Numerical investigation on turbulent forced convection and heat transfer characteristic in a square channel with discrete combined V-baffle and V-orifice

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\textbf{Abstract}

Turbulent forced convection, heat transfer and performance improvement in a square channel with discrete combined baffles (DCB), which combined V-baffle and V-orifice, are investigated numerically. The influences of the flow blockage ratios ($BR = 0.05, 0.10$ and $0.15$) and V-tip directions (V-tip pointing downstream called “V-Downstream” and V-tip pointing upstream called “V-Upstream”) are examined with a single pitch spacing ratio, $PR = 1$, and attack angle, $\alpha = 30^\circ$, for the Reynolds number, $Re = 5000$–$20,000$. The computation results are reported in terms of flow visualizations, heat transfer characteristics, performance assessments. The results are compared with the smooth channel and the previous works. As the results, the DCB enhances the heat transfer rate and thermal efficiency due to the disturbance of the thermal boundary layer. The improvement of the heat transfer rate is around 2.8–6 times higher than the smooth channel depended on $BR$, V-tip directions and $Re$. In addition, the computational result reveals that the optimum thermal enhancement factor, $TEF$, is around 1.72 at $BR = 0.1$, $Re = 3000$ and V-Upstream.

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1. Introduction

Passive technique had been used to augment heat transfer rate and thermal performance in various types of heat exchangers; fin-and-tube heat exchanger, solar air heater channel, shell-and-tube heat exchanger, etc. The main objective of the passive technique is to generate the vortex flow, longitudinal vortex flow, swirling flow, impinging flow in the heating section. These behaviors disturb the thermal boundary layer on the heat transfer surface that helps to increase the heat transfer rate and efficiency.

The numerical and experimental investigations on flow and heat transfer characteristics in heating or cooling systems installed with V-shaped turbulators had been widely reported [1–14]. The parameters such as height, attack angle, arrangement, etc., of the V-shaped turbulators were studied. The researchers found that the use of the V-shaped turbulators in the system performs higher heat transfer rate and thermal efficiency than the typical system. They also concluded that the V-shaped turbulators give higher effectiveness than the other shapes of the turbulators. The V-shaped turbulators were also combined with the orifice called “V-orifice”. The installation of the V-orifice in the heat exchanger was investigated by
Boonloi and Jedsadaratanachai [16]. They stated that the high pressure loss is found when installed with the V-orifice. In conclusion, the V-shaped turbulators not only increase in heat transfer rate, but also enhance pressure loss. Therefore, the modified shape and installation method, which help to reduce the pressure loss in the heating system, were investigated [15].

As the previous literature reviews, it is found that the V-shaped baffle has effectiveness higher than the other types of the vortex generators. In the current investigation, the discrete combined baffles (DCB), which are modified from the V-baffle [14] and V-orifice [16], are inserted in the channel heat exchanger to improve the heat transfer rate and performance. The diagonal installation of the DCBs in the channel heat exchanger is referred from Refs. [14,15]. The proposed DCBs can be readily manufactured by forming process and conveniently installed in the actual heat exchanger unit. The discrete configuration [15] is applied to the DCBs to decrease the pressure loss and also helps to enhance the strength of the vortex flow (increase vortex intensity). The influences of the blockage ratios \(b/H\) and flow directions (V-Downstream and V-Upstream) are investigated for the turbulent regime, \(Re=3000–20,000\). The pitch ratio and flow attack angle of the DCB are fixed at 1 and 30°, respectively. The numerical method is selected to explain the behaviors of the presented problem. Knowledge of the mechanisms on flow and heat transfer characteristics is a powerful tool for further improvement of the heat exchanger.

### 2. Computational configuration

The configurations of the V-baffle, V-orifice, full combined baffle and discrete combined baffle are reported in the Fig. 1a, b, c and d, respectively. The square channel with the DCBs and the computational domain are depicted as Fig. 2a. The DCB is designed from the combination between V-baffle and V-shaped orifice. The discrete configuration (like staggered arrangement) is used to reduce the pressure loss in the heating section. The square channel height, \(H\), is set around 0.05 m. The DCBs inserted in the square channel heat exchanger with gap ratio, \(e/H\), of 0.01. The effects of the blockage ratios \(b/H\) \(H=0.05, 0.10\) and 0.15, and flow directions (V-Downstream and V-Upstream) are investigated for the Reynolds numbers, \(Re=3000–20,000\), with a single pitch ratio \((P/H)\) of 1 and flow attack angle of 30°. The grid resolution on the square channel walls is presented in Fig. 2b.

### 3. Mathematical foundation

The incompressible turbulent flow with steady operation in three dimensions and heat transfer characteristic in the square duct agree governed by continuity, Navier-Stokes and energy equations.

The realizable \(k-\varepsilon\) turbulent model with enhanced wall treatment is used to solve the present problem.
and
\[
\frac{\partial}{\partial t} (\rho e) + \frac{\partial}{\partial x_j} (\rho e u_j) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_1}{\sigma} \right) \frac{\partial e}{\partial x_j} \right] + \rho C_S h e - \rho C_2 \frac{e^2}{K + \sqrt{\pi}} + C_t \frac{e}{k} C_b G_b + S_e
\]  

where
\[
C_t = \max \left[ 0.43, \frac{\eta}{\eta + 5} \right], \nonumber \eta = \frac{s_k}{\epsilon}, \nonumber S = \sqrt{2s_{ij}}
\]
Fig. 2. (a) Square channel with DCBs and the computational domain and (b) mesh resolution of the square channel walls.
the constants in the model are given as follows:

\[ C_1 = 1.44, C_2 = 1.9, \sigma_k = 1.0, \sigma_f = 1.2 \]  

(4)

The QUICK numerical scheme is selected for all governing equations, decoupling with the SIMPLE algorithm using a finite volume method with FLUENT code. The solutions are measured to be converged when the normalized residual are less than $10^{-9}$ and $10^{-5}$ for the energy equation and the other variables, respectively.

The main parameters for the current investigation are Reynolds number ($Re$), friction factor ($f$), Nusselt number ($Nu$) and thermal enhancement factor ($TEF$).

The Reynolds number is calculated from Eq. (5).

\[ Re = \frac{\rho u_0 D_h}{\mu} \]  

(5)

\( \rho, \mu \) and \( u_0 \) are density, viscosity and inlet velocity of the fluid, respectively, while \( D_h \) is the hydraulic diameter of the square channel.

The friction factor is defined as

\[ f = \frac{(\Delta P/L)D_h}{2\rho u^2} \]  

(6)

\( \Delta P \) is the pressure drop across the periodic module, \( L \), and \( u \) is mean flow velocity.

The local Nusselt number is computed by

\[ Nu_x = \frac{h_x D_h}{k} \]  

(7)

\( h_x \) is local heat transfer coefficient based on bulk temperature and \( k \) is thermal conductivity of the air.

The average Nusselt number can be printed by

\[ Nu = \frac{1}{A} \int Nu_x dA \]  

(8)

where, \( A \) is heat transfer area of the square channel heat exchanger.

Thermal performance enhancement factor ($TEF$) is defined as the ratio of the heat transfer coefficient of an augmented surface, \( h \) to that of a smooth surface, \( h_0 \), under the constant pumping power condition. The $TEF$ can be expressed as follow;

\[ TEF = \left( \frac{h}{h_0} \right)_{pp} = \frac{Nu}{Nu_0} = \left( \frac{f}{f_0} \right) \left( \frac{Nu}{Nu_0} \right)^{1/3} \]  

(9)

The \( f_0 \) and \( Nu_0 \) are the friction factor and the Nusselt number of the smooth tube, respectively.

4. Boundary condition and assumption

No-slip wall condition is used for all sides of the channel heat exchanger and DCBs. Constant heat flux around 600 W/m² is applied to the channel walls, while the baffles are set as an adiabatic wall condition (insulator). The air (\( Pr=0.7 \)) as test fluid with constant properties at 300 K flows into the channel heat exchanger. The periodic condition is applied for inlet and outlet of the computational domain. The flow and heat transfer of the domain are developed under steady state. The fluid flow is turbulent and incompressible. The body force, viscous dissipation, natural convection and radiation heat transfer are ignored.

5. Numerical result and discussion

5.1. Numerical validation

The validations of the computational domain are separated into three parts; grid independence, validation with the smooth channel and validation with the experimental result. The hexahedral mesh is selected for the present computational domain. The grid independence is done by compared nine sets of the grid cells (42000, 62700, 81600, 127500, 187200, 232400, 352000, 418500 and 619400) on the Nusselt number and friction factor for the DCB with \( BR=0.1, PR=1 \) and \( \alpha=30^\circ \).

The grid cell of 232400 performs around 0.74% and 0.54% deviation for the Nusselt number and friction factor, respectively, in compared with 619400 cells. Therefore, the grid of 232400 is selected for all cases.

The validations of the Nusselt number and friction factor for the smooth channel are done. The turbulent models (Realizable $k$-$\varepsilon$ and SST $k$-$\omega$ models) are also compared in this section. The Realizable $k$-$\varepsilon$ provides the lowest relative errors
around 4.27% and 5.4%, respectively, for the Nusselt number and friction factor when compared with the Dittus-Boelter and Blasius correlations [18].

Fig. 3 reports the comparison between the values of the previous work [19] and the present work on the Nusselt number.

Fig. 4. Streamlines in transverse planes for the square channel with DCB at $BR = 0.1$ and $Re = 3000$ for (a) V-Upstream and (b) V-Downstream.
and friction factor. The current results are found in good agreement with the previous results on both the heat transfer rate and friction loss. Therefore, it can be concluded that the computational domain has reliable to predict the heat transfer and flow structure in the channel heat exchanger.

5.2. Flow pattern and heat transfer behavior

Fig. 4a and b display the streamlines in transverse planes of the DCB in the channel with $BR=1$ and $Re=3000$ for V-Upstream and V-Downstream, respectively. It is found that the DCB can create eight small vortices through the channel. The positions of the vortex cores for both cases are found similarly, but in an opposite direction. The vortex flow helps to improve fluid mixing in the channel. The use of the DCB in the heating section gives the number of the vortices higher than the inclines baffle [15], V-baffle [14] and orifice [16]. The enhancement of the vortices leads to a better distribution of the local Nusselt number on the channel walls, but the strength of each vortex may decrease.

The plots of the impinging flow on the channel wall with the local Nusselt number distributions are presented in Fig. 5 for V-Upstream case at $BR=0.1$ and $Re=3000$. As the figure, the V-Upstream produces the impinging flow on the channel wall.

Fig. 5. Impinging flow on the square channel walls for V-Upstream at $BR=0.1$, $Re=3000$.

Fig. 6. Local Nusselt number distributions in transverse planes for the square channel with DCBs at $BR=0.1$ and $Re=3000$ for (a) V-Upstream and (b) V-Downstream.
walls. The impinging areas (red contour) give higher heat transfer rate than the other regimes. The impingement of the fluid flow disturbs the thermal boundary layer on the wall regions that helps to improve the heat transfer rate and thermal efficiency in the channel heat exchanger. The impinging flow on the channel wall is also detected in case of V-Downstream. In conclusion, the impingement of the fluid flow is a main key for the heat transfer enhancement.

The local Nusselt number distributions on the channel walls for the V-Upstream and V-Downstream cases are reported in the Fig. 6a and b, respectively, at $BR=0.1$ and $Re=3000$. As the figures, the V-Upstream performs greater Nusselt number than the V-Downstream. The reason of this may be that the V-Upstream of the DCB can produce stronger impinging flow than the V-Downstream.

Fig. 7. Variations of the (a) $Nu/Nu_0$, (b) $f/f_0$ and (c) TEF with the Reynolds number of the square channel with DCBs and comparison with the V-baffle inserted diagonally in the channel.
5.3. Performance assessment

The performance assessments in the square channel heat exchanger with the DCB are plotted in terms of the Nusselt number ratio \( \dfrac{\text{Nu}}{\text{Nu}_0} \), friction factor ratio \( \dfrac{f_f}{f_0} \) and thermal enhancement factor \( \text{TEF} \). Fig. 7a presents the variations of the \( \dfrac{\text{Nu}}{\text{Nu}_0} \) with the Reynolds number at various cases for the channel with the DCB. Generally, the \( \dfrac{\text{Nu}}{\text{Nu}_0} \) tends to decrease with increasing the Reynolds number for all cases. The DCB in the channel heat exchanger can improve the heat transfer rate higher than the smooth channel (\( \dfrac{\text{Nu}}{\text{Nu}_0} > 1 \)). The BR = 0.15 gives the highest heat transfer rate, while the BR = 0.05 performs the reverse trend for both V-Upstream and V-Downstream cases. The V-Upstream provides higher heat transfer rate than the V-Downstream around 13%, 17.31% and 12.5% for BR = 0.15, 0.10 and 0.05, respectively. The maximum Nusselt number is around 6, 5.2 and 3.2 times higher than the smooth channel for BR = 0.15, 0.10 and 0.05, respectively, for V-Upstream case, while around 5.2, 4.3 and 2.8 times for V-Downstream. In range studied, the \( \dfrac{\text{Nu}}{\text{Nu}_0} \) is found around 2.8–6.0.

Fig. 7b illustrates the variations of the \( \dfrac{f_f}{f_0} \) with the Reynolds number at various BRs and flow directions. In general, the \( \dfrac{f_f}{f_0} \) increases with raising the Reynolds number. The highest friction loss is detected at BR = 0.15, while the BR = 0.05 performs the lowest values. The V-Upstream provides higher friction loss than the V-Downstream. The friction factor is around 7.65–127.55 and 7.12–97.82 times above the smooth square channel for the V-Upstream and V-Downstream, respectively. Additionally, the V-Downstream with low BR can reduce the pressure loss in the channel heat exchanger.

Fig. 7c reports the variations of the TEF with the Reynolds number at various cases. In general, the TEF tends to decrease with augmenting the Reynolds number. The optimum TEF is around 1.72, 1.6 and 1.3, respectively, for BR = 0.10, 0.05 and 0.15, for V-Upstream case. The V-Downstream gives the maximum TEF around 1.5, 1.42 and 1.28 for BR = 0.05, 0.10 and 0.15, respectively, at \( Re = 3000 \). Although, the BR = 0.15 yields the highest heat transfer rate, but also offers very large pressure loss, therefore, lowest TEF is detected for both V-Downstream and V-Upstream cases.

In comparison, the DCB performs higher TEF than the V-baffle [14]. The optimum TEF for the V-baffle is around 1.34, while around 1.72 for the DCB. The V-Upstream of the DCB also provides upper TEF than the full and discrete inclined baffles [17].

6. Conclusion

The numerical investigations on heat transfer, pressure loss and thermal enhancement factor in the square channel heat exchanger with the DCB are performed. The influences of the blockage ratios (\( BR = 0.05, 0.10 \) and 0.15) and flow directions (V-Upstream and V-Downstream) are studied for the turbulent flow, \( Re = 5000–20,000 \), and also compared with the previous results. The conclusions from the current investigation are as follows:

The disturbance of the thermal boundary layer is found when inserted DCB in the heating section, that helps to improve thermo-hydraulic performance.

The strengths of the vortex flow and impinging flow increase when increasing BR. In the range investigates, the enhancements are 2.8–6 and 7.12–127.55 times over the smooth channel for the Nusselt number and friction factor, respectively, depended on BR, Re and flow direction. The optimum TEF is around 1.72 for BR = 0.10, Re = 3000, V-Upstream.

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