Heat Transfer Augmentation in Round Tube with Diamond-shaped Baffle Inserts

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Abstract. Influences of diamond-shaped baffle (DSB) insertion in a heat exchanger tube on its thermal performance are experimentally presented. The DSB elements were repeatedly mounted into the tube using two straight small rods to connect the DSBs together to produce the longitudinal vortex flows inside. The experiments were carried out in a constant surface heat-fluxed tube equipped with DSBs. The DSBs were placed periodically with an attack angle of 45° and a pitch spacing of three times of tube diameter (PR= P/D=3) while four different DSB heights called blockage ratios (BR= b/D= 0.1, 0.2, 0.3 and 0.4) were offered. Air was used as the test fluid flowed into the tube to yield the Reynolds number (Re) in a range of 4190–26,000. The heat transfer rate and the pressure drop of the current work were presented in terms of Nusselt number (Nu) and friction factor (f), respectively. The experimental results showed that the DSB provides the increase in the heat transfer around 3.53–4.16 times higher than the smooth tube while friction factor increases around 16.68–29.52 times. To evaluate the gain of using the DSB, the thermal performance factor (TEF) is determined in a range of 1.25–1.55 whereas the highest TEF is found at BR= 0.1.

1. Introduction
The important part of thermal energy equipment, such as air conditioner, refrigerators, etc., is the heat exchanger, which is used to exchange thermal energy from warmer medium to colder one. Many researches were focused on the ways to obtain higher thermal performance of the heat exchangers. Several techniques have been used for heat transfer enhancement in order to meet the energy costs and to obtain the compact size. The tubular heat exchanger is widely used in most of heat exchangers [1–6]. In enhancing the heat transfer, many different types of turbulators have been developed to produce swirl/vortex flows inside by inserting these devices into the round duct/tube. All kinds of the
turbulators have been used to improve the rate of heat transfer in the round tube such as twisted-tape [7–11], coiled square-wire [12], snail entry and coiled wire [13], combined non-uniform wire coil and twisted tape [14], equilateral triangle cross-sectioned coiled wire [15], artificial roughness surface and wire coils [16]. The modified twisted tapes were performed to yield the thermal performance improvement higher than the typical twisted tape around 10–40%.

Apart from the turbulators mentioned earlier, there were conical rings, cross-sectional rings, V-nozzles, and conical nozzles. Promvonge [17] investigated the effect of three different conical ring diameters and its arrangement on heat transfer behaviors in a round tube. Ozeyhan et al. [18] studied the heat transfer enhancement in a tube with five different spacing circular cross-sectional rings. Thermal characteristics of a heat exchanger tube contained with V-nozzle turbulators was reported by Eiamsaard and Promvonge [19]. Again, Promvonge and Eiamsaard [20] examined the effect of three different conical-nozzle pitches and nozzle arrangements on flow friction and thermal behaviors. Also, thermo-hydraulic characteristics in a constant surface heat-flux tube with V-ring inserts were investigated by Chintuaythong et al. [21]. They indicated that the V-ring performed the best among the other published rings.

The use of punched delta-wing/winglet tape inserts [22–24] is considered as another kind of turbulators employed to enhance the heat transfer in a round duct/tube with the slight increase in friction factor. Also, several investigations have been carried out to examine the effect of delta-wing/winglet tapes on heat transfer and pressure drop. Skullong et al. [25] studied experimentally the thermal and flow friction characteristics in a tube inserted with staggered-winglet perforated-tapes. Eiamsaard and Promvonge [26] investigated the influence of double-sided delta-wing tape insert with alternate-axes on heat transfer characteristics in a heat exchanger tube. Influences of quadruple perforated-delta-winglet inserts on thermal behaviors in a round tube with different winglet porosities were presented by Skullong et al. [27].

As cited above, most of the performance improvements in many heat exchanger systems have been made by considering the different geometries and arrangements of swirl/vortex flow devices with different their height/widths, pitches and angles. Convection heat behaviors for turbulent flow through the oblique diamond-shaped baffles have never been reported. The aim of the current work is to study the heat transfer as well as the performance augmentation in a tube equipped with diamond-shaped baffles. The experiment was carried out with air as the working fluid in a turbulent regime, Reynolds number from about 4190 to 26,000.

2. Experimental setup

A schematic diagram of the apparatus with the basic components is presented in figure 1. The experimental facility consisted of an inlet section, test section, air supply system (high-pressure blower) and heating arrangement. In the experimental apparatus, the copper tube having inner diameter (D) of 50.8 mm and 2 mm thickness (t) was 3000 mm long included the test section length (L) of 1200 mm while the inlet section was used to get a fully developed flow before entering the test section. To achieve a constant heat-fluxed tube, an electrical wire was employed and continually wound round the test tube and a variac transformer (AC power supply) was used to control the output power of the electric heater as desired. The outermost of the test section wrapped by flexible insulations was done to decrease the convection loss to air in the surrounding.

A blower with 1.9-kW motor power was used to supply the room air to the system by flowing through the orifice-plate for measuring the volumetric airflow rate before entering the test tube where the pressure drop measurement of the orifice plate was done via an inclined/U-tube manometer. The rates of airflow through the test section were varied using an inverter to control the motor speed of the blower. Two Resistance Temperature Detectors (RTD, Pt-100) were employed to measure the temperatures of air at the outlet and inlet while 24 type-T thermocouples equally located on the top and side walls of the test tube were used for measuring the surface temperatures along the test section. When outlet and surface temperature values were not changed for about half an hour indicating a steady state condition of the system, all values of temperatures were recorded via a data acquisition
device (Fluke 2680A). Also, the pressure drop measurements of the test tube was conducted using a Dwyer Instruments, 475-1-FM 475 mark III digital manometer. The uncertainties in velocity, pressure and temperature measurements were, respectively, evaluated to be under ±5%, ±3% and ±0.4%. The maximum measurement uncertainties in various instruments were within ±1.4% for the orifice plate and ±1.2% for the digital manometer.

**Figure 1.** Experimental set-up.

The detail of 45° DSB inserted into the test tube is presented in figure 2. The DSB in the current work was made of an aluminum sheet with 0.3 mm thickness (t). The DSB was cut, drilled to form the diamond-shaped hole and placed repeatedly in the tube with four baffle blockage ratios (BR=b/D= 0.1, 0.2, 0.3 and 0.4) at a single relative baffle pitch or pitch ratios (PR=P/D=3). The DSB elements were linked to each other using two straight small-rods on two edges of the baffles. The interfaces between the DSB and the rod were tightly attached using superglue.

**Figure 2.** Detail and arrangement of 45° DSBs in test tube.

### 3. Data reduction

The purpose of this experiment is to examine the flow resistance and thermohydraulic behaviors in a heat exchanger tube equipped with DSBs. The parameters of interest are Nusselt number (Nu), friction factor (f) and Reynolds number (Re). The Re can be evaluated by:

\[
Re = \frac{\mu U}{\rho} \quad (1)
\]

While pressure loss (ΔP) of the test tube is used to estimated f,

\[
f = \frac{2}{(L/D)} \frac{\Delta P}{\rho U^2} \quad (2)
\]

where \(U, \mu\) and \(\rho\) are, respectively, the average velocity, dynamic viscosity and density of air.
The heat transfer inside the tube is equal to the convection heat of air in the current work. Therefore, the average convection coefficient \( h \) is estimated by
\[
h = mC_p,\left(T_0 - T_i\right)/A\left(\tilde{T}_a - T_b\right)
\]

in which
\[
\tilde{T}_a = \sum T_w / 24
\]

The Nusselt number (Nu) of the system is given as
\[
Nu = \frac{hD}{k}
\]

Thermal enhancement factor (TEF) at identical pumping power, is defined by Webb [28] as
\[
TEF = \frac{h}{h_{ij}} = \left(\frac{Nu}{Nu_{ij}}\right)^{1/3} \left(\frac{f}{f_{ij}}\right)^{-1/3}
\]

where \( h_0 \) and \( h \) stand for convection coefficients for smooth and inserted tubes, respectively.

4. Results and discussion

4.1. Verification of plain tube

The Nu and \( f \) values of the current smooth/plain tube were compared to those of the correlations of Gnielinski and Petukhov [29].

Gnielinski’s correlation,
\[
Nu = \frac{\left(\frac{f}{8}\right)(Re-1000)Pr}{1 + 12.7\left(\frac{f}{8}\right)^{1/3}\left[Pr^{1/7} - 1\right]} \quad \text{for} \ 3000 \ < \ Re \ < 5 \times 10^6
\]

Petukhov’s correlation
\[
f = (0.79\ln Re - 1.64)^2 \quad \text{for} \ 3000 \ < \ Re \ < 5 \times 10^6
\]

The validation of both the results is shown in figure 3a for Nu and 3b for \( f \). A reasonably good agreement between the experimental and correlation data can be observed while the mean discrepancy is within 5.3% for Nu and 6.7% for \( f \).

4.2. Influence of BR on heat transfer

The heat transfer rate in a circular tube with 45° DSB insert against Re values is clearly depicted in figure 4a. Obviously, Nu of the DSB is much higher than that of the plain/smooth tube alone and
displays the rising tendency with the increase in Re. This is because the presence of DSBs can produce longitudinal vortex flows inside apart from promoting the turbulence intensity. The DSB with larger BR value can create higher flow obstruction leading to stronger vortex flow strength than the one with smaller value.

The variation of the ratio of augmented Nu to Nu of plain tube alone (called “Nusselt number ratio”, Nu/Nu₀) with Re is displayed in figure 4b. It can be seen that Nu/Nu₀ shows the uptrend with the increase in BR owing to stronger degree of turbulence level. The mean values of Nu/Nu₀ for BR=0.4, 0.3, 0.2 and 0.1 are, respectively, about 4.16, 3.93, 3.71 and 3.53 times. This implies that the use of larger BR gives rise to stronger circulation of flow apart from higher turbulence intensity and thermal boundary layer destruction resulting in the considerable increase in the rate of heat transfer.

![Figure 4. Nu (a) and Nu/Nu₀ (b) versus Re for DSBs.](image)

4.3. Influence of BR on friction loss

The presence of DSBs leads to the increase in pressure drop which is presented in the form of $f$ and friction factor ratio, $(f/f₀)$. Figure 5a and 5b depicts, respectively, $f$ and $f/f₀$ plotted against Re for different BRs. It is found in figure 5a that $f$ for DSBs is considerably higher than the smooth/plain tube alone and it shows the slightly decreasing tendency with rising Re. $f$ at BR = 0.4 is much higher than that at smaller BR. This is because of higher flow blockage, the dissipation of dynamic pressure of the fluid due to larger surface area and the act caused by the vortex flow.

![Figure 5. $f$ (a) and $(f/f₀)$ (b) against Re for DSBs.](image)
In figure 5b, it is visible that \( f/f_0 \) tends to rise with increasing Re. The use of DSBs yields the substantial increase in \( f/f_0 \) with raising Re. \( f/f_0 \) for BR = 0.4 is much higher than that for smaller BR values at the same operating condition. It is worth noting that \( f/f_0 \) increases with the increase in BR. The mean values of \( f/f_0 \) at BR = 0.4, 0.3, 0.2 and 0.1 are found to be about 29.52, 24.89, 20.37 and 16.68 times, respectively.

4.4. Thermal performance

The potential of using DSBs in real practical is estimated by a performance indicator, namely, thermal enhancement factor (TEF) as defined in equation (6) which is plotted against Re as depicted in figure 6. In the figure, it can be noted that all of the TEF values are higher than unity. This means that employing the DSBs provides the merit over the smooth tube. For the DSBs, TEF shows the decreasing trend with the increment of Re and BR where its peak around 1.54 is seen at BR= 0.1. TEF values at BR = 0.1, 0.2, 0.3 and 0.4 are, respectively, around 1.29–1.55, 1.27–1.52, 1.26–1.51 and 1.25–1.50.

![Figure 6. Effect of BR on TEF for DSBs.](image)

5. Conclusion

An experimental investigation on thermal behaviours in a constant heat fluxed tube equipped with 45° DSBs for various BRs in turbulent flow, Re = 4190–26,000 has been carried out. The heat transfer rate in the inserted tube is enhanced at about 3.53–4.16 times over the plain/smooth tube alone while the pressure drop penalty of the tube in the form of \( f \) is increased around 16.68–29.52 times. Furthermore, the DSB at BR=0.1 provides the maximum TEF around 1.55, indicating that for practical use, this optimum value is considered as a promising device for energy saving.

References

[1] Promvonge P and Eiamsa-ard S 2007 Int. Commun. Heat Mass Transf. 34 72-82
[2] Thianpong C, Eiamsa-ard P, Promvonge P and Eiamsa-ard S 2012 Energy Procedia 14 1117-23
[3] García A, Solano JP, Vicente PG, Viedma A, App. Therm. Eng. 35 196-201.
[4] Bhuiya MMK, Chowdhury MSU, Saha M, and Islam MT 2013 Int. Commun. Heat Mass Transf. 46 49-57
[5] Nanan K, Thianpong C, Promvonge P and Eiamsa-ard S 2014 Int. Commun. Heat Mass Transf. 52 106-12
[6] Gautam A, Pandey L and Singh S 2018 Heat Mass Transf. 54 2009-21
[7] Chang SW Yang TL and Liou JS 2007 Exp. Therm. Flu. Sci. 32 489-501.
[8] Eiamsa-ard S, Thianpong C, Eiamsa-ard P and Promvonge P 2009 Int. Commun. Heat Mass Transf. 36 365-71
[9] Eiamsa-ard S and Promvonge P 2010 App. Therm. Eng. 30 1673-80
[10] Eiamsa-ard S, Yongsiri K, Nanang K and Thianpong C 2010 Chem. Eng. and Process. 60 58-65
[11] Bas H and Ozceyhan V 2012 Exp. Therm. and Flu. Sci. 41 51-8
[12] Promvonge P 2008 Energy Convers. Manag. 49 980-7
[13] Promvonge P 2008 Int. Commun. Heat Mass Transf. 35 623-9
[14] Eiamsa-ard S, Nivesrangsan P, Chokphoemphun S and Promvonge P 2010 Int. Commun. Heat Mass Transf. 37 850-6
[15] Gunes S, Ozceyhan V and Buyukalaca O 2010 Exp. Therm. Flu. Sci. 34 684-91
[16] Garcia A, Solano JP, Vicente PG and Viedma A 2012 Appl. Therm. Eng. 35 196-201
[17] Promvonge P 2008 Energ. Convers. Manag. 49 8-15
[18] Ozceyhan V, Gunes S, Buyukalaca O and Altuntop N 2008 Appl. Energy 85 988-1001
[19] Eiamsa-ard S and Promvonge P 2006 Int. Commun. Heat Mass Transf. 33 591-600
[20] Promvonge P and Eiamsa-ard S 2007 Int. Commun. Heat and Mass Transfer. 34 72-82
[21] Chingtuyathong W, Promvonge P, Thianpong C and Pimsarn M 2017 Appl. Therm. Eng. 110 1164-71
[22] Skullong S, Promvonge P, Jayranaiwachira N and Thianpong C 2016 Chem. Eng. and Process. 109 164-77
[23] Koolnapadol N, Sripattanapipat S and Skullong S 2016 J. Res. Appl. Mech. Eng. 4 166-74
[24] Skullong S Promvonge P Thianpong C Jayranaiwachira N Pimsarn M 2018 Appl. Therm. Eng. 129 1197-211
[25] Eiamsa-ard S and Promvonge P 2011 Chinese J. Chem. Eng. 19 410-23
[26] Skullong S, Promvonge P, Thianpong C and Pimsarn M 2016 Int. J. Heat Mass Transf. 95 230-42
[27] Skullong S, Promvonge P, Thianpong C and Jayranaiwachira N 2017 Appl. Therm. Eng. 115 229-43
[28] Webb RL 1981 Int. J. Heat Mass Transf. 24 715-26
[29] Incropera FP, Witt PD, Bergman TL and Lavine A 2006 Fundamentals of Heat and Mass Transfer John-Wiley & Sons