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Research Article

NUMERICAL ANALYSIS OF TURBULENT HEAT TRANSFER IN A SQUARE CHANNEL WITH V-BAFFLE TURBULATORS

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ABSTRACT:
Turbulent periodic flow and heat transfer in a three dimensional horizontal channel with isothermal walls and with diagonal broken V-baffles in form of tail-end cut (DBB-Ts) were numerically studied. The fluid flow and heat transfer characteristics were investigated under constant heat flux condition for Reynolds numbers based on the hydraulic diameter of the channel ranging from 6000 to 20,000. The effect of open corner ratio (d/H = 0.0, 0.01, 0.02, 0.03, 0.04 and 0.05) of DBB-Ts on thermohydraulic characteristics was investigated. The results reveal that the DBB-Ts with a smaller open corner ratios (d/H) gives higher heat transfer rate, friction factor as well as thermal enhancement factor than the one with larger d/H as a result of a stronger vortex intensity and thus better fluid mixing and more efficient thinning thermal boundary layer. In the present study, the maximum thermal enhancement is given by DBB-Ts with d/H = 0.

Keywords: *Thermal performance, baffle, heat transfer enhancement, channel flow, turbulator*

1. INTRODUCTION

Passive heat transfer enhancement is extensively used in a heat exchanger system in order to augment heat transfer and increase the thermal performance of the system. The passive technique can be attained without any external energy. One approach of passive heat transfer enhancement involves the modifications of flow surfaces as well as incorporations of inserts. Among the passive techniques, using baffle turbulators is one of the most promising techniques.

The researches on heat transfer enhancement utilizing baffle turbulators have been extensively carried out [1-4]. Mousavi and Hooman [5] numerically studied the heat transfer behavior in the entrance region of a channel with staggered baffles for Reynolds numbers ranging from 50 to 500 and baffle heights between 0 and 0.75 and reported that the Prandtl number affected the precise location of the periodically fully developed region. Sriromreun [6] reported the effect of baffle turbulators on heat transfer enhancement in a channel fitted with a zigzag shape (Z-shaped baffle). The Z-baffles inclined to 45° relative to the main flow direction are characterized at three baffle- to channel-height ratios (e/H = 0.1, 0.2 and 0.3) and baffle pitch ratios (P/H = 1.5, 2.0 and 3.0). The Nusselt number, friction factor and thermal enhancement factor for the in-phase 45° Z-baffles were found to be considerably higher than those for the out-phase 45° Z-baffle at a similar operating condition.

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Sripattanapipat and Promvonge [7] studied the heat transfer and pressure loss in a two dimensional horizontal channel with isothermal walls and with staggered diamond-shaped baffles by using finite volume method for Reynolds numbers between 100 and 600. The influence of baffle tip angle on fluid flow and heat transfer in the channel were investigated. The results of the diamond baffle were also compared with those of a flat baffle. It was found that 5°-10°diamond baffles gave superior heat transfer enhancement to the flat baffle, for all studied Reynolds numbers. In the present research, the numerical simulation for three dimensional turbulent periodic channel flows over diagonal broken V-baffles in form of tail-end cut (DBB-Ts) was carried out with the aim of studying the flow structure, heat transfer and thermal enhancement.

2. Flow Configuration

The system of the present research is a square channel with diagonal broken V-baffles in form of tail-end cut (DBB-Ts). The DBB-Ts were located on a plate which was diagonally placed in a channel as shown in Fig. 1. The height of the DBB-Ts, the height of the channel (*H*) and the rib pitch (*p*) were kept constant at 7.5, 50 and 50 mm, respectively. The dimension is in accord with constant pitch ratio (PR = *p/H*) and blockage ratio (BR = *b/H*) of 1.0 and 0.15, respectively. To investigate a geometry effect of the interaction between both baffles, the open corner ratios, *d/H* was varied from 0.00 to 0.05 (*d/H* = 0.00, 0.01, 0.02, 0.03, 0.04 and 0.05).

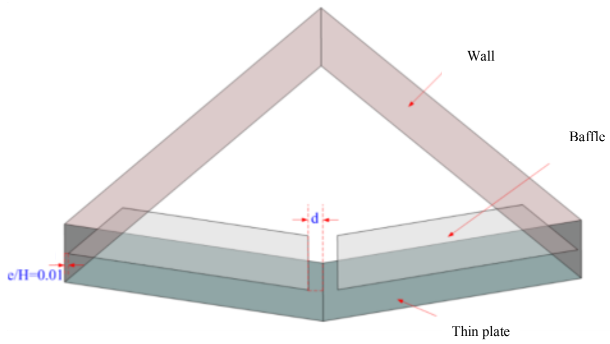


Fig. 1. Channel fitted with DBB-Ts and computational domain of periodic flow.

3. Mathematical Foundation

The numerical model for fluid flow and heat transfer in a square channel was developed under the following assumptions: steady three-dimensional fluid flow and heat transfer, the flow was turbulent and incompressible, constant fluid properties, body forces and viscous dissipation were ignored, negligible radiation heat transfer. Based on the above assumptions, the square channel flow was governed by the continuity, the Navier-Stokes equations and the energy equation. The computational domain was resolved by regular Cartesian elements. Based on the above assumptions, the three-start spirally twisted tube flow is governed by equations as below.

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum equation:

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

where $-\rho\overline{u_i' u_j'} = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_i}\right) - \frac{2}{3}\left(\rho k + \mu_t \frac{\partial u_i}{\partial x_i}\right)\delta_{ij}$

where ρ is the density of fluid, and u_i is a mean component of velocity in the direction x_i, p is the pressure, μ is the dynamic viscosity, and u' is a fluctuating component of velocity. Repeate indices indicate summation from one to three for three-dimensional problems. The parameter k is the turbulent kinetic energy, defined as $k = \frac{1}{2}\overline{u_i'u_j'}$ and δ_{ij} is a Kronecker delta. An advantage of the Boussinesq approach with the computation of the relatively low computational cost associated with the computation of the turbulent viscosity, μ_t given is $\mu_t = \rho c_\mu k^2 / \varepsilon$.

Energy equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i T\right) = \frac{\partial}{\partial x_j}\left(\left(\Gamma + \Gamma_t\right)\frac{\partial T}{\partial x_j}\right) \tag{3}$$

In this investigation, the effects of the square channel with diagonal broken V-baffles in form of tail-end cut (DBB-Ts) on friction factor, Nusselt number and thermal enhancement factor were reported. The Reynolds number (Re) is given as

$$\mathrm{Re} = \rho u D_h / \mu \tag{4}$$

The friction factor (f) is defined as

$$f = \frac{\left(\Delta p / L\right) D_h}{\frac{1}{2}\rho \bar{u}^2} \tag{5}$$

The average Nusselt number is given by

$$Nu_{ave} = \frac{1}{A}\int Nu_x \partial A \tag{6}$$

The thermal enhancement factor (*TEF*) is defined by

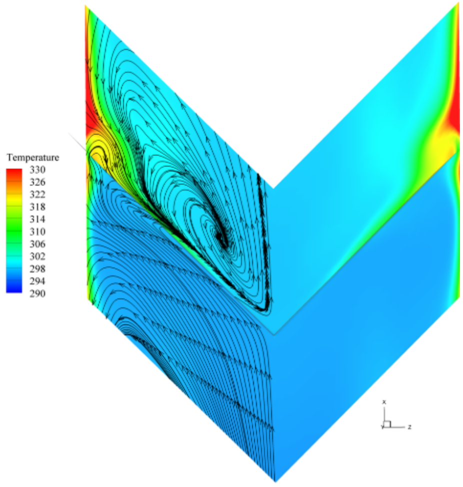
$$TEF = \left(Nu / Nu_0\right) / \left(f / f_0\right)^{1/3} \tag{7}$$

where Nu_0 and f_0 is the Nusselt number and friction factor for the smooth channel.

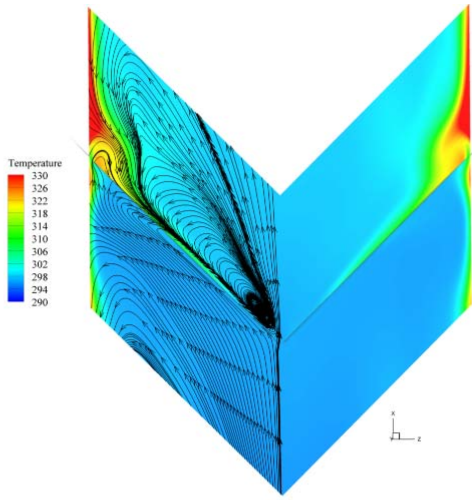
The computational domain was resolved by regular Cartesian elements. For this channel flow, regular grid was applied throughout the domain. Grid independent solution was obtained by comparing the solution for different grid levels. Periodic boundaries were used for the inlet and outlet of the flow domain. Constant mass flow rate of air with 300 K was assumed in the flow direction rather than constant pressure drop, due to periodic flow conditions. The inlet and outlet profiles of the velocities as well as dimensionless temperatures were identical. The physical properties of the air were assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions were implemented over the channel wall as well as the DBB-Ts.

4. Numerical Results

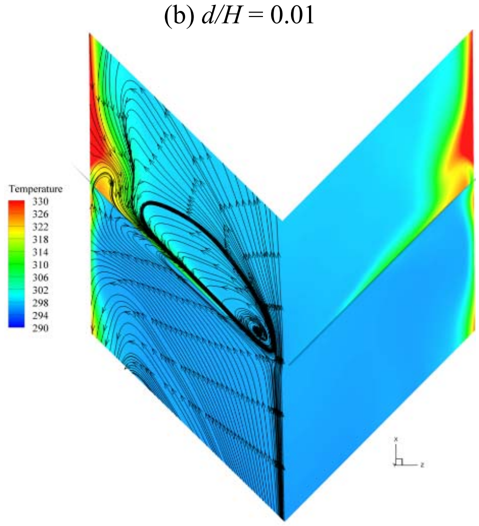
The flow structure in channel with DBB-Ts could be easily discerned by considering the streamline plots as depicted in Fig. 2(a-f) for the cases of open corner ratios (*d/H*). For all cases, recirculation flows exist behind DBB-Ts. The recirculation flow brings rotating motion into fluid and enhances fluid exchange between wall and core regions. The recirculation flow also helps to induce impingement flows on the wall leading to drastic increase in heat transfer rate in the channel. The size of the recirculation flow increases as *d/H* decreases. From the phenomena mentioned above, it can be concluded that the presence of the DBB-Ts leads to longer flow path and also stronger central vortex intensity. Figures 2 and 3 also show the contour plots of temperature field and local Nusselt number for different open corner ratios. It can be observed that the temperature field and local Nusselt numbers are influenced by the size of the recirculation flow. The recirculation flow with a larger size generated by DBB-Ts with smaller open corner ratio, gives better fluid mixing between the wall and the core regions, leading to a higher temperature gradient along the heating channel wall (Fig. 2) and thus more efficient heat transfer enhancement (Fig. 3).



(a) *d/H* = 0.0

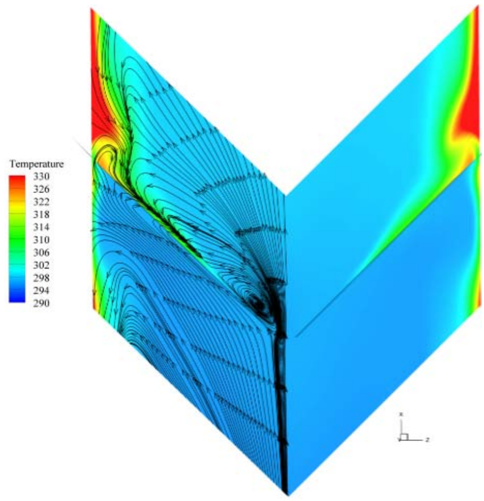


(b) *d/H* = 0.01

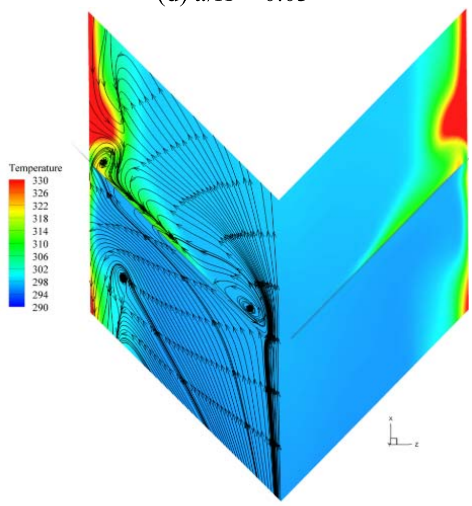


(c) *d/H* = 0.02

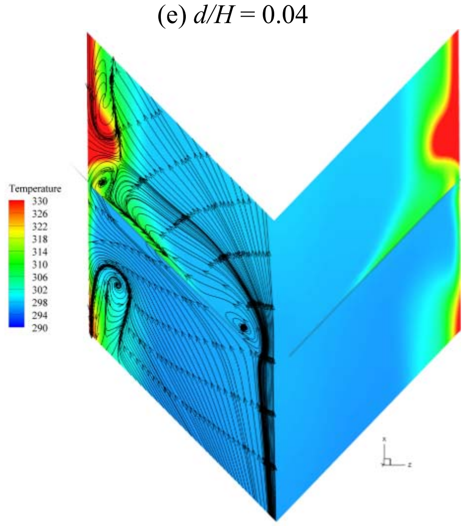
Fig. 2. Flow structure and temperature contours for various d/H at Re = 3000.



(d) d/H = 0.03

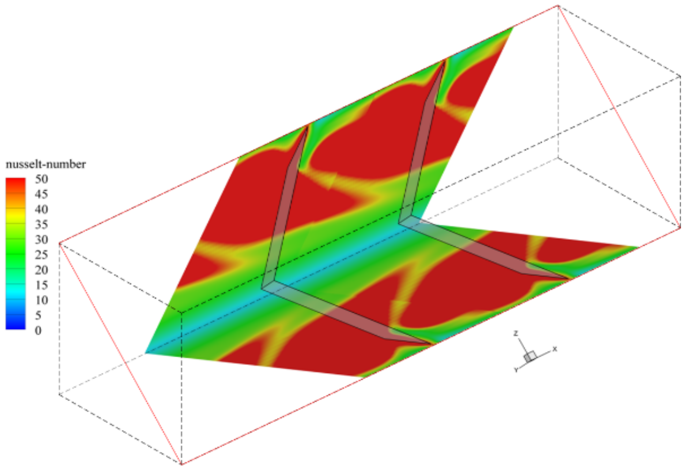


(e) d/H = 0.04

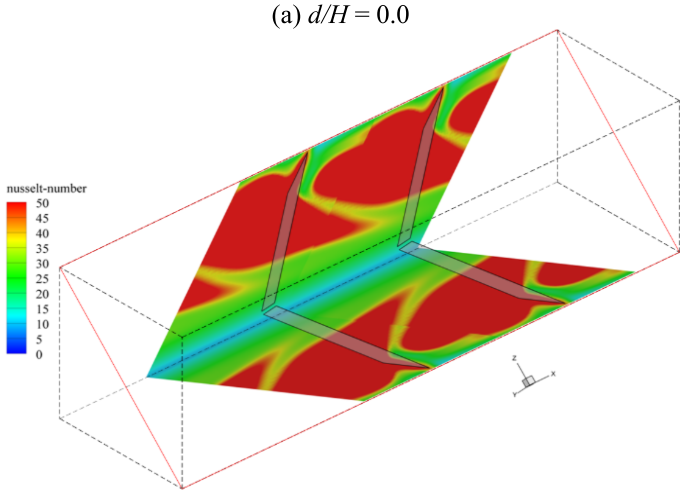


(f) d/H = 0.05

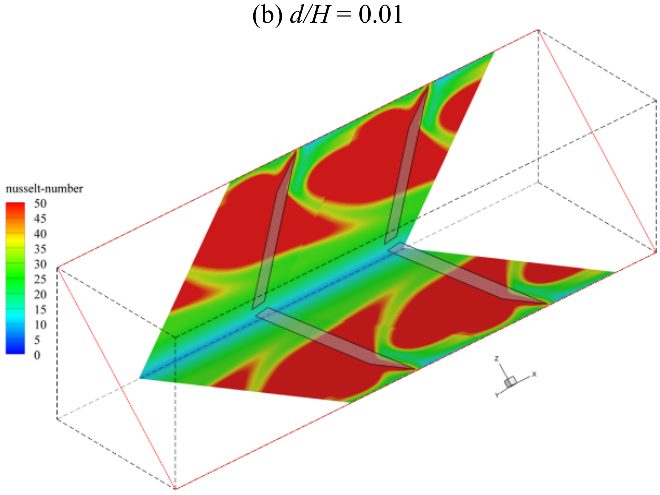
Fig. 2. Flow structure and temperature contours for various d/H at Re = 3000. (Cont.)



(a) d/H = 0.0

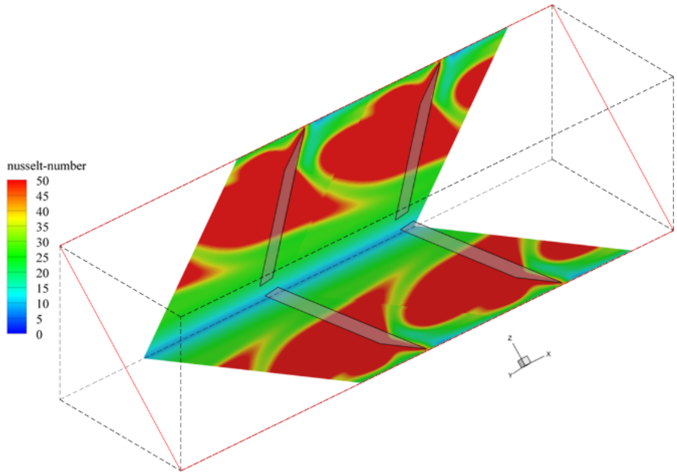


(b) d/H = 0.01

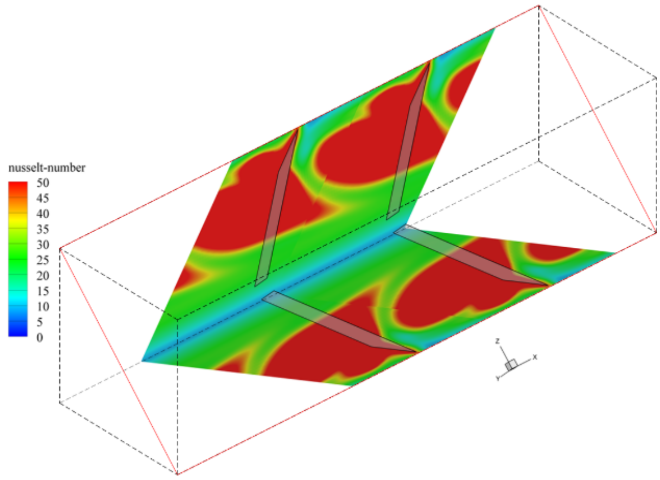


(c) d/H = 0.02

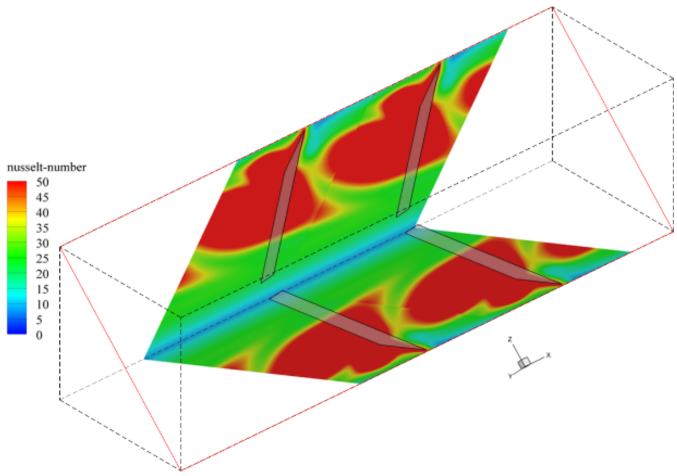
Fig. 3. Local Nusselt number distribution for various d/H at Re = 3000.



(d) *d/H* = 0.03

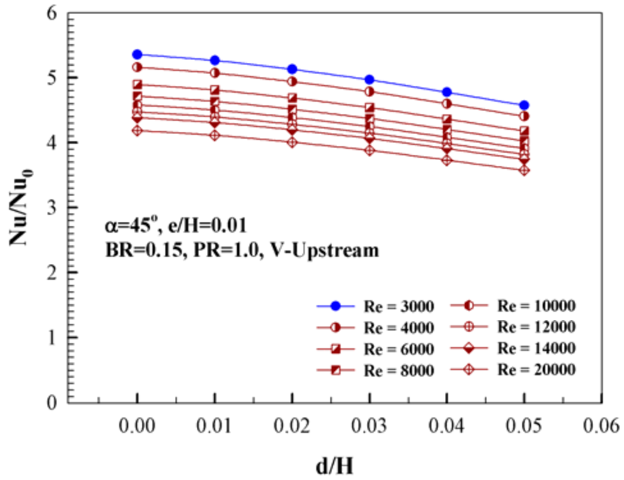


(e) *d/H* = 0.04

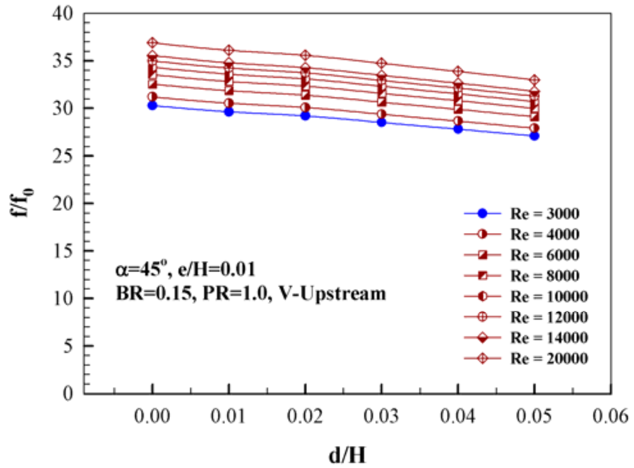


(f) *d/H* = 0.05

Fig. 3. Local Nusselt number distribution for various *d/H* at Re = 3000. (Cont.)



(a) *Nu/Nu₀*



(b) *f/f₀*

Fig. 4. Variation of Reynolds number with *d/H*.

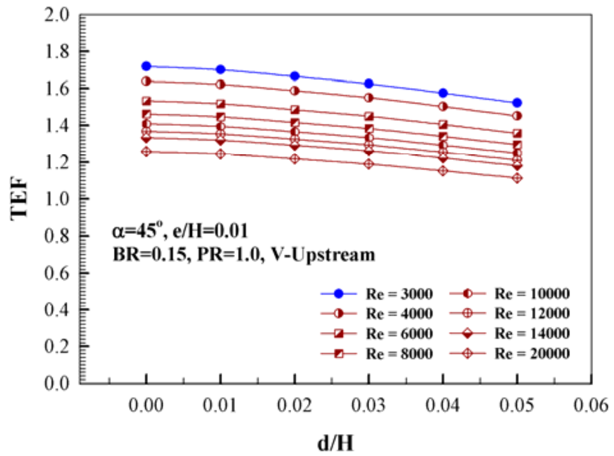


Fig. 5. Thermal enhancement factor for various *d/H*.

Figure 4a presents the variation of the average Nu/Nu_0 ratio with Reynolds number for DBB-Ts with different open corner ratios (*d/H*). For a given Reynolds number, Nu/Nu_0 value tends to increase with decreasing *d/H*, as DBB-Ts with smaller *d/H* give stronger central vortex intensity and better fluid mixing between the wall and the core regions. The variation of the friction factor ratio, f/f_0 with Reynolds number values for various open corner ratios (*d/H*) is depicted in Fig. 4b. Similar to Nu/Nu_0, f/f_0 tends to increase with decreasing *d/H*, as the flow blockage becomes more significant. It should be noted that the use of the DBB-Ts leads to substantial increase of friction factor in comparison with that of the smooth channel with no DBB-Ts. The high turbulence intensity in a core region due to flow blockage is a major factor to cause an extreme pressure drop.

Figure 5 presents the thermal enhancement factor (TEF) for air flowing in the channel with different open corner ratios (*d/H*) and Reynolds numbers. In the figure, the thermal enhancement factor tends to increase with the decreasing open corner ratio (*d/H*) and Reynolds number. The enhancement factor with all open corner ratios (*d/H*) is found to be higher than unity for all Reynolds numbers. This indicates that the channels with DBB-Ts are advantageous with respect to smooth channel without DBB-Ts. For the range investigated, the DBB-Ts with d/H = 0.0 gives the maximum thermal enhancement factor or overall thermal enhancement at the lowest Reynolds number. This suggests that the use of DBB-Ts is feasible at small open corner ratios (d/H = 0.0-0.02) and low Reynolds number.

5. Summary

Numerical results of performance of turbulent heat transfer in a channel fitted with diagonal broken V-baffles in form of tail-end cut (DBB-Ts) are reported. The results reveal that heat transfer rate, friction factor as well as thermal enhancement factor increase as open corner ratios (*d/H*) of DBB-Ts decreases. For the studied range, the DBB-Ts with open corner ratio, d/H = 0.0 give maximum heat transfer enhancement of about 535% and cause increased friction factor of 36.89 times as compared to those of the smooth channel. The maximum thermal enhancement factor of the channel with DBB-Ts with d/H = 0.0 is around 171% higher than that of the smooth channel under the constant pumping power criterion.

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