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Numerical Investigation on Performance Improvement in Tube Inserted with Delta Winglets on Discrete Right-Angled Triangular Plates

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Abstract: Numerical investigations on flow topology and heat transfer behavior in a round tube heat exchanger inserted with Delta Winglets (*DW*) on Discrete Right-angled Triangular Plates (*DRTP*) are performed. The influences of *DW* heights; b/D or BR = 0.05-0.25 and flow directions; *DW* with V-Downstream (*DWVD*) and *DW* with V-Upstream (*DWVU*), are studied for turbulent flow; Re = 3000-10,000. The finite volume method with SIMPLE algorithm is selected to solve the present problem. The numerical results are presented in terms of flow and heat transfer mechanisms; tangential velocity vector, *TKE* contour, Nu_x contour, temperature contour. The heat transfer rate, pressure loss and thermal performance in the test tube are also analyzed in forms of Nusselt number (Nu), friction factor (f) and Thermal Enhancement Factor (*TEF*), respectively. The results show that the tube inserted with the *DW* on *DRTP* can help to improve the heat transfer rate and thermal performance in the heating section due to the *DW* can generate the vortex flow that disturbs the thermal boundary layer on the tube wall. In range investigates, the maximum heat transfer rate is around 4.5 times above the smooth tube with the *TEF* around 1.9.

Keywords: Delta Winglet, Heat Exchanger, Thermal Performance, Finite Volume Method

Background

In general, the use of the vortex generators or turbulators in heat exchangers helps to improve heat transfer rate and thermal performance due to they can induce the vortex flow that disturbs the thermal boundary layer on the tube wall. Energy saving, cost reduction, reduce the size of heating/cooling section, etc., are benefits when enhancing the thermal performance of the heat exchanger. Many researchers studied on the performance improvement and heat transfer mechanism in the heat exchangers by both experimental and numerical methods. The investigated results are important factors to improve the heat exchangers in various industries.

The winglet (Caliskan, 2014; Gholami *et al*., 2014; He *et al*., 2012; Khoshvaght-Aliabadi *et al*., 2015; Wang *et al*., 2015) is a type of the vortex generator, which always uses to augment heat transfer rate and thermal performance. The researchers found that the winglet has high effectiveness to improve performance of thermal system when compared with the other types of the vortex generators. Min and Zhang (2014) numerically investigated heat transfer enhancement in a membrane channel with rectangular winglet. They concluded that the rectangular winglet in the channel improves the heat transfer rate with moderate pressure loss penalty. Chokphoemphun *et al*. (2015) found that the winglet vortex generators in a circular tube heat exchanger gives higher heat transfer rate around 2.03-2.34 times above the smooth tube. They also summarized that the thermal performance for the winglet vortex generators is higher than the wire coil and twisted tape around 1.35-1.59. Saha *et al*. (2014) selected the winglet to develop the thermal performance of a plate-fin heat exchanger. Li *et al*. (2015) investigated flow and heat transfer in a fin-and-tube heat exchanger inserted with rectangular and delta winglets. They reported that the optimum flow attack angle is around 25° and 45° for rectangular and delta winglets, respectively. Lin *et al*. (2015) presented heat transfer augmentation in a tube bank fin heat exchanger with curved delta winglet. They concluded that the curved delta winglet can reduce wake region and also create secondary flow.

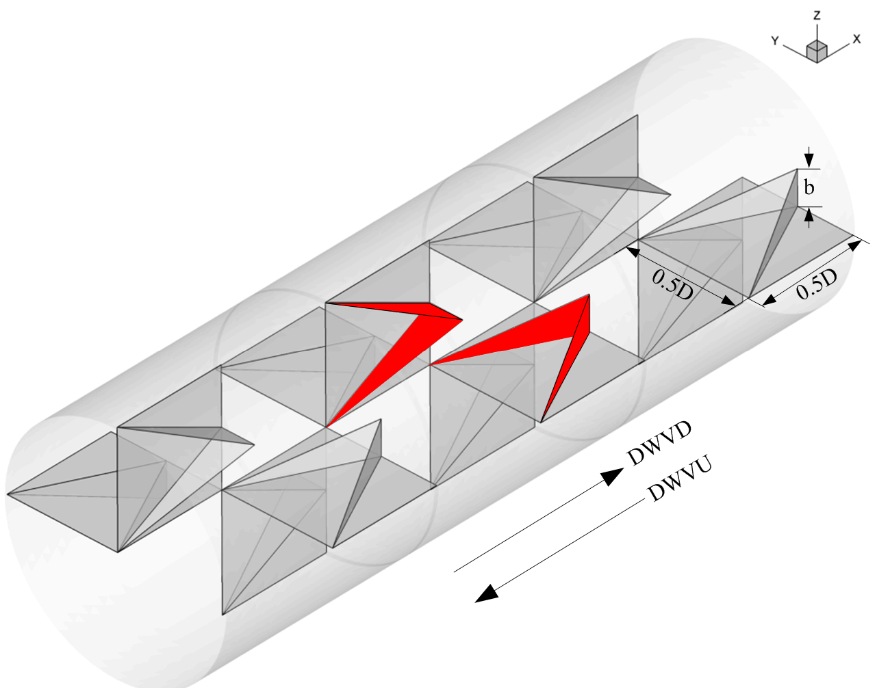
In the present research, the Delta Winglets (*DW*) are selected to enhance the heat transfer rate and thermal performance in the circular tube heat exchanger. The *DW* can help to reduce the pressure loss when compared with the rectangular shape. Moreover, the *DW* is an optimum shape for the round tube (curve tube wall). The *DW*s are placed on the Discrete Right-angled Triangular Plates (*DRTP*) and inserted in the round tube heat exchanger. The *DRTP*s may help to increase the turbulence of the air flow in the test tube. The *DW* heights and arrangements of the vortex generators are investigated for turbulent regime, *Re* = 3000-10,000. The mechanisms and performance evaluations of the heating tube are illustrated. The numerical method is designated to study the current problem. The numerical results on both flow and heat transfer are main factors to help to design the compact heat exchanger and also help to develop the heating and cooling systems in various industries.

The paper is divided as follows:

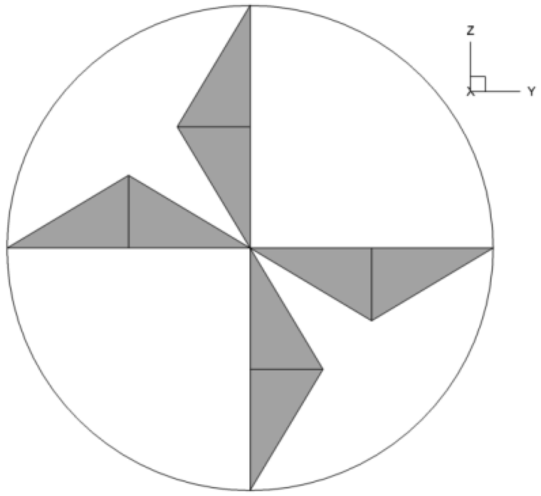
- Background
- Physical model
- Assumption, boundary condition and mathematical foundation
- Numerical validation
- Numerical result
    - Flow and heat transfer mechanisms
    - Performance analysis
- Conclusion

Physical Model

The *DW*s are placed on the *DRTP*s and inserted in the round tube heat exchanger as Fig. 1a, while the tube geometry in cross sectional view is depicted as Fig. 1b. The *DW*s are arranged as V-shaped baffles. The influences of the *DW* heights, *b*, are studied; *b*/*D* or *BR* = 0.05, 0.10, 0.15, 0.20 and 0.25, with the V-Downstream (*DWVD*) and V-Upstream (*DWVU*) arrangements. The length of the *DW* from leading edge to V-tip is fixed around 0.05*D* which equals to length of the *DRTP*. The turbulent flow with the Reynolds number around 3000-10,000 is considered for the current work.



(a)



(b)

Fig. 1. (a) Details of the tube heat exchanger inserted with *DW* on *DRTP* and (b) tube geometry in *y-z* plane

Assumption, Boundary Condition and Mathematical Foundation

The flow and heat transfer in the round tube are steady in three dimensions. The flow is turbulent and incompressible. The radiation heat transfer, body force, natural convection and viscous dissipation are regarded. The fluid properties are assumed as constant at average bulk mean temperature. The periodic condition is used for the entry and outlet of the computational domain. The uniform heat flux around 600 W/m^2 is applied for the tube wall. The *DW* and *DRTP* are set as adiabatic wall condition (insulator). No-slip wall condition is used for all surfaces of the computational domain.

The round tube flow is solved by the continuity equation, Navier-Stokes equation and energy equation. The realizable k-ε model (Launder and Spalding (1974)) is selected for the current numerical solution:

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_j}(\rho k u_j)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] +G_k+G_b-\rho\varepsilon+Y_M+S_k \tag{1}$$

and:

$$\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x_j}(\rho \varepsilon u_j)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] +\rho C_1 S\varepsilon+\rho C_2\frac{\varepsilon^2}{k+\sqrt{\upsilon\varepsilon}}+C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b+S_\varepsilon \tag{2}$$

where:

$$C_1=\max\left[0.43,\frac{\eta}{\eta+5}\right],\eta=S\frac{k}{\varepsilon},S=\sqrt{2S_{ij}S_{ij}} \tag{3}$$

the constant values are as follows:

$$C_{1\varepsilon}=1.44,C_2=1.9,\sigma_k=1.0,\sigma_\varepsilon=1.2 \tag{4}$$

All governing equations are discretized with the SOU numerical scheme decoupling with the SIMPLE algorithm. The finite volume method is selected to solve the present problem. The solutions are set to be converged when the normalized residual are less than 10^{-9} for the energy equation and around 10^{-5} for the other variables.

The Reynolds number, friction factor, local Nusselt numer, average Nusselt number and thermal enhancement factor are presented follows:

$$\mathrm{Re}=\frac{\rho u_0 D}{\mu} \tag{5}$$

ρ, μ and u_0 are density, viscosity and velocity of the air, while D is diameter of the tube heat exchanger:

$$f=\frac{(\Delta P/L)D}{\frac{1}{2}\rho\bar{u}^2} \tag{6}$$

The friction loss is calculated from the pressure drop, ΔP, across the test section, L:

$$Nu_x=\frac{h_x D}{k} \tag{7}$$

The local Nusselt number is measured from the local heat transfer coefficient, h_x:

$$Nu = \frac{1}{A}\int Nu_x dA \tag{8}$$

The thermal enhancement factor is an important factor to evaluate the thermal performance of the tube heat exchanger, which calculated by the enhancements of the heat transfer rate and friction loss at similar pumping power:

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(\frac{Nu}{Nu_0}\right) / \left(\frac{f}{f_0}\right)^{1/3} \tag{9}$$

The Nusselt number and friction factor of the smooth tube are presented as Nu_0 and f_0, respectively.

Numerical Validation

The numerical model is validated with the correlations of the smooth tube on both Nusselt number and friction loss (Patankar (1980)). The maximum deviations for the Nusselt number and friction factor are around ±1.15 and 2.10%, respectively. The verification of grid number is considered by compared difference number of grid cells; 120000, 180000 and 240000. The result shows that the variation of grid cell has no effect for the numerical results on both flow and heat transfer. Therefore, the grid around 120000 is applied for all numerical domains when considering the time for investigation.

Numerical Result

Flow and Heat Transfer Mechanisms

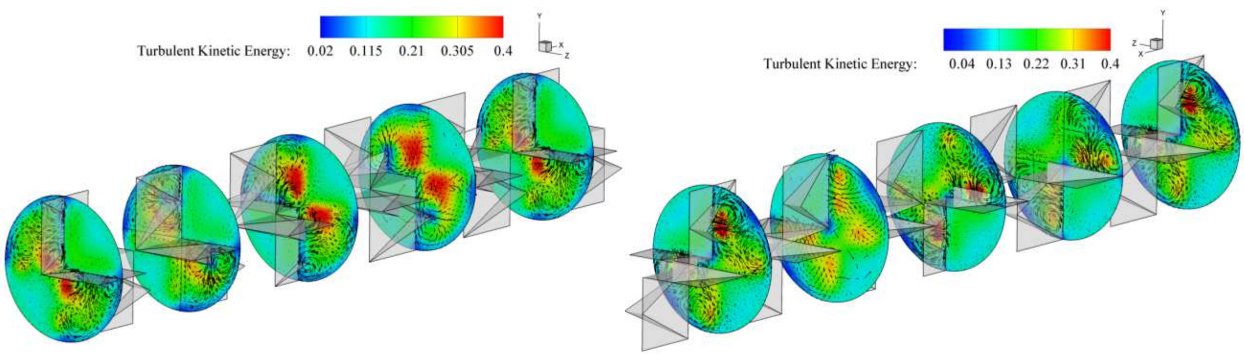
Tangential velocity vector in transverse planes is plotted with Turbulent Kinetic Energy (*TKE*) as Fig. 2a and 2b for *DWVD* and *DWVU*, respectively. The tangential velocity vector and *TKE* are indicators of the vortex flow and turbulence level in the tube heat exchanger, respectively. As the figures, the vortex flow is found along the tested tube for all arrangements. The distribution of *TKE* is found uniformly in all planes. The difference arrangement of the vortex generator leads to the variance of the *TKE* distribution.

The temperature contour in transverse planes for the round tube heat exchanger inserted with the vortex generator is presented as Fig. 3a and 3b for *DWVD* and *DWVU*, respectively. The insertion of the *DW* in the test section provides better mixing of the flow between the core of the tube and near the wall regime. The red layer near the wall performs thinner, while the blue layer spreads out from the center of the tube. The disturbance of the thermal boundary layer near the tube wall is clearly found in all cases.

Figure 4a and 4b report the local Nusselt number distribution on the tube wall for the tube heat exchanger inserted with *DW*s of V-Downstream and V-Upstream, respectively. The use of the *DW* gives higher heat transfer rate than the smooth tube in all cases. Considering at red contour, the *DWVD* provides higher intensity of the Nusselt number contour than the *DWVU*, while the *DWVU* performs larger area of high heat transfer rate than the *DWVD*. The reason of this may be that the *DWVD* and *DWVU* can generate different flow structure. The *DWVD* can create stronger with smaller size of the vortex flow, while the *DWVU* performs the reverse trend.

Performance Analysis

The variations of the Nusselt number ratio (Nu/Nu_0), friction factor ratio (f/f_0) and Thermal Enhancement Factor (*TEF*) are plotted with the Reynolds number as Fig. 5a to 5c, respectively. The relations of the Nu/Nu_0, f/f_0 and *TEF* with *BR* are reported as Fig. 6a to 6c, respectively.



(a) (b)

Fig. 2. *TKE* distributions with tangential velocity vector in transverse planes for round tube inserted with *DW* at *Re* = 4000, *BR* = 0.2 of (a) *DWVD* and (b) *DWVU*

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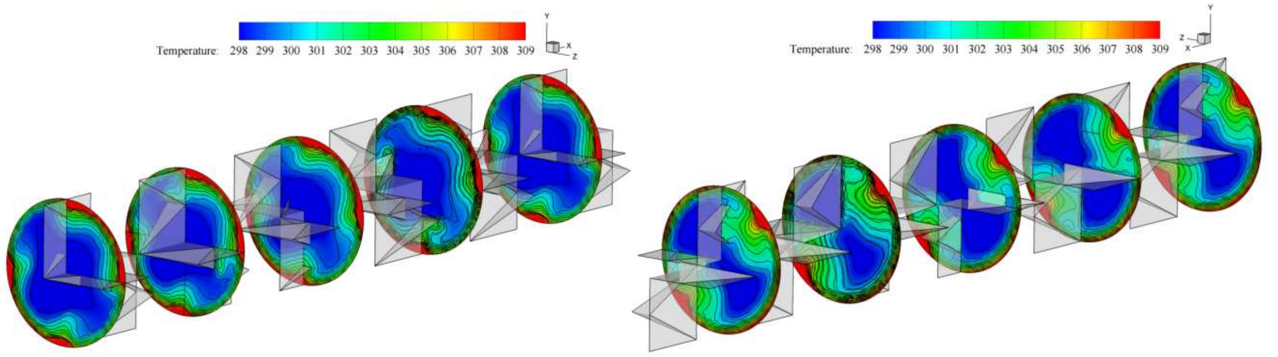
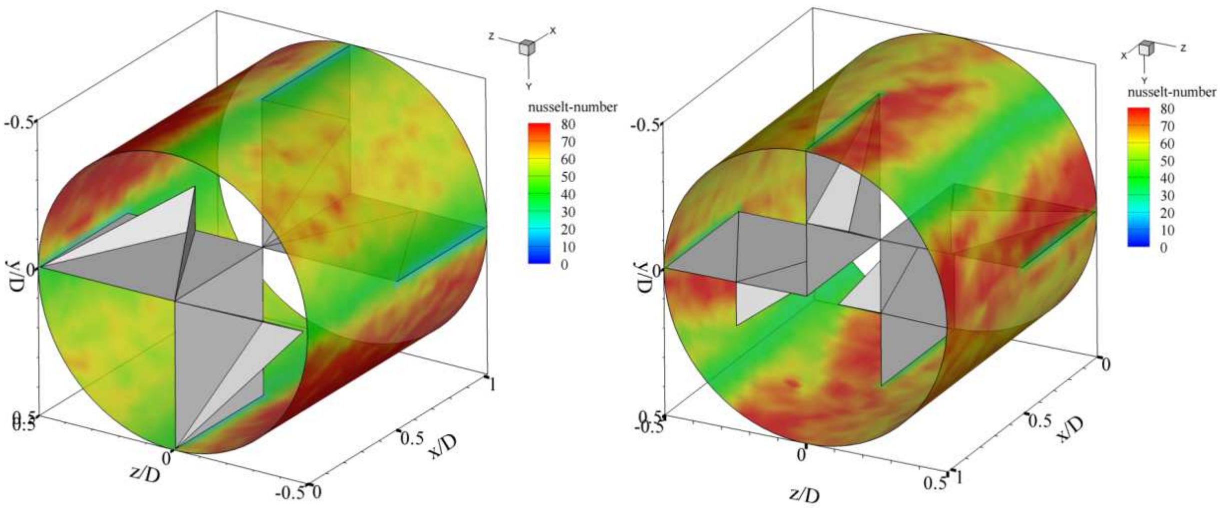


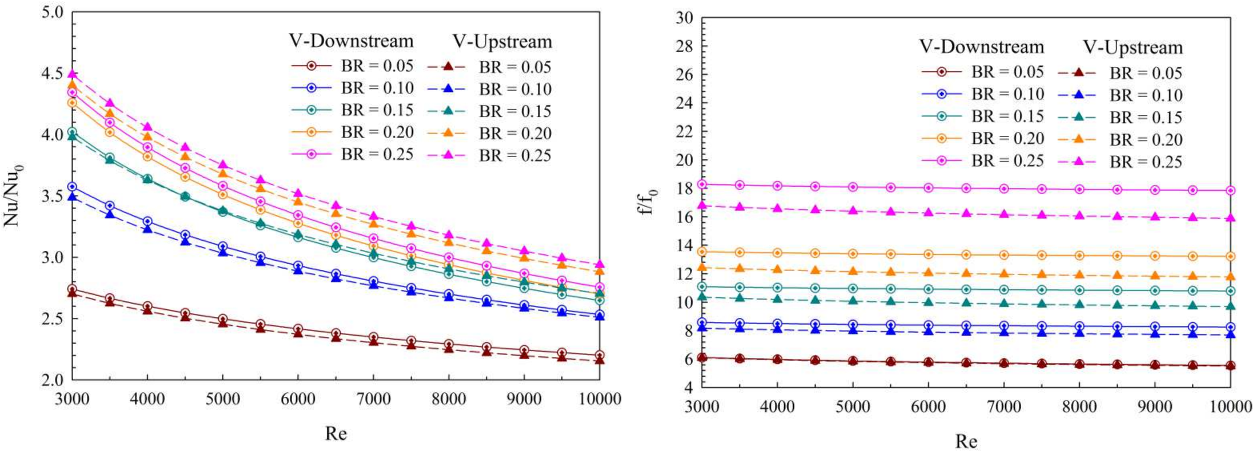
(a) (b)

Fig. 3. Temperature distributions in transverse planes for round tube inserted with *DW* at *Re* = 4000, *BR* = 0.2 of (a) *DWVD* and (b) *DWVU*

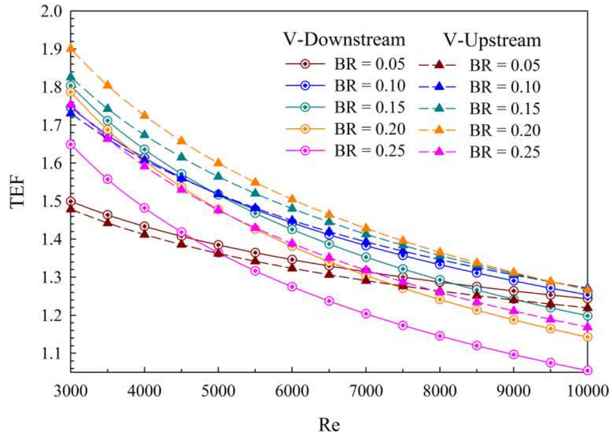


(a) (b)

Fig. 4. Local Nusselt number distributions on tube wall for round tube inserted *DW* at *Re* = 4000, *BR* = 0.2 of (a) *DWVD* and (b) *DWVU*

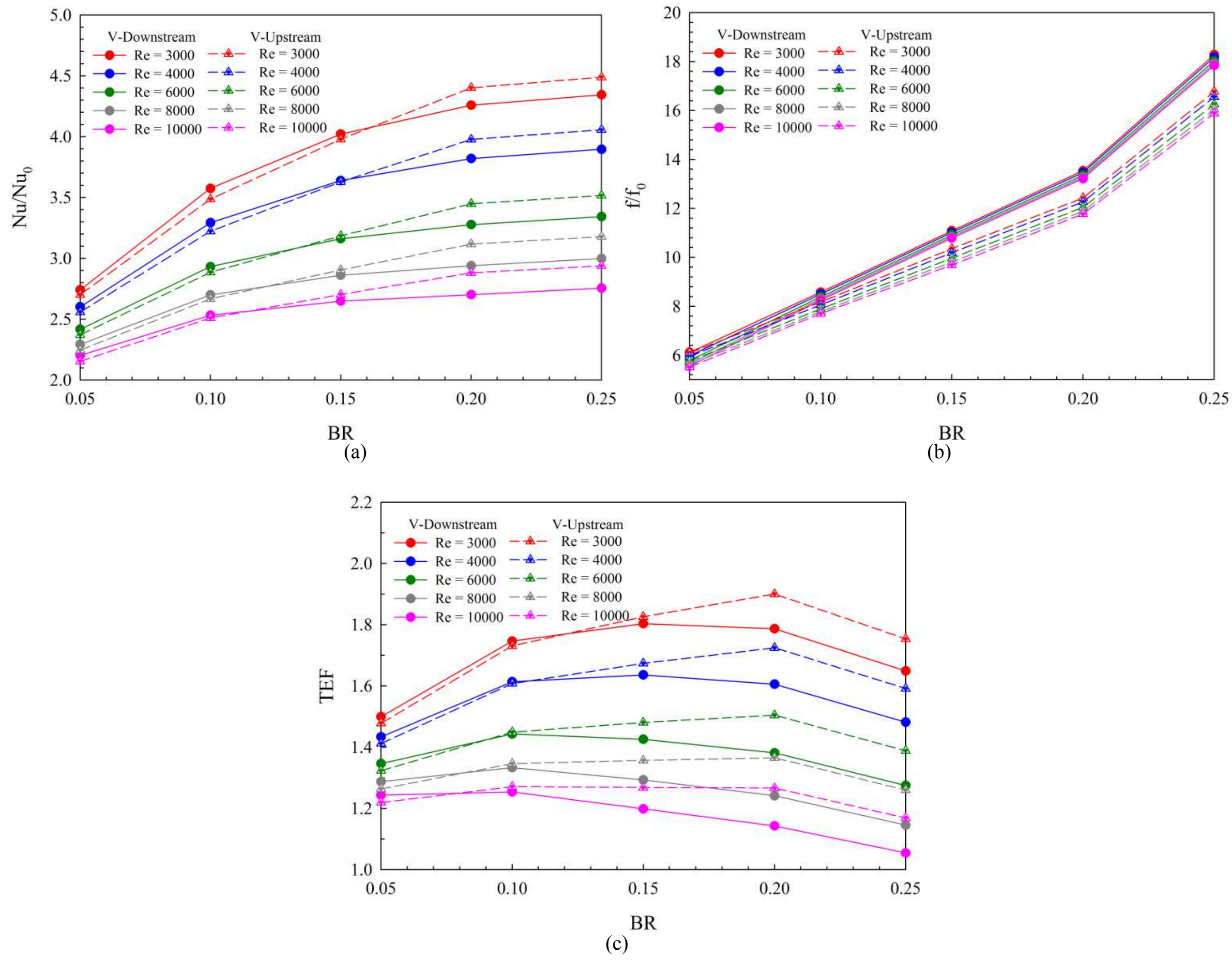


(a) (b)



(c)

Fig. 5. (a) Nu/Nu_0 Vs Re, (b) f/f_0 Vs Re and (c) TEF Vs Re



(c)

Fig. 6. (a) Nu/Nu_0 Vs BR, (b) f/f_0 Vs BR and (c) TEF Vs BR

The Nu/Nu_0 tends to decrease with increasing Re for all BRs and arrangements. The $BR = 0.25$ provides the highest heat transfer rate, while the $BR = 0.05$ gives the opposite result. The $DWVD$ of $BR = 0.05$-0.10 and $BR = 0.15$ in range $3000 \leq Re \leq 4500$ leads to higher heat transfer rate than the $DWVU$. The maximum Nusselt number is around 4.5 for *BR* = 0.25 of *DWVU* at *Re* = 3000.

The insertion of the vortex generator can increase the heat transfer rate in the heating section, but also produce the pressure loss. The *BR* = 0.25 gives the largest pressure loss, while the *BR* = 0.05 offers the reverse results on both *DWVD* and *DWVU* cases. At *BR* = 0.05, the friction loss for both arrangements is found equally. The *DWVD* performs higher friction loss than the *DWVU*, when *BR* > 0.05. The maximum friction loss is around 18 times, while the minimum value is around 6 times above the smooth tube.

In general, *TEF* decreases when increasing the Reynolds number. The thermal performance of the tube heat exchanger inserted with vortex generators is higher than the smooth tube for all cases (*TEF* > 1). The optimum *TEF* is found to be around 1.9 for *BR* = 0.2, *Re* = 3000 of *DWVU*.

Conclusion

Thermal performance analysis in round tube heat exchanger inserted with *DW* is performed. The numerical method is selected to investigate the present problem and to help to describe the mechanisms in the test tube. The effects of height and arrangement for the *DW* are considered for turbulent regime, *Re* = 3000-10,000. The main conclusions are as follows.

The insertion of the *DW* in the tube heat exchanger can improve the thermal performance due to the better mixing of the flow and the disturbance of the thermal boundary layer on the tube wall. The heat transfer rate and thermal performance are found to be higher than the smooth tube; $Nu/Nu_0 > 1$ and *TEF* > 1. However, the pressure loss of the tube is also detected to be higher than the smooth tube, especially at high *BR*.

The *DRTP* increases the turbulence level in the heating section and helps to improve the local Nusselt number distribution on the tube wall.

In range studies, the round tube heat exchanger inserted with *DW* on *DRTP* gives the heat transfer rate and pressure loss around 2.2-4.5 and 6-18 times above the smooth round tube, respectively. The optimum thermal performance is found at *BR* = 0.2 of *DWVU* about 1.9 at the lowest Reynolds number, *Re* = 3000.

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Ethics

This article is original and contains unpublished material. The corresponding author confirms that all of the other authors have read and approved the manuscript and no ethical issues involved.

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Nomenclature

b	Delta winglet height, m
BR	Blockage ratio, (b/D)
D	Diameter of the tube, m
f	Friction factor
h	Convective heat transfer coefficient, W m^{-2} K^{-1}
k_a	Thermal conductivity of air, W m^{-1} K^{-1}
L	Periodic length (distance between baffles), m
Nu	Nusselt number
P	Static pressure, Pa
Pr	Prandtl number
Re	Reynolds number, ($\rho u_0 D/\mu$)
T	Temperature, K
TEF	Thermal Enhancement Factor, $(Nu/Nu_0)/\ (f/f_0)^{1/3}$

Greek Symbol

μ	Dynamic viscosity, kg s^{-1} m^{-1}
ρ	Density, kg m^{-3}

Subscript

0	smooth duct
pp	pumping power