Numerical heat transfer study in a round tube with 60° V-shaped rings

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Abstract. Turbulent flow and heat transfer behaviors in a 3-dimensional isothermal tube mounted repeatedly with 60° V-shaped rings (V-rings) are numerically investigated. The computation based on the finite volume method was conducted for the airflow rate in terms of Reynolds number (Re) in the range of 4000 to 20,000. The ring parameters include three flow blockage ratios, (BR=e/D=0.1, 0.15 and 0.2) and a single ring pitch ratio, PR=1. The computed result shows that friction factor and Nusselt number (Nu) increase with the increment in BR and the maximum thermal enhancement factor (η) of about 1.73 with Nu/Nu₀ = 4.5 is found at BR=0.1 and Re=4000. For the need of maximum heat transfer enhancement (Nu/Nu₀), the case of BR=0.2 V-ring is selected and modified since this case provides the highest heat transfer. The modification is made by cutting off some area of both ends of the BR=0.2 V-ring and the effect of cutting off (called end-cut ratio, ER=c/e) on thermal performance is also examined. The study reveals that for the end-cut V-ring, the friction factor is considerably decreased while the heat transfer rate reduces moderately with increasing ER. The end-cut V-ring yields the maximum η of 1.64 at ER=0.9, Re=4000 with Nu/Nu₀ = 5.4. Thus, the Nu/Nu₀ of the end-cut one is about 20% higher than that of the BR=0.1 V-ring.

1. Introduction
The application of vortex generators (VGs) in a heat exchanger system is commonly known as enhancement devices for augmenting the convective heat transfer coefficient leading to the compact heat exchanger and increasing thermal performance of such a system. The outstanding characteristic of the compact heat exchanger with VGs is considerably higher heat transfer rate, leading to reduced space, weight, energy requirements and costs [1]. The series of baffle/ring/rib is often used in the design of heat exchanger systems in order to increase the degree of cooling/heating levels. Although the heat transfer rate is increased through the baffle/ring arrangement, the pressure drop of the tube is also increased to the decreased flow area effects. Therefore, ring geometry and arrangements such as ring spacing, angle of attack and height/width are among the most significant parameters in the design of heat exchanger tubes. Promvonge et al. [2] studied the effect of flow blockage ratio and pitch ratio of using inclined rings inserted into a round tube on thermal performance behaviors. They found that the ring with larger BR provided higher heat transfer and pressure loss while the one with larger PR yielded the reversing trend. Chingtuaythong et al. [3] examined the influence of relative ring-pitch and blockage ratios at a constant attack angle of the V-shaped rings on heat transfer and flow resistance characteristics. They found that the heat transfer rate was up to 5.8 times above the plain tube whereas the friction factor was up to 82 times. Jedsadaratanachai et al. [4] investigated numerically the laminar
periodic flow and heat transfer characteristics in a circular tube fitted with 45° V-baffles and reported that the optimum thermal enhancement factor was about 3.20 at BR=0.2 for the V-upstream and BR=0.25 for the V-downstream baffles. Poomsalood et al. [5] numerically studied the turbulent flow and heat transfer characteristics in a round tube fitted with angled rings and indicated that the maximum $\eta$ was about 1.35 at PR=1.75. In general, the experimental investigation on thermal performance in a heat exchanger system is very costly and takes time due to lots of parameters involved. The numerical investigation is considered to be a suitable technique for predicting a wide range of parameters, due to its advantages of geometric change and flexibility. Numerical study has a vital role in developing a heat transfer enhancement technique for application purposes. Nevertheless, thermal behaviors of turbulent flow is quite complicated and difficult to understand the heat transfer mechanism occurring in a heat exchanger tube, especially for experimental work. In the present work, a numerical computation for three dimensional turbulent flows through V-shaped rings mounted repeatedly in a round tube is conducted to examine the flow structure and heat transfer characteristics for turbulent regime.

2. Flow description

2.1. Geometry and configuration of V-shaped rings

The flow system under consideration is a circular tube fitted repeatedly with 60° V-rings as depicted in Fig. 1. The flow model is considered as fully periodic flow by using the concept of periodically fully developed flow [6]. In the figure, the ring pitch, $P$ is a distance between the adjacent V-rings and set to $P=D$ in which $P/D$ is defined as the pitch ratio, PR. The air enters the tube having inner diameter of $D$ set to 0.05 m. $e$ is the ring width and $e/D$ is known as the blockage ratio, BR. To investigate the effect of V-ring inserts on thermal performance of the tube, BR is varied between 0.1 and 0.2 while the PR and attack angle ($\alpha$) are kept constant at 1.0 and 60°, respectively. To obtain the maximum heat transfer rate, further investigation is made by cutting off both ends of the BR=0.2 V-ring to reduce the pressure loss. Then, the effect of end cut-off ratio (ER=$c/e$) for the BR=0.2 ring on thermal performance is examined by varying ER values set to 0.0 (no cut), 0.1, 0.5 and 0.9 in which $c$ is the cut-off length at both ends of the V-ring as seen in Fig. 1.

![Figure 1. Tube geometry and computational domain of periodic flow.](image)

Due to ring shape symmetry, only one-fourth of the tube flow model is used as the computational domain as shown in Fig. 1. For grid independence test, the variations in Nu and $f$ for PR=1.0, BR=0.1 and Re=10,000 are found to be less than 0.5% when increasing the number of cells from 110,156 to 227,274, hence there is no such advantage in increasing the number of cells beyond this value.
Considering both convergence time and solution precision, the grid system of 110,156 is employed for the current computation.

2.2. Boundary conditions

Periodic boundaries are used for the inlet and outlet of the flow domain. Constant mass flow rate of air at 300 K is assumed in the flow direction due to fully periodic flow conditions. The physical properties of the air have been assumed to remain constant at the average air bulk temperature.

Table 1. Range of geometric parameters and operation conditions.

| Parameters                                 | Range          |
|--------------------------------------------|----------------|
| **Flow Conditions**                        |                |
| Reynolds number \((Re)\)                   | 4000 to 20,000 |
| **V-ring geometry and tube**               |                |
| Diameter of tube, \(D\) (mm)               | 50             |
| Ring Pitch, \(P\) (mm)                     | 50             |
| Pitch ratio, \(PR=P/D\)                    | 1.0            |
| Ring width, \(e\) (mm)                    | 5, 7.5 and 10  |
| Blockage ratio, \(BR=e/D\)                 | 0.1, 0.15 and 0.2 |
| Cut-off length, \(c\) (mm)                 | 0 (No cut), 0.5, 2.5 and 4.5 |
| End cut-off ratio, \(ER=c/e\)              | 0 (No cut), 0.1, 0.5 and 0.9 |
| Attack angle, \(\alpha\)                   | 60°            |

3. Mathematical foundation and key parameters

In the present study, the numerical simulation of fluid flow and heat transfer in a round tube is developed under the following assumptions: steady three-dimensional, turbulent and incompressible flow; omitting all the body forces, viscous dissipation and radiation heat transfer. Based on the above assumptions, the tube flow model is governed by the continuity, the Navier-Stokes equations and the energy equation [7]. In the Cartesian tensor system these equations can be written as follows:

Continuity equation:

\[ \frac{\partial}{\partial x_i} (\rho u_i) = 0 \]  (1)

Momentum equation:

\[ \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho u_i u_j \right] \]  (2)

Energy equation:

\[ \frac{\partial}{\partial x_j} (\rho u_i T) = \frac{\partial}{\partial x_j} \left[ \left( \Gamma + \Gamma_j \right) \frac{\partial T}{\partial x_j} \right] \]  (3)

For the Realizable \(k-\varepsilon\) turbulence model, its transport equations are expressed as:

\[ \frac{\partial}{\partial x_j} (\rho k) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + 2\mu S_{ij} S_{ij} - \rho \varepsilon \]  (4)

\[ \frac{\partial}{\partial x_j} (\rho \varepsilon) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_{1\varepsilon} \varepsilon - \rho C_{2\varepsilon} \frac{\varepsilon^2}{k + \sqrt{\varepsilon}} \]  (5)

In the above equations, \(\sigma_k\) and \(\sigma_\varepsilon\) are the turbulent Prandtl numbers for \(k\) and \(\varepsilon\), respectively.
All the governing equations are discretized by the QUICK numerical scheme with the SIMPLE algorithm for handling the pressure-velocity coupling and are solved by a finite volume method. For closure of the equations, the realizable $k-e$ turbulence model is used in the present study. The solutions are converged when the normalized residual values are less than $10^{-6}$ for all variables but less than $10^{-9}$ only for the energy equation.

There are four key parameters of interest in the present work, namely, the Reynolds number, friction factor, Nusselt number and thermal enhancement factor. The Reynolds number is defined as

$$Re = \frac{\rho u_0 D}{\mu}$$

(6)

The friction factor, $f$ is computed by pressure drop, $\Delta P$ across the length of tube flow model, $L$ as

$$f = \frac{(\Delta P/L)D}{(1/2)\rho u_0^2}$$

(7)

The local heat transfer is measured by local Nusselt number which can be written as

$$Nu_x = h_x D/k_a$$

(8)

in which $k_a$ is thermal conductivity of air. The area-average Nusselt number can be obtained by

$$Nu = \left\langle f/A \right\rangle \int Nu_x dA$$

(9)

The thermal enhancement factor ($\eta$) defined as the ratio of the dimensionless heat transfer coefficient of an inserted tube, $Nu$ to that of a smooth tube, $Nu_0$, at an equal pumping power is given by

$$\eta = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f/f_0}\right)^{1/3}$$

(10)

Where $Nu_0$ and $f_0$ stand for Nusselt number and friction factor for the smooth tube, respectively.

4. Results and discussion

4.1. Validation

In the present simulation, the numerical results are validated with the previous experimental data taken from [3]. The comparison of the predicted friction factor and Nusselt number is presented in Fig. 2. The predicted friction factor and Nusselt number obtained from the numerical solutions are in reasonable agreement with the experimental data. The maximum deviations of the predicted friction factor and Nusselt number with measured data are, respectively, within $\pm 19\%$ and $\pm 15\%$ for the 30° V-ring at PR=2.0 and BR=0.1.

![Figure 2. Verification of Nu and $f$ with measurements [3].](image-url)
4.2. Flow structure
Fig. 3(a) and (b) displays the temperature contours with streamlines in transverse planes at different locations for (a) typical (no end-cut) and (b) ER=0.9, end-cut V-rings, respectively. It can be observed that there are four main longitudinal vortex flows along the tube, two counter-rotating vortices appear on the upper and lower tube due to V-ring symmetry for both the typical V-ring and end-cut V-ring at ER=0.9. The scrutiny of the figure reveals that two pairs of common-flow-down vortices appear on both sides of the tube. These vortices are responsible for the increase in heat transfer rate along both sidewalls of the tube, lower temperature regions can be observed along both sidewalls. Higher temperature areas (red color regions) are found on the upper and lower parts of the tube.

4.3. Heat transfer
Temperature contours of using the V-rings at Re=4000 for the typical and cut-off V-rings, ER=0.9 are also displayed in Fig. 3(a) and (b), respectively. In Fig. 3(a) temperature become poorer around the top and bottom areas where the end of V-rings attached. In Fig. 3(b), the high temperature area on the tube wall is pretty linear strip in axial direction over the rings on the top and bottom. It is visible that the vortex flows provide a significant influence on the temperature field. When the end-cut ratio (ER) increases, the high temperature zone appears. This can be explained that the end-cut with larger ER allow the fluid flows through the cut area more easily, resulting in weaker turbulence and fluid mixing and consequently less efficient heat transfer. The local Nusselt number (Nu) contours on the tube wall for the V-ring with BR=0.1, 0.15 and 0.2 at Re=4000 are presented in Fig. 4(a), 4(b) and 4(c), respectively. The intensity of the Nu contours is displayed by different colors. The peaks (red color) can be observed at the impingement areas on the tube wall where the rings attached. In the cases studied, it is apparent that the typical V-ring with BR=0.2 gives the highest Nu as seen in Fig. 4c. For all cases, the heat transfer rate is intense around the ring, except for a small region on the top and bottom.
Figure 4. Nu contours for 60° typical V-ring at Re=4000, PR=1; (a) BR=0.1, (b) BR=0.15, (c) BR=0.2.

4.4. Effect of BR

Fig. 5 presents the pressure drop of V-rings in terms of friction factor ratio, \( \frac{f}{f_0} \) against Re for various BRs. In the figure, it is noted that \( \frac{f}{f_0} \) tends to increase with the rise of Re and BR values. The use of the V-rings leads to considerable increase in friction factor in comparison with the smooth tube with no ring. The larger BR yields higher pressure loss across the tube. According to numerical results, \( \frac{f}{f_0} \) is found to increase considerably with the increment in BR. The lowest and highest \( \frac{f}{f_0} \) are, respectively, found to be 24 and 235 times for BR=0.1, Re=4000 and BR=0.2, Re=20,000. The decrease in BR gives rise to the reduction of friction factor. The \( \frac{f}{f_0} \) for BR=0.1 and BR=0.15 is, respectively, about 24–39 and 65–102 times while that for BR=0.2 is highest around 150–235 times depending on Re values. Thus, the use of larger BR should be avoided due to extreme pressure drop increase.

Figure 5. Variations of \( \frac{f}{f_0} \) and Nu/Nu\(_0\) with Re for V-rings.

The variation of heat transfer in terms of Nu/Nu\(_0\) against Re is also plotted in Fig. 5. Nu/Nu\(_0\) tends to decrease with the rise in Re for all BRs and the lower BR leads to the decrease of Nu/Nu\(_0\). It is observed that Nu/Nu\(_0\) increases with the increment of BR while the minimum and maximum Nu/Nu\(_0\) values are, respectively, about 3.25 and 7.33 times at BR=0.1, Re=20,000 and BR=0.2, Re=4000. The use of the V-ring yields the heat transfer rate of about 3–7 times higher than the smooth tube.
4.5. Performance evaluation

![Figure 6](image_url)

Figure 6. $\eta$ versus BR for V-ring.

Fig. 6 displays the variation of $\eta$ with BR at different Re values. It is visible that $\eta$ shows the decreasing trend with the increase in BR and Re. The highest $\eta$ is 1.73 at BR=0.1, Re=4000 while the lowest $\eta$ is 0.71 at BR=0.2 and Re=20,000. Hence, to obtain higher thermal performance the smaller BR should be used.

4.6. Thermal performance improvement for larger BR

Apart from optimum thermal performance as mentioned in section 4.5, the possibly highest heat transfer enhancement is required to reduce the heat exchanger size. In this section, only the BR=0.2 case is considered to be modified because it gives the maximum Nu/Nu$_0$ as can be seen in Fig. 5. The modification is made by cutting off both ends of the V-ring as mentioned earlier and the numerical results are described next. The variation of $f/f_0$ and Nu/Nu$_0$ for the end-cut ring with Re at various ERs is depicted in Fig. 7(a). At the same ER, $f/f_0$ increases with the increase of Re whereas the larger ER provides lower resistance to the flow. For the range investigated, the use of typical V-ring (BR=0.2) leads to the increase in $f/f_0$ from 150-235 times and it can be reduced considerably for ER=0.9 (37-62 times). This indicates that the use of end-cut V-ring can reduce the friction factor around 3.7-4 times below the ring with no cut. The Nu/Nu$_0$ decreases with increasing ER and the highest Nu/Nu$_0$ is about 5.4 compared with 7.3 times for the no cut ring or around 30% lower.

![Figure 7](image_url)

Figure 7. Comparison of (a) $f/f_0$ and Nu/Nu$_0$ (b) $\eta$ with Re for various ER.
Fig. 7(b) exhibits the variation of $\eta$ with Re for various ERs. The larger ER causes extremely lower friction loss and also gives lower heat transfer enhancement. The effect of ER on $\eta$ signifies that the influence of decreasing friction loss is more significant than that of decreasing heat transfer. The maximum $\eta$ of 1.64 is achieved by the end-cut V-ring with ER=0.9 at Re=4000 while the typical BR=0.2 V-ring is 1.37 as seen in Fig. 7(b).

5. Conclusions
A numerical study has been conducted to investigate the effect of BRs of using the V-ring insert on turbulent flow, heat transfer, flow resistance and thermal performance characteristics in a round tube. The computed result for typical V-rings shows that the friction factor and Nusselt number increase with the increment of BR. The lower BR leads to the lowest friction factor and Nusselt number. The maximum thermal performance for the V-ring is about 1.73 at BR=0.1 and Re=4000 with Nu/Nu$_0$ around 4.5. The improvement on thermal performance for using larger BR by allowing some of air flow through the space above the V-rings in order to reduce the flow resistance. The increase of end-cut ratio (ER) results in the decrease of heat transfer and friction loss. Among the end-cut rings, the larger ER gives the highest thermal enhancement factor of 1.64 at BR=0.2 and Re=4000 with Nu/Nu$_0$ around 5.4.

6. References

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