taking all the thermophysical properties of air as follows:

The overall bulk air temperature ($T_b$) is calculated as follows:

$$T_b = \frac{T_i + T_o}{2}$$

The heat transferred to the air by convection can be calculated as follows:

$$Q_a = \dot{m}C_p(T_o - T_i)$$

$$\dot{m} = \rho UA_c$$

The heat transferred from the test tube to the working fluid by convection ($Q_{conv}$) can be calculated in another manner as follows:

$$Q_{conv} = Q_e - Q_{cond} - Q_{rad} - Q_{loss}$$

Where ($Q_e$) is the heat supplied by the electrical winding heater on the test tube that is taken as the total input power which is calculated from:

$$Q_e = VI$$
$Q_{cond}$ represents the heat loss by conduction that is found by Fourier's law as follows:

$$Q_{cond} = \frac{T_{\text{ins}} - T_{\text{w}}}{R_T}$$

(6)

where ($T_{\text{ins}}$) is the average insulation outer surface temperature and ($T_{\text{w}}$) is the average test tube surface temperature that is recorded from eighteen points as follows:

$$\bar{T}_w = \frac{\sum T_w}{18}$$

(7)

where ($T_{\text{w}}$) is the local test tube surface temperature. While ($R_T$) is the total thermal resistance of the multilayer insulation which is found by:

Each one of the thermal resistances ($R$) could be found by:

$$R = \ln \left(\frac{r_o}{r_i}\right) / 2\pi k_L L$$

(8)

the heat transfer by radiation is calculated by Stefan-Boltzmann law:

$$Q_{\text{rad}} = F_s F_G \sigma A_s (\bar{T}_w^4 - \bar{T}_b^4)$$

(9)

the heat lost axially from the test tube is calculated to give an average value of 3% from the total heat supplied by the electrical heater:

$$Q_{loss} = 0.03 Q_T$$

(10)

Finally, the heat gained by air with convection ($Q_c$) can be calculated as follows:

$$Q_c = (Q_a + Q_{\text{conv}}) / 2$$

(11)

To find the average heat transfer coefficient ($h$), Newton's second law of cooling is used as follows:

$$h = Q_c / \left[A_s (\bar{T}_w - \bar{T}_b)\right]$$

(12)

The friction factor ($f$) is found from:

$$f = \Delta P_t / (L / D) (\rho U^2 / 2)$$

(14)

where:

$\Delta P_t$ = pressure drop through the test tube (Pa).

V. Results and Discussion

A. Nozzle-Turbulators

Generally all the nozzle-turbulator arrangements leads to a considerably higher heat transfer rates than that of the plain tube. Fig.(5) reveals that the DN arrangement gives higher values of heat transfer rate (average Nusselt number) than that of other nozzle-turbulator arrangements due to its geometry that provided the greatest interruption to the flow with the greatest contact surface area between the heated test tube wall and the tested fluid, and these values increase with increasing of Reynolds number. The enhancement of the heat transfer rate by using the nozzle-turbulators is attributed to its influence in disrupting the boundary layer and the reverse flow that enhance the convection heat transfer process. In fact the best mixing between the wall region and the core region results in the highest disruption in the boundary layer that enhances the process of convection heat transfer. Also the reverse flow resulted from the nozzle-turbulator enhances the convection by augmenting the effective axial Reynolds number with reducing the flow cross sectional area, which affect directly and severely the mean velocity and temperature gradient that produce greater heat fluxes and momentum.

because of the larger effective potential force. The DN, CN, CDN, CDCN, CCDN and CDCN presented a heat transfer rate higher than that of the plain tube by 317%, 231%, 268%, 295%, 111% and 187% respectively.
fitted with the perforated nozzle-turbulator (regardless of the holes shape) are higher than that of the plain tube at the same Reynolds number. This can be attributed to the perforated nozzle-turbulator effect on the destruction of the thermal boundary layer near the surface of the test tube which is caused by the interruption of the air flow through the perforated nozzle-turbulator. Although all these perforated nozzle-turbulators have the same number of holes with the same area (in spite of its different hole shapes), but the results show that they do not provide the same heat transfer rate. Actually the heat transfer rate for triangle holes, circle holes and square holes were 253%, 245% and 231% higher than those of the plain tube, respectively; on the corresponding Reynolds number. This is mainly because of the geometry of the hole that has a specific effect on the air flow inside the hole. The vortices formed and the reverse flow that takes place in the area between the wall of the test tube and the nozzle-turbulator have a direct impact on the rate of heat transfer. The triangle hole creates a higher disruption to the flow than that created by the circular hole and the squire hole. On the other hand the results reveal that using the traditional nozzle-turbulator gives a higher heat transfer rates than the heat transfer rates given by the perforated nozzle-turbulator at the same Reynolds number in about (64%-86%) due to the lower turbulence intensity in the tube provided by the perforated Nozzle-turbulator.

The effect of the perforated nozzle-turbulator on the pressure drop across the plain tube is shown in Fig.(8). This figure reveals that there is a significant increase in the pressure drop by using the perforated nozzle-turbulator compared with that of the plain tube at the same Reynolds numbers. The experimental data shows that the friction factor for the perforated Nozzle-turbulator with triangle holes, circle holes and square holes were about 747%, 740% and 738%, respectively; higher than those of the plain tube. As it could be noticed that the difference between these friction factors percentage ratios do not cross 9% but in fact they lead to a much bigger difference between there corresponding average Nusselt numbers that reaches up to 22%. It can be observed that the friction factors in the tube fitted with perforated nozzle-turbulator are smaller than those provided by the traditional nozzle-turbulator at the corresponding Reynolds number. This can be attributed to the fact that the perforation with any shape reduce the obstructing of the nozzle-turbulator to the air flow which lead certainly to reduce the pressure drop across the plain tube.

**C. Nozzle-Turbulators Combined with Wire Coil**

The effect of using combined divergent nozzle-turbulator with wire-coil on the convective heat transfer rate is plotted in Fig.(9). It illustrates that the heat transfer rates of the tube equipped with the combined nozzle-turbulator and wire-coil are higher than those for the plain tube by (252% and 194%) and lower than those for the divergent nozzle-turbulator by (65% and 123%) for the same given Reynolds number at wire-coils with 14 mm coil pitch and the 8 mm coil pitch respectively. This is due to the fact that combining the nozzle-turbulator with the wire-coil produces an increase in the reverse flow that enhances the convection process. But at the same time the combination of these two devices reduce the mixing between the wall flow region and the core flow region that reduces the convection due to the wire-coil effect which makes the heat transfer rates of the combined nozzle-turbulator with wire-coil lower that those obtained from the divergent nozzle-turbulator alone. The results also
demonstrate that the increasing of coil pitch increases the heat transfer rate because of a better mixing between the flow regions which obviously increase the convection. It is noticed also that the tube fitted with combined nozzle-turbulator and wire-coil with 14 mm coil pitch gives a higher heat transfer rate than those provided by the combination with 8 mm pitch.

![Figure 9](image1.jpg)

**Fig. 9.** Variation of average Nusselt number with the Reynolds number for combined Nozzle-Turbulator with Wire-Coil.

The effect of fitting the test tube with combined nozzle-turbulator and wire-coil on the pressure drop through the test tube is illustrated in Fig. (10) which shows that the pressure drop provided by the combined nozzle-turbulator and the wire-coil was significantly higher than that of the plain tube at the same Reynolds numbers. Also the results show that using 8 mm coil pitch gives a higher pressure drop than that with 14 mm coil pitch and that of the divergent nozzle-turbulator alone in the Reynolds number range (7000 to 11500). This is a reasonable result because the wire-coil insert provides an extra friction between the air flow core region and the wall region.

![Figure 10](image2.jpg)

**Fig. 10.** Variation of friction factor with the Reynolds number for combined Nozzle-Turbulator with Wire-Coil.

This figure demonstrates that at a given Reynolds number and for the whole range investigated, the heat transfer rates in the tube fitted with this combination of turbulators are higher than those of the plain tube and those of the traditional divergent nozzle-turbulator. In other words, this combination of turbulators provides an enhancement to the heat transfer rate in the tube with 408% and 91% over that provided by the plain tube and divergent nozzle-turbulator, respectively. This enhancement could be attributed to the high destruction of the thermal boundary layer and the great turbulence generation. Also, the augmentation of heat transfer rate is greatly amplified at high Reynolds numbers and leads to higher turbulence level which considerably increase the convection heat transfer rates.

In addition, these combined nozzle-turbulators are arranged in the test tube with array that the holes in the plates to be crossed that provide a high circulation of the air flow to give a superior heat transfer rate than any other augmentation devise or combination of devices tested in the present study.

**Nozzle-Turbulators Combined with Drilled Plate in the Outlet of the Nozzle**

The influence of using a nozzle-turbulator combined with drilled plate on the heat transfer rate is plotted in Fig. (11).

![Figure 11](image3.jpg)

**Fig. 11.** Variation of average Nusselt number with the Reynolds number for combined Nozzle-Turbulator with drilled plate.

This figure demonstrates that at a given Reynolds number and for the whole range investigated, the heat transfer rates in the tube fitted with this combination of turbulators are higher than those of the plain tube and those of the traditional divergent nozzle-turbulator. In other words, this combination of turbulators provides an enhancement to the heat transfer rate in the tube with 408% and 91% over that provided by the plain tube and divergent nozzle-turbulator, respectively. This enhancement could be attributed to the high destruction of the thermal boundary layer and the great turbulence generation. Also, the augmentation of heat transfer rate is greatly amplified at high Reynolds numbers and leads to higher turbulence level which considerably increase the convection heat transfer rates.

In addition, these combined nozzle-turbulators are arranged in the test tube with array that the holes in the plates to be crossed that provide a high circulation of the air flow to give a superior heat transfer rate than any other augmentation devise or combination of devices tested in the present study. Fig. (12) presents the influence of using the combined nozzle-turbulator with drilled plate on the pressure drop. It shows that the use of this combination of turbulators leads to a substantial increase in friction factor 36 times that of the tube alone. And with 91% if the divergent nozzle-turbulator is used without drilled plate.

![Figure 12](image4.jpg)

**Fig. 12.** Variation of friction factor with the Reynolds number for combined Nozzle-Turbulator with drilled plate.
This high friction factor can be attributed to the increase in the generated turbulence provided by the effect of the drilled plate in presenting a high blockage to the flow that is offered even a stronger turbulence intensity.

VI. PERFORMANCE EVALUATION

It is important to evaluate the performance of each enhancement technique that has been used in the present study to find the most practical technique. This evaluation is done by calculating both effects of the enhancement device; on the heat transfer rate and on the pressure drop together by presenting them in to the form of the thermal performance factor ($\eta$) at constant pumping power which is:

$$\eta = \frac{(Nu_{d,E}/Nu_{d,t})}{(f_E/f_t)^{1/2}} \quad (15)$$

where:

- $Nu_{d,E}$ = average Nusselt number with the enhancement device.
- $Nu_{d,t}$ = average Nusselt number for the plain test tube.
- $f_E$ = friction factor with the enhancement device.
- $f_t$ = friction factor for the plain test tube.

As shown in Figs. (13 to 16), the variation of the thermal performance factor increases with the increasing of Reynolds number for all the cases of the augmentation devices. This is attributed to that increasing the Reynolds number leads to increasing the Nusselt number and decreasing the friction factor.

Fig. (13) reveals that for most of the tested range the divergent nozzle-turbulator arrangement provide the highest thermal performance factor except for that at Reynolds number less than 9000, at which the divergent-convergent arrangement provide a greatest thermal performance factor. Therefore, the divergent nozzle-turbulator arrangement will be taken as a comparison reference with other augmentation cases. Fig. (14) illustrates that for the perforated nozzle-turbulator, the triangle holes provide the highest thermal performance factor. Also, they provide thermal performance factor higher than that of the divergent nozzle-turbulator.

Fig. (15) shows that combined nozzle-turbulator and wire-coil with 14 mm coil pitch provides a higher thermal performance factor than that of the 8 mm coil pitch. Also the combined nozzle-turbulator with the wire-coil for both coil pitches gives a lower thermal performance factor than that of the divergent nozzle-turbulator. Fig. (16) explains that the combined nozzle-turbulator with the drilled plate provide a higher thermal performance factor than that of the divergent nozzle-turbulator.
without drilled plate for the Reynolds number higher than 10500.

Fig. 15. The effect of combining the Nozzle-Turbulator with the Wire-Coil on the thermal performance factor.

Fig. 16. The effect of combining the Nozzle-Turbulator with the drilled plate on the thermal performance factor.

VII. EXPERIMENTAL CORRELATIONS

By analyzing the data obtained from using all the augmentation devices in the present study to develop the experimental correlations in terms of average Nusselt number and friction factor for the range of Reynolds number from 6000 to 13500, with using the least square method in the (STATISTICA 6.0) program to evaluate the constant values (a, b, n and m) for the following general empirical equations of average Nusselt number and friction factor:

\[ \text{Nu}_d = a \text{Re}_d^b \text{Pr}^{0.4} \]  \hspace{1cm} (16)

\[ f = n \text{Re}_d^m \]  \hspace{1cm} (17)

The values of a, b, n, and m are given in Table I.

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For perforated divergent Nozzle-turbulator and Nozzle-turbulator fitted with Wire-Coil; the average Nusselt number and friction factor empirical equations at the same range of Reynolds number given above are respectively given as follows:

\[ \text{Nu}_d = 0.194917 \text{Re}_d^{0.5746} \text{Pr}^{0.4} Z^{0.1111} \]  \hspace{1cm} (18)

\[ f = 118.2535 \text{Re}_d^{-0.6384} Z^{0.0247} \]  \hspace{1cm} (19)

\[ \text{Nu}_d = 0.062047 \text{Re}_d^{0.814} \text{Pr}^{0.4} y^{0.31} \]  \hspace{1cm} (20)

\[ f = 3956.992 \text{Re}_d^{-0.942} y^{-0.13} \]  \hspace{1cm} (21)

where:

- \( Z \) is the ratio of hole perimeter to the nozzle-turbulator length.
- \( Y = e/B \) is the Wire-Coil pitch ratio.

It was found that the maximum deviation for the Nusselt number were (±3%) while the maximum deviation for the friction factor was (±9%) for all cases.

VIII. VALIDITY

The present experimental results with the heat transfer in a uniformly heated circular tube without turbulators (plain tube) have been compared for validity with the previous works in the form of average Nusselt number and friction factor. The correlations of Kongkait Paiboon are given as follows:

\[ \text{Nu} = 0.057 \text{Re}^{0.709} \text{Pr}^{0.4} \]  \hspace{1cm} (22) For \( 4000 \leq \text{Re} \leq 2 \times 10^4 \)

\[ f = 0.458 \text{Re}^{-0.284} \]  \hspace{1cm} (23) For \( 4000 \leq \text{Re} \leq 2 \times 10^4 \)

The verification result are shown in Fig.(17) and Fig.(18). It can be noticed that from the verification results that the average Nusselt number and friction factor values of the present plain tube are in agreement with the previous correlations.
IX. CONCLUSIONS

Experimental investigation to study the enhancement of forced convection heat transfer by means of passive techniques for a turbulent air flow through an Aluminum test tube has been carried out. Several conclusions can be drawn as follows:
1. All the turbulators used in the present study give an increasing heat transfer rate and the pressure drop through the test tube compared with the case of plain tube for the tested range of Reynolds number.
2. The divergent arrangement of the Nozzle-Turbulator provides the highest heat transfer rate and the highest pressure drop among all other nozzle-turbulator arrangements.
3. For the perforated divergent nozzle-turbulator:
   a) The triangle hole shape perforation provided a higher heat transfer rates than those of the circular and square hole shapes perforation.
   b) The shape of the perforation hole does not affect on the pressure drop through the test tube in all the tested shapes.
   c) The perforated divergent nozzle-turbulator (for all the tested hole shapes) reduces the heat transfer rate and the pressure drop compared with the case of non perforated divergent nozzle-turbulator.
4. For combined divergent nozzle-turbulator with the wire-coil, the heat transfer process improves as coil pitch increases with decreasing of pressure drop, and gives a lower heat transfer rate than that of the divergent nozzle-turbulator alone.
5. The combined divergent nozzle-turbulator with the drilled plate offers the highest heat transfer rate and the highest pressure drop over all other cases taken in the present study.
6. The perforated nozzle-turbulator with the triangle holes provides the highest thermal performance factor except for Reynolds number higher than (11500) after which the combined nozzle-turbulator with the drilled plate gives the greatest thermal performance factor.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area ( (m^2) )</td>
</tr>
<tr>
<td>B</td>
<td>coil outer diameter ( (m) )</td>
</tr>
<tr>
<td>C_o</td>
<td>orifice discharge coefficient</td>
</tr>
<tr>
<td>C_p</td>
<td>specific heat of air ( (kJ/kg.K) )</td>
</tr>
<tr>
<td>D</td>
<td>tube diameter ( (m) )</td>
</tr>
<tr>
<td>d</td>
<td>nozzle-turbulator small end diameter ( (m) )</td>
</tr>
<tr>
<td>e</td>
<td>coil pitch ( (m) )</td>
</tr>
<tr>
<td>f</td>
<td>friction factor</td>
</tr>
<tr>
<td>F_Ɛ</td>
<td>emissivity=0.04</td>
</tr>
<tr>
<td>F_G</td>
<td>view factor=1</td>
</tr>
<tr>
<td>h</td>
<td>average heat transfer coefficient ( (W/m^2.K) )</td>
</tr>
<tr>
<td>I</td>
<td>current (Ampere)</td>
</tr>
<tr>
<td>j</td>
<td>base length of the triangle perforation in the nozzle-turbulator ( (m) )</td>
</tr>
<tr>
<td>L</td>
<td>test tube length ( (m) )</td>
</tr>
<tr>
<td>l</td>
<td>pitch length between nozzle-turbulator ( (m) )</td>
</tr>
<tr>
<td>m</td>
<td>mass flow rate of air ( (kg/s) )</td>
</tr>
<tr>
<td>Nu_d</td>
<td>average Nusselt number ( =h.D/k )</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number ( =\mu C_p/k )</td>
</tr>
<tr>
<td>Q&quot;</td>
<td>volumetric flow rate ( (m^3/s) )</td>
</tr>
<tr>
<td>R</td>
<td>thermal resistance ( (m^2.K /W) )</td>
</tr>
<tr>
<td>Re_d</td>
<td>Reynolds number based on diameter ( =\rho U D/\mu )</td>
</tr>
<tr>
<td>r_i</td>
<td>inner radius of lagging ( (m) )</td>
</tr>
<tr>
<td>r_o</td>
<td>outer radius of lagging ( (m) )</td>
</tr>
<tr>
<td>s</td>
<td>diameter of the hole in the drilled plate ( (m) )</td>
</tr>
<tr>
<td>t</td>
<td>diameter of the circular perforation in the nozzle-turbulator ( (m) )</td>
</tr>
<tr>
<td>T</td>
<td>temperature ( (K) )</td>
</tr>
<tr>
<td>T_o</td>
<td>outlet air temperature ( (K) )</td>
</tr>
<tr>
<td>T_i</td>
<td>inlet air temperature( (K) )</td>
</tr>
<tr>
<td>T_w</td>
<td>average wall temperature ( (K) )</td>
</tr>
<tr>
<td>U</td>
<td>mean axial velocity of air ( (m/s) )</td>
</tr>
<tr>
<td>V</td>
<td>voltage ( (volt) )</td>
</tr>
<tr>
<td>Z</td>
<td>hole perimeter to nozzle-turbulator length ratio</td>
</tr>
<tr>
<td>ΔP</td>
<td>pressure drop ( (Pa) )</td>
</tr>
</tbody>
</table>

Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>μ</td>
<td>viscosity of air ( (kg/s.m) )</td>
</tr>
<tr>
<td>β</td>
<td>diameter ratio</td>
</tr>
<tr>
<td>κ</td>
<td>thermal conductivity ( (W/m.K) )</td>
</tr>
<tr>
<td>η</td>
<td>thermal performance factor</td>
</tr>
<tr>
<td>ρ</td>
<td>air density ( kg/m^3 )</td>
</tr>
</tbody>
</table>
\[ \sigma \] Stefan-Boltzmann constant \(=5.669 \times 10^{-8} \text{ (W/m}^2\text{.K)}\)

**Subscripts**

- a: air
- b: bulk air
- c: cross section
- E: enhancement device
- s: surface
- t: total
- w: wall

**Abbreviations**

- CDN: Convergent Divergent Nozzle-Turbulator
- CHF: Constant Heat Flux
- CN: Convergent Nozzle-Turbulator
- CWT: Constant Wall Temperature
- DCN: Divergent Convergent Nozzle-Turbulator
- DN: Divergent Nozzle-Turbulator
- OTEC: Ocean Thermal Energy Conversion

**REFERENCES**


