Heat Transfer Enhancement in a Tube Fitted with NozzleTurbulators, Perforated Nozzle-Turbulators with Different hole shap

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ABSTRACT

The present study is to investigate experimentally the enhancement of forced convection heat transfer by means of passive techniques for a turbulent air flow through an Aluminum test tube. Reynolds number range is from 6000 to 13500 with boundary conditions of constant heat flux. The augmentation process is done by using divergent Nozzle-Turbulator arrangement with and without perforation models (triangle holes, square holes, and circle holes). The experimental results at the same Reynolds number show that the divergent nozzle-turbulators without perforation provides the highest heat transfer rate 317% and highest friction factor 17 times over that of plain tube with a performance factor of (1.58). On the other hand the perforated Nozzle-Turbulators with triangle holes gave a thermal performance factor of (1.7) which is the highest thermal performance factor among all other perforated and non perforated nozzle turbulators used in the present study

تحسين انتقال الحرارة في انبوب مزود بمضطربات مخروطية، مضطربات مخروطية مثقبة ومضطربات مخروطية متحدة مع صفيحة مثقبة

الخلاصة

يقدم العمل الحالي دراسة عملية لتحسين انتقال الحرارة بالحمل القسري باستخدام طريق التحسين الكامن لجريان الهواء المضطرب المار خلال مقطع اختبار مصنوع من الالمنيوم ليغطي مدى من عدد رينولدز من (6000 الى 13500) بثبوت الفيض الحراري كشرط حدي. تحسين انتقال الحرارة تم باستخدام مضطربات مخروطية متباعدة مثقبة و غير مثقبة. الثقوب ذات أشكال (مثلثة, مربعة, و دائرية). أظهرت النتائج العملية لعدد Re واحد بانه المرحلة الاولى للتحسين ترتيب المضطربات المخروطية المتباعدة توفر معدل انتقال حرارة وبمقدار (317%) اعلى من معدلات انتقال الحرارة التي يوفرها مقطع الاختبار لوحده, كذلك فانها تزيد معامل الاحتكاك بمقدار (17) مرة اكثر من الذي يوفره مقطع الاختبار لوحده كما انه يوفر اعلى معامل اداء وهو (1.85). من ناحية اخرى فان

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المضطربات المخروطية المثقبة بثقوب مثلثة الشكل توفر معامل اداء مقداره (1.7) وهو اعلى معامل اداء من جميع طرق التحسين المستخدمة للدراسة الحالية.

INTRODUCTION

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. Generally most of the augmentation techniques in the engineering applications widely employ the reverse flow devices "re-circulation flow". The reveres flow with high turbulence can improve convection of the tube wall by increasing the effective axial Reynolds number, decreasing the flow cross-section area, and increasing the mean velocity and temperature gradient. The insertion of the turbulators in the flow passage is one of the favorite heat transfer augmentation techniques in the petrochemical industry where the specifications codes are required. Insert devices can be used since they do not modify round tube mechanical properties as integral roughness does. Another application is in some certain systems, particularly those in space vehicles where the requirement for an augmentation device is essential for a successful operation.

Many researchers studied the heat transfer augmentation techniques using modified twisted tape, modified conical ring, coil wire, etc. Promvonge and Eiamsa [1], 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 in another study **[2]** 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 but the increase in friction factor was much higher than that of the increase in Nusselt number at the same Reynolds number for all pitch ratios with Reynolds number from 8000 to 18000. Yongsiri, et al [3], used nozzles placed inside the inner test tube with three various pitch ratios (2, 4 and 7), while the snail swirl generator was mounted at the entry of the test tube with the Reynolds number range 8000 to 18000. 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 (374%, 342% and 309%) respectively, in comparison with the plain tube. Promvonge and Eiamsa [4], 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. Kongkaitpaiboon et al [5], studied the perforated conical-ring with different pitch ratios and different numbers of perforated holes, they found that the perforated conical-rings leads to a heat transfer rate up to 137% over that in the plain tube. The perforated conical-rings enhanced heat transfer more than typical conical-rings on the basis of thermal performance factor of around 0.92 at the same pumping power, the thermal performance factor of 0.92 was found at the smallest pitch ratio and maximum number of holes. Bankar and

Pathare [6], 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 [7], 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 [8], 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 [9], devised the conical spring turbulators for three different conical arrangements (converging, diverging and converging diverging) 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 diverging, converging diverging and converging arrangements, while the turbulator best results were obtained, for cone angles 30°, 45° and 60°; respectively.

In order to investigate the effect of the inserts of perforated nozzle-turbulator on the heat transfer process and pressure drop, and to develop an empirical equation for the best tested augmentation procedures; an appropriate testing unit was designed and constructed to investigate the turbulent forced convection heat transfer augmentation in a uniformly heated horizontal circular test tube for a specific range of Reynolds number from (6000 to 13500). The heat transfer augmentation has been investigated by using perforated nozzle-turbulator. The insertion of the nozzle-turbulators were made thoroughly inside the heating section by press with pitch ratio of (PR = l/D =5).

Experimental Setup

A schematic layout of the open test loop is shown in Fig. (1). The loop consisted of 764W blower, orifice meter to measure the flow rate, and the heat transfer test section. The Aluminum test tube has a length of 1350mm and 45mm inner diameter with 2.5mm 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. The electrical output power was controlled by a variac transformer to obtain the constant heat flux along the entire length of the test tube. A multilayer insulation was used on the outer surface of the test tube to reduce the convective heat loss to the surroundings. A multichannel temperature measurement was submitted by using a selector switch and digital thermometer connected to eighteen thermocouples tapped on the local wall of the test tube. The inlet and the outlet bulk air temperature of the air was mustered by using a thermocouple placed at the inlet of the test tube and two thermocouples placed at the outlet of test tube, respectively. Figs.(2, 3, 4 and 5) shows the nozzle-turbulators used in the present experiments. The nozzle-turbulators were constructed from Aluminum with the length of a=45 mm and with ends diameters of (D=45 mm and d=22.5 mm). For all types of perforated Nozzle-turbulators, each Nozzle-turbulator was perforated with four holes. All the holes (regardless of its shape) were fabricated to have the same area of 100 mm^2.

Data Reduction

Air was used as the working fluid. The heat transfer rate (Q_a) of air was calculated as:

$$Q_a = \dot{m}Cp_a\left(T_{exit} - T_{inlet}\right) \quad \text{... (1)}$$

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

$$Q_{conv} = Q_T + Q_{cond} - Q_{loss} \quad \text{... (2)}$$

Where (Q_T) is the heat supplied by the electrical heater which was calculated from:

$$Q_T = VI \quad \text{... (3)}$$

(Q_{cond}) represents the heat loss by conduction which was calculated as:

$$Q_{cond} = \frac{\overline{T}_w - T_{ins}}{\overline{R}_T} \quad \text{... (4)}$$

Where (T_{ins}) is the average insulation outer surface temperature and ($\overline{T}_w$) is the average test tube surface temperature which was calculated as:

$$\overline{T}_w = \frac{\Sigma T_w}{18} \quad \text{... (5)}$$

Taking (T_w) as the local test tube surface temperature, while ($\overline{R}_T$) is the total thermal resistance of the multilayer insulation. The heat gained by air with convection (Q_g) can be calculated as follows:

$$Q_g = \frac{Q_a + Q_{conv}}{2} \quad \text{... (6)}$$

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

$$Q_g = \overline{h}A_s(\overline{T}_w - T_b)$$

$$\overline{h} = \frac{Q_g}{A_s(\overline{T}_w - T_b)} \quad \text{... (7)}$$

Where

$$T_b = \frac{T_{exit} + T_{inlet}}{2} \quad \text{... (8)}$$

The average Nusselt number ($\overline{Nu_d}$) can be calculated from the following equation:

$$\overline{Nu}_d = \frac{\bar{h}.D_h}{k_a} \qquad ...(9)$$

Reynolds number (Re_d) was calculated from the equation:

$$\mathrm{Re}_d = \frac{\rho_a U D_h}{\mu} \qquad ...(10)$$

The friction factor (*f*) was calculated as follows:

$$f = \frac{\Delta P_t}{\left(\frac{L}{D}\right)\left(\frac{\rho U^2}{2}\right)} \qquad ...(11)$$

where (U) was the mean air velocity in the tube.

All of the thermophysical properties of the air were determined at the overall bulk temperature from Eq.(8).

Results and Discussion

The experimental results on heat transfer characteristics in a circular tube with effects of perforated nozzle-turbulators are presented. The results of the plain tube are compared with the past correlations as shown in Fig.(6) and Fig.(7) which shows the variation of average Nusselt number versus Reynolds number and the variation of friction factor versus Reynolds number with deviation of 8.4% and 9.3% respectively, as can be seen from this figures that the agrees well with the correlations of Kongkait Paiboon [5].

The influence of using perforated divergent Nozzle-turbulator on the average Nusselt number is illustrated in Fig.(8). The average Nusselt number for the tube fitted with the perforated nozzle-turbulator (regardless of the holes shape) is 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 average Nusselt number. Actually the average Nusselt number for triangular holes, circular 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 which has a direct impact on the average Nusselt number. This behavior leads to the fact that the triangle hole creates a higher disruption to the flow than that created by the circular hole and the square hole. On the other hand the results reveal that using the traditional nozzle-turbulator gives a higher average Nusselt number 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 friction factor of the plain tube is shown in Fig.(9). This figure reveals that there is a significant increase in the friction factor 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 excess 9% but in fact they lead to a much bigger difference between their 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 same Reynolds number. This can be attributed to the fact that the perforation with any shape reduces the obstructing of the nozzle-turbulator to the air flow which leads certainly to reduce the friction factor across the plain tube. By analyzing the data obtained from using all the augmentation devices to develop the experimental correlations in terms of average Nusselt number and friction factor, with using the least square method in the (STATISTICA 6.0) program to evaluate the constant correlations values for the following equations:

For the divergent Nozzle-turbulator arrangement:

$$\overline{Nu}_d = 0.139594\,\mathrm{Re}_d^{\,0.731}\,\mathrm{Pr}^{0.4} \qquad \text{... (12)}$$

$$f = 6809.99\,\mathrm{Re}_d^{\,-0.999} \qquad \text{... (13)}$$

For the perforated divergent Nozzle-turbulator:

$$\overline{Nu}_d = 0.194917\,\mathrm{Re}_d^{\,0.6746}\,\mathrm{Pr}^{0.4}\,Z^{0.1111} \qquad \text{... (14)}$$

$$f = 118.2535\,\mathrm{Re}_d^{\,-0.6384}\,Z^{0.0247} \qquad \text{... (15)}$$

Where:
Z = the ratio of hole perimeter to the nozzle-turbulator length.

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 (η) at constant pumping power witch is:

$$\eta = \frac{\left(\overline{Nu}_{dE} / \overline{Nu}_{dt}\right)}{\left(f_E / f_t\right)^{\frac{1}{3}}} \qquad \text{... (12)}$$

Fig.(10) illustrates that for the perforated nozzle-turbulator the triangle holes provide the highest average value of thermal performance factor of 1.734, while the circle holes and the square holes provided 1.698 and 1.631 respectively. Also, all shapes of perforation provide thermal performance factor higher than that of the divergent nozzle-turbulator.

CONCLUSIONS

1. Nozzle-turbulators with or without perforation could be inserted inside the flow tube to enhance the heat transfer rate, this is because of its influence in disrupting the boundary layer and the reverse flow that enhance the convection heat transfer process by increasing the average Nusselt number, actually 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, Reducing cross sectional area affects directly and severely the mean velocity and temperature gradient that produce greater heat fluxes and momentum because of the larger effective potential force,
2. Inserting Nozzle-turbulators with or without perforation increase pressure drop because of the secondary flow obtained from the interaction of the pressure forces with the internal forces of the boundary layer.
3. Correlations for Nusselt number and friction factor based on the present experimental data are introduced for practical use.
4. The triangle hole shape perforation provided a higher heat transfer rates than those of the circular and square hole shapes perforation, 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 which has a direct impact on the average Nusselt number which leads to the fact that the triangle hole creates a higher disruption to the flow than that created by the circular hole and the squire hole.
5. 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 reduces the obstructing of the nozzle-turbulator to the air flow which leads certainly to reduce the friction factor across the plain tube
6. Triangular perforation gives higher thermal performance factor than non perforated nozzle-turbulators, this is because it increases the average Nusselt number more than the increasing in the friction factor.
7. In all cases, the heat transfer rates increase at the expense of high friction losses that is attributed to the high viscous losses near the pipe wall and to the forces exerted by the Nozzle-turbulator blocking the flow to cause dissipation of the dynamic pressure of the tested fluid..

Nomenclature

English Symbols		
Symbol	**Description**	**Units**
A	Area	m^2
a	Nozzle-turbulator length	m
c	Side length of the square perforation in the nozzle-turbulator	m
C_p	Specific heat	kJ/kg.K
D	Diameter	m
d	Nozzle-turbulator small end diameter	m
D_h	Hydraulic diameter of the test tube	m
f	Friction factor	-
h	Hight length of the triangle perforation in the nozzle-turbulator	m
$\bar{h}$	Average heat transfer coefficient	$W/m^2.K$
I	Current	A
j	Base length of the triangle perforation in the nozzle-turbulator	m
k	Thermal conductivity	W/m.K
L	Test tube length = wire-coil length	m
l	Pitch length between nozzle-turbulator	m
m'	Mass flow rate of inlet air	kg/s
$\bar{N}u_d$	Average Nusselt number $\overline{Nu}_d = \dfrac{\bar{h}.D_h}{k_a}$	-
Pr	Prandtl number $\mathrm{Pr} = \dfrac{\mathrm{Cp}.\mu}{k}$	-
Q	Volumetric flow rate	m^3/s
$\bar{R}$	Thermal resistance	$m^2.K/W$
Re_d	Reynolds number $\mathrm{Re}_d = \dfrac{\rho_a U D_h}{\mu}$	-
t	Diameter of the circular perforation in the nozzle-turbulator	m
T	Temperature	K
T_{exit}	The air temperature at the exit of the test tube	K
T_{inlet}	The air temperature at the entrance of the test tube	K
$\bar{T}_w$	Average test tube surface temperature	K
U	Air mean axial velocity	m/s
V	Voltage	V
Z	the ratio of hole perimeter to the nozzle-turbulator length	-

Greek Symbols

Symbol	Description	Units
ΔP	pressure drop	Pa
μ	Viscosity of air	kg/s. m
η	Thermal performance factor	-
ρ	Density	kg/m^3

Subscripts

Symbol	Description
a	Air
b	Bulk air
cond	Conduction heat transfer
conv	Convection heat transfer
E	Enhancement device
g	Heat gained by air with convection
ins	Insulation outer surface
loss	Loss axially from the test tube
s	Test tube surface
t	Plain test tube
T	Total
w	Test tube wall

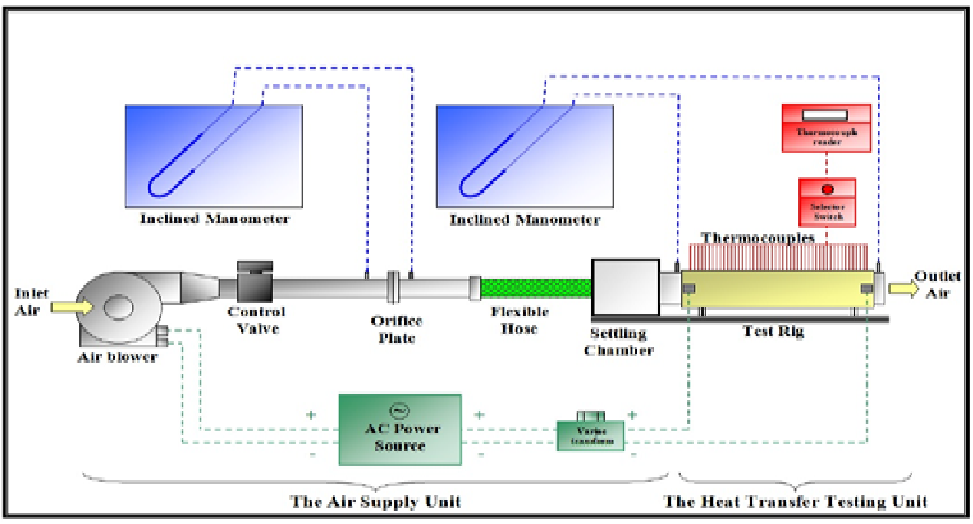

Figure.(1) Schematic diagram of the experimental setup.

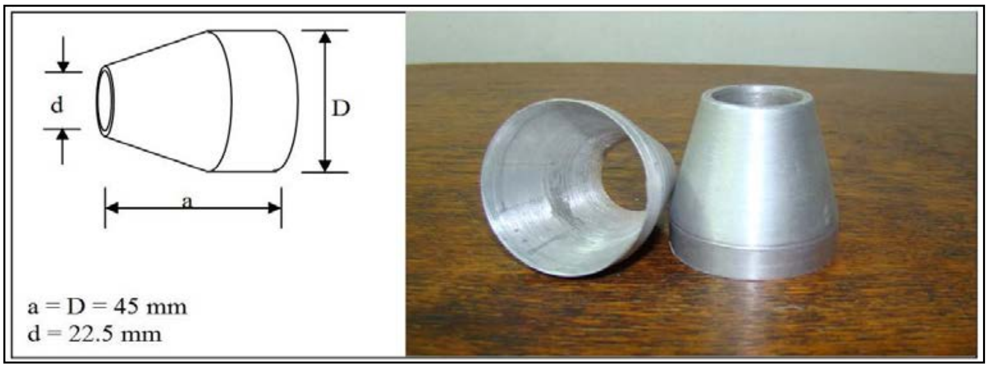

Figure. (2) Dimensions of the nozzle-turbulators.

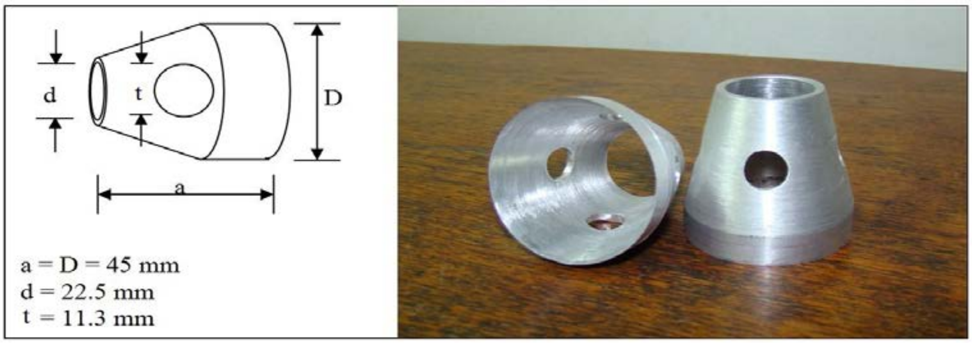

Figure. (3) Circle shape perforation Nozzle-turbulators

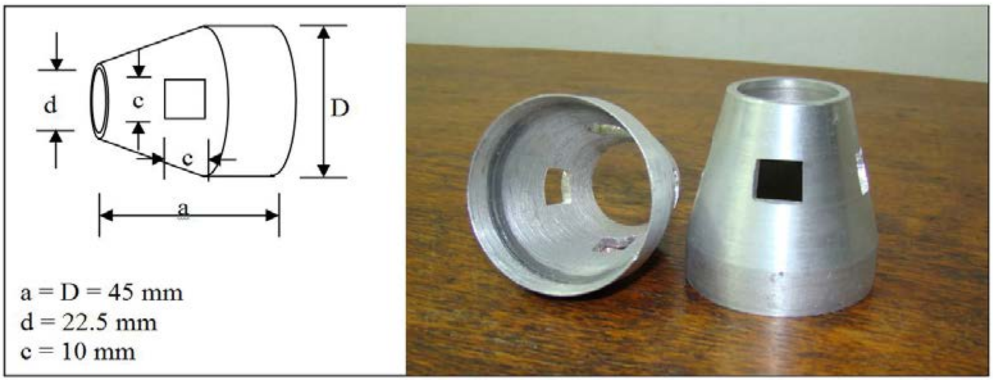

Figure. (4) Square shape perforation Nozzle-turbulators

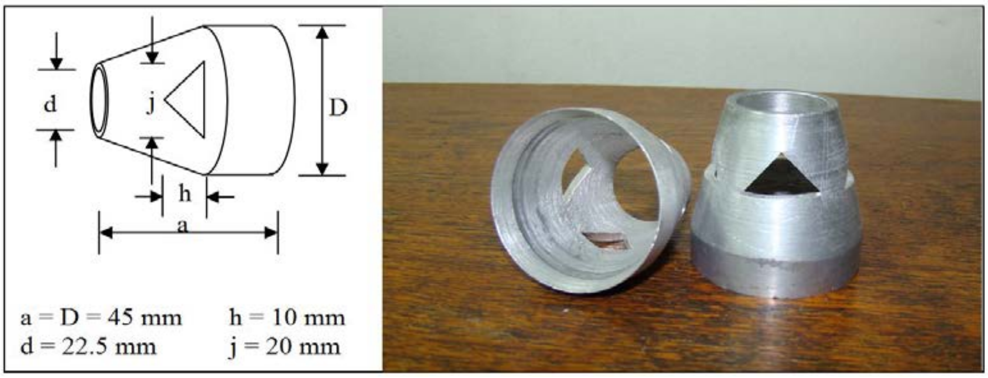

Figure. (5) Triangle shape perforation Nozzle-turbulators

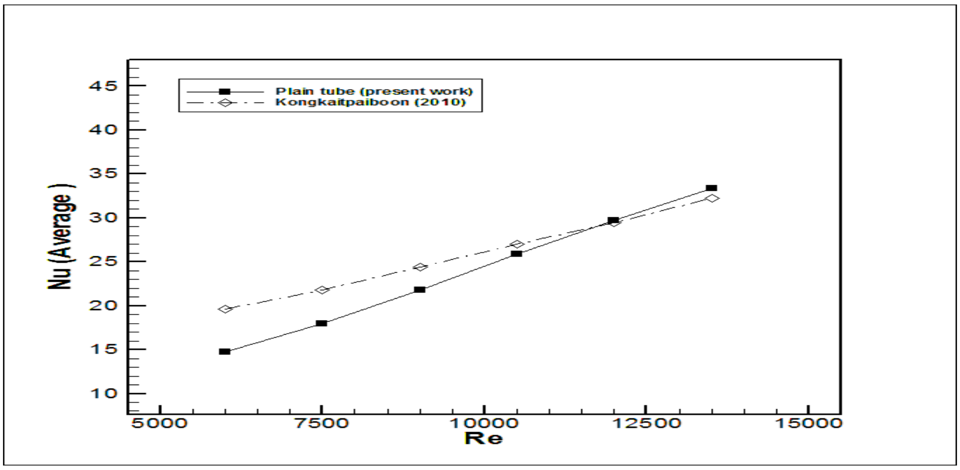

Figure.(6) Verification of Nusselt number of plain tube

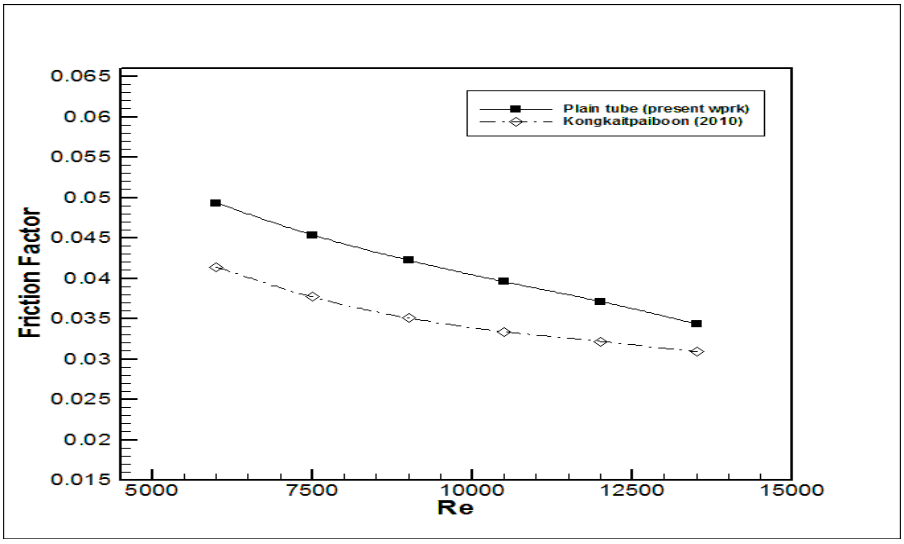

Figure.(7) Verification of Friction factor of plain tube

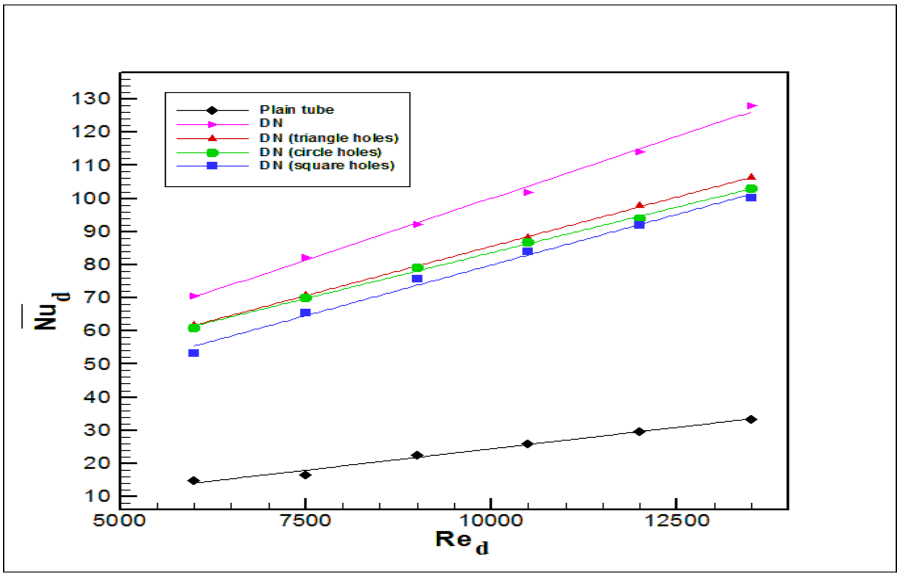

Figure.(8) Average Nusselt number vs. Reynolds number

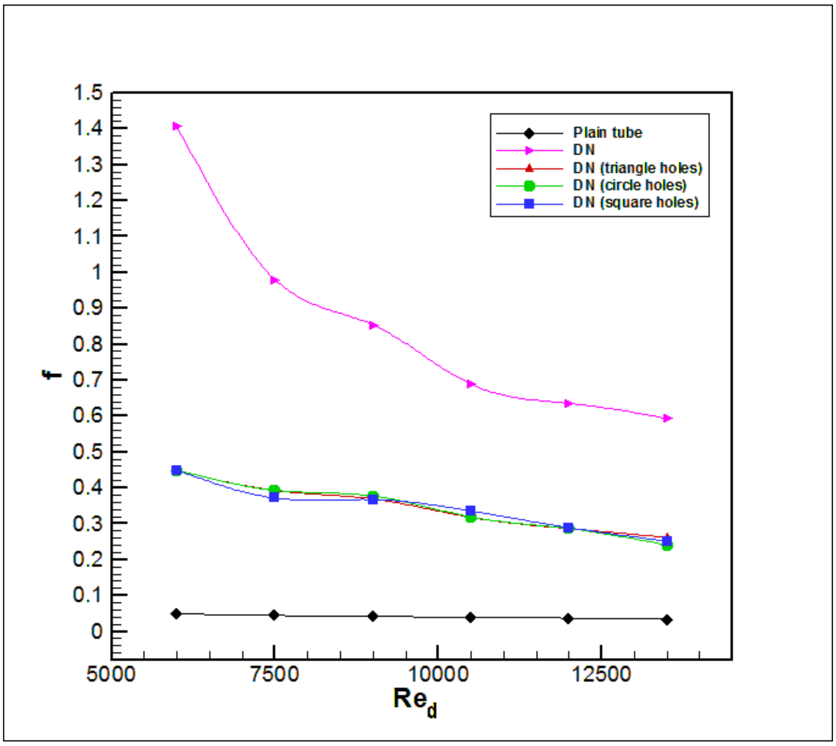

Figure.(9) Friction factor vs. Reynolds number

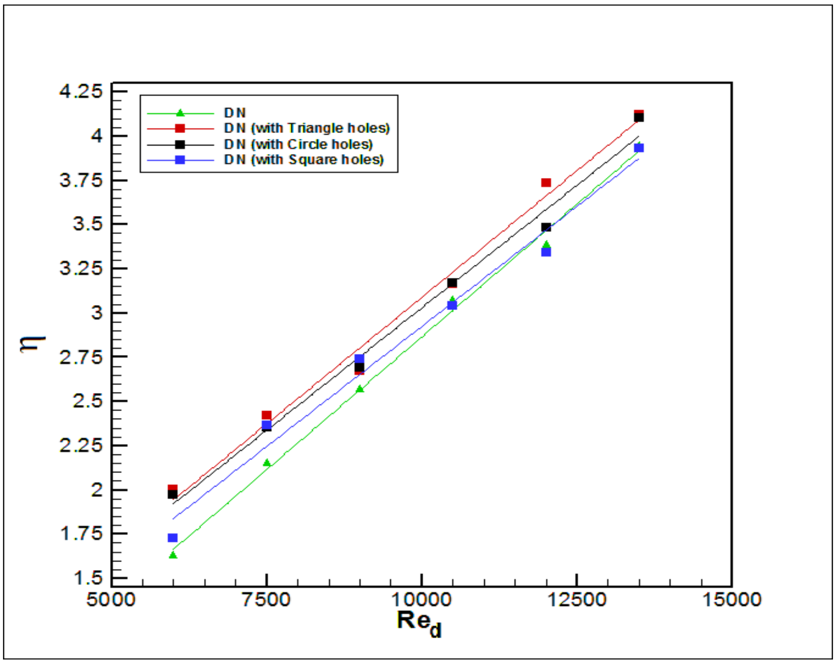

Figure.(10) Thermal performance factor vs. Reynolds numbe

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