

Enhancing Heat Transfer in Turbine Blade Cooling Channels Using Perforated V-Shaped Ribs: A CFD Study

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ABSTRACT

This study numerically investigates the thermal and hydraulic performance of square cooling channels fitted with V-shaped ribs, emphasizing the effect of rib perforations. Three configurations were examined: solid V-ribs (VR), circular-perforated V-ribs (CPV), and semi-circular-perforated V-ribs (SCPV), each incorporating one to three holes. Simulations were conducted for Reynolds numbers between 5,000 and 15,000 using a three-dimensional, steady-state, incompressible flow model with the RNG- $k\epsilon$ turbulence model. Performance was evaluated through local Nusselt number distributions, Nu/Nu_0 , friction factor ratio (f/f_0), and thermal enhancement factor (TEF). Results indicate that rib perforations reduce recirculation zones and improve fluid mixing, producing more uniform wall heat transfer. Compared with solid ribs, perforated designs consistently achieve higher Nusselt numbers and TEF values, with semi-circular perforations showing slightly superior performance to circular ones. Additionally, perforations help limit frictional penalties typically associated with rib turbulators, especially when multiple holes are used. Overall, rib perforation emerges as an effective approach for enhancing heat transfer while controlling flow resistance in turbine blade internal cooling.

Keywords: Cooling passages; Circular perforated; Gas turbines blade; Semi circular perforated; V-ribs

1. Introduction

Gas turbines operate at turbine inlet temperatures that are considerably higher

than the melting points of blade materials. To ensure the integrity and durability of turbine blades under these severe ther-

mal loads, internal cooling systems are indispensable. Among the available cooling techniques, the introduction of turbulence promoters such as ribs inside internal cooling passages has proven to be particularly effective. These ribs disturb the thermal boundary layer and generate secondary flows that enhance convective heat transfer; however, such benefits come with an inherent increase in flow resistance. As a result, the main design challenge is to achieve a high level of heat transfer enhancement while controlling the associated pressure losses, which are usually assessed in terms of the thermal enhancement factor.

In recent years, considerable attention has been given to the influence of geometric and flow parameters such as Reynolds number (Re), rib pitch-to-height ratio (p/e), blockage ratio (Br), and attack angle (α) on the cooling performance of ribbed passages. Yildiz et al. [1] demonstrated via CFD that rib cross-section (e.g., wedge, roof, slope, stepped) significantly affects Nusselt number distribution and friction factor—with sharper ribs yielding superior heat transfer but greater pressure losses. Wen et al. [2] highlighted how Coriolis induced secondary flows in rotating serpentine passages produce non uniform thermal patterns, necessitating rib design adaptation under rotation. Wang et al. [3] introduced cross scale ribs (combining macro and micro features), achieving a 10–15% increase in Nusselt number while reducing friction losses, thereby improving thermal enhancement at specified p/e (~ 10) and Br (~ 0.1).

Hybrid and novel rib configurations have also been explored. Xiao et al. [4] found that ribbed swirl cooling channels generate strong swirling secondary flows that yield more uniform heat transfer, even under rotating conditions. Say et al. [5] de-

veloped V spline ribs that alleviate structural stress concentration while preserving high thermal enhancement. Li et al. [6] introduced a parametric modeling framework enabling systematic exploration of α , p/e , and Br , confirming their pivotal roles in flow field development. Alrajhi [7] reported that recirculation zones downstream of ribs critically influence local heat transfer, with their size and intensity contingent on rib spacing and Re .

In addition to ribs, alternative surface modifications have demonstrated effectiveness. Mehrjardi & Mazaheri [8] showed that dimpled walls generate vortex patterns that boost heat transfer by $\sim 25\%$ while incurring lower drag than conventional ribs. Kim et al. [9] and Ruan et al. [10] investigated lattice structured ribs and U shaped perforated latticework, revealing that perforations promote bypass flow, reduce pressure drop, and maintain thermal uniformity. Schekman & Kim [11] combined perforated blockages with pin fins, achieving an $\sim 18\%$ rise in thermal performance via enhanced vortex shedding. Liu et al. [12] developed a triple layer cooling channel that improves coolant distribution and effectively mitigates thermal non uniformities.

Innovative rib layouts such as broken V ribs [13] and separated ribs with bypass gaps [14] have been shown to generate enhanced turbulence and flow bypass pathways, improving cooling while reducing localized hotspots. Zhang et al. [15] performed comprehensive parametric analysis indicating that rib pitch, Br , and α are decisive in determining the thermal enhancement factor, particularly within the transitional Re range of 5000–15000.

The influence of perforations within V shaped ribs in square channels remains insufficiently addressed. Limited studies have systematically examined how the

number (one, two, three), shape (circular vs semi circular near the wall), and arrangement of perforations affect flow behavior, heat transfer, and friction losses, particularly in the transitional flow regime. To address this gap, the present study conducts numerical simulations of a square cooling channel featuring V ribs on both top and bottom walls. Three configurations are examined—normal V ribs (VR), circular perforated V ribs (CPV), and semi circular perforated V ribs (SCPV)—each with one, two, and three holes. The study evaluates variations in Nu/Nu_0 , f/f_0 , and TEF to elucidate optimal rib perforation strategies for turbine blade internal cooling.

2. Geometry of Cooling Passages and V-Shaped Baffles

The cooling channel was designed with a square cross-section. V-shaped ribs were mounted on both the upper and lower heated walls, as illustrated in Fig. 1, in order to generate a dominant vortex flow combined with a secondary motion similar to that of impinging jets. This combined flow behavior enhances heat transfer performance by reducing pressure losses while simultaneously improving thermal exchange efficiency. Different arrangements of V-shaped ribs were incorporated within the duct, as shown in Fig. 2. The computational domain consisted of a single-pass duct with a width (W) and height (H) of 30 mm, giving an aspect ratio (AR) of 1:1 and a hydraulic diameter (D_h) of 30 mm. The configuration parameters included an attack angle (α), pitch ratio ($PR = p/H$), and blockage ratio ($BR = b/H$), which were set to 30° , 1.0, and 0.1, respectively. The diameter of the perforations was chosen as half of the rib thickness ($d = 1.5$ mm). Simulations were carried out for turbulent flow conditions over a Reynolds number range

from 5,000 to 15,000.

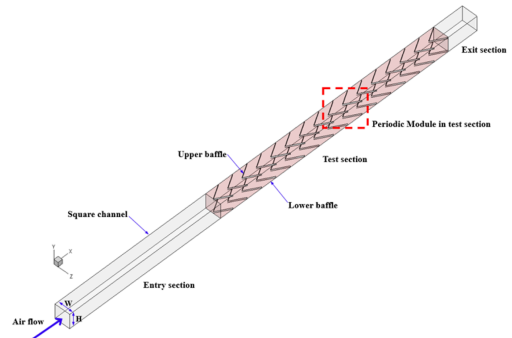


Fig. 1. Details of cooling passages.

3. Boundary Conditions

The computational domain and mesh setup are illustrated in Fig. 3. The periodic module was one quarter of it, symmetry on both of top and left wall. A nonslip condition was imposed on all walls. A uniform heat flux of 600 W/m^2 was applied to the bottom surfaces. The right-side walls were treated as adiabatic, and the heat conduction through these walls was considered negligible. The thermophysical properties of air at 300 K, was used as the working fluid in this study [16]. The Prandtl number (Pr) of the air was 0.707, while the density (ρ) was 1.1614 kg/m^3 . The specific heat capacity at constant pressure (C_p) was $1007 \text{ J/kg}\cdot\text{K}$, and the thermal conductivity (k_a) was $0.0263 \text{ W/m}\cdot\text{K}$. The dynamic viscosity (μ) of the air was $1.846 \times 10^{-5} \text{ kg/m}\cdot\text{s}$. These properties were assumed to be constant throughout the simulations. The inlet was defined as a velocity inlet based on the mean velocity corresponding to the specified Reynolds number, with a turbulence intensity of 5% imposed at the entrance. The outlet was treated as a periodic pressure boundary, where the pressure gradient was automatically adjusted to satisfy overall mass conservation within the computa-

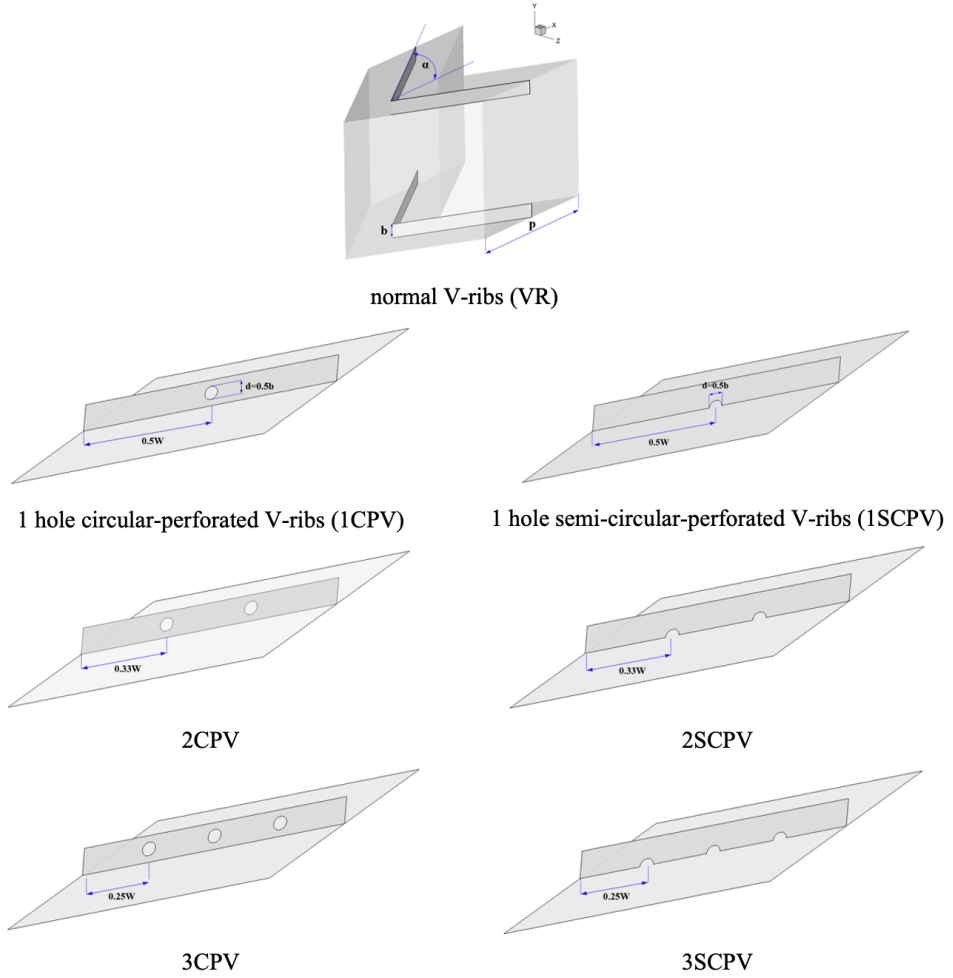


Fig. 2. V-shaped configuration, details of circular perforate ribs and semi-circular perforate ribs.

tional domain.

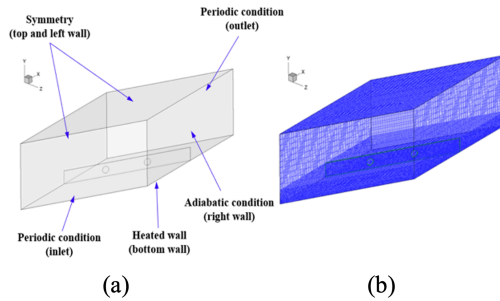


Fig. 3. (a) Computational domain and (b) mesh configuration.

4. Mathematical Model

The governing equations were employed to model the turbulent flow and associated heat transfer, assuming a steady-state, incompressible, three-dimensional flow field. In this numerical model, heat transfer was restricted to the convective mechanism only.

The governing equations were as follows: Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0. \quad (4.1)$$

Momentum equation:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} - \rho \overline{u'_i u'_j} \right) \right]. \quad (4.2)$$

Equation of Energy:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j} \left(\left(\frac{\mu}{\text{Pr}} - \frac{\mu_t}{\text{Pr}_t} \right) \frac{\partial T}{\partial x_j} \right), \quad (4.3)$$

where

$$\mu_t = \rho C_\mu k^2 / \varepsilon \text{ and } k = 0.5 \overline{u'_i u'_i}.$$

The RNG $k-\varepsilon$ turbulence model was utilized to address this problem and is described as follows:

$$\begin{aligned} \frac{\partial}{\partial x_i}(\rho k u_i) &= \frac{\partial}{\partial x_j} \left(\sigma_k (\mu + \mu_t) \frac{\partial k}{\partial x_j} \right) \\ &+ G_k + \rho \varepsilon \text{ for } k, \end{aligned} \quad (4.4)$$

and

$$\begin{aligned} \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) &= \frac{\partial}{\partial x_j} \left(\sigma_\varepsilon (\mu + \mu_t) \frac{\partial \varepsilon}{\partial x_j} \right) \\ &+ C'_{1\varepsilon} \frac{\varepsilon}{k} G_k - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \text{ for } \varepsilon, \end{aligned} \quad (4.5)$$

where the $C'_{1\varepsilon}$ can be expressed as

$$C'_{1\varepsilon} = C_{1\varepsilon} - \frac{\eta(1 - \eta/\eta_0)}{1 + \lambda\eta^3}$$

where

$$\eta = (2 \cdot E_{ij} \cdot E_{ij})^{1/2} \frac{k}{\varepsilon} \text{ and } E_{ij} = \frac{1}{2} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right).$$

The constant values were

$$C_\mu = 0.09, \sigma_\varepsilon = 1.3, \sigma_k = 1.0, C_{1\varepsilon} = 1.42,$$

$$C_{2\varepsilon} = 1.68, \eta_0 = 4.377, \lambda = 0.012.$$

The governing equations were discretized using the finite volume approach [18]. For the iterative solution, a residual convergence criterion of 10^{-9} was imposed on the energy equation, while a value of 10^{-5} was adopted for all other governing equations. The principal parameters analyzed in this investigation are outlined as follows: The Reynolds number:

$$\text{Re} = \rho \bar{u} D_h / \mu, \quad (4.6)$$

which $\bar{u}$ denotes the mean velocity The friction factor:

$$f = ((-dp/dx) D_h) / \left(\frac{1}{2} \rho \bar{u}^2 \right). \quad (4.7)$$

The local Nusselt number:

$$\text{Nu}_x = h_x D_h / k_a. \quad (4.8)$$

The average Nusselt number:

$$\text{Nu} = \frac{1}{A} \int \text{Nu}_x dA. \quad (4.9)$$

The hydraulic diameter of duct, D_h is defined as

$$D_h = (2 \times W \times H) / (W + H).$$

The thermal enhancement factor (TEF) was adopted as the key metric for performance evaluation [17]:

$$\text{TEF} = \left(\frac{\text{Nu}}{\text{Nu}_0} \right) / \left(\frac{f}{f_0} \right)^{\frac{16}{55}}_{\text{Re}}. \quad (4.10)$$

Nu_0 and f_0 were the values of the smooth rectangular duct only.

5. Results and Discussion

5.1 Numerical validation and grid independence

To validate the predicted average values of Nu/Nu_0 and f/f_0 , the present numerical results were compared with the

experimental data for a square duct fitted with V shaped ribs as reported by Turakarn et al. [19]. Since published computational results for rectangular channels with V shaped baffles remain limited, the simulated outcomes of this study for f/f_0 and Nu/Nu_0 in a rectangular duct incorporating flapped V baffles were assessed against those reference data as shown in Fig. 4. The figure demonstrates that the predictions obtained from the RNG $k - \varepsilon$ model agree well with the reported data, showing similar trends. The deviations between the numerical and reference values of f/f_0 and Nu/Nu_0 were approximately 5.3% and 7.6%, respectively. Based on this close agreement, the RNG $k - \varepsilon$ model was adopted for all subsequent simulations in this work.

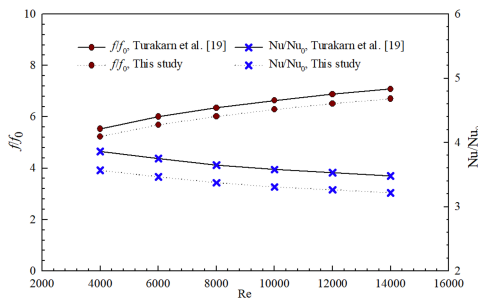


Fig. 4. Variation of Nusselt number and friction factor.

A grid independence study was conducted for the configuration with a single circular-perforated V-rib at $Pr=1$ and $Br=0.1$ under $Re=10,000$. Five mesh densities, consisting of 420,000; 570,000; 650,000; 720,000; and 850,000 cells, were tested using a periodic flow model. The results showed that further refinement of the grid beyond approximately 720,000 cells leads to only minor changes in both the friction factor and the Nusselt number. Therefore, a grid size of around 720,000 elements was adopted for subsequent simulations, as

it provides sufficient accuracy while maintaining computational efficiency.

5.2 Temperature distribution and flow regime

The bottom-wall temperature fields together with the corresponding flow streamlines for normal V ribs (VR) and 1 hole circular-perforated V ribs (1CPV) at Reynolds numbers of 5,000, 10,000, and 15,000, respectively, are illustrated in Figs. 5-6. These distributions highlight how the rib induced vortices interact with the recirculation zones that develop behind the ribs.

For the VR channel a dominant vortex structure is generated immediately after the rib apex. This primary vortex sweeps fluid over the heated wall and improves local mixing. Nevertheless, a low-velocity recirculation zone develops just downstream of the rib, which becomes a region where heat accumulates, as indicated by the red-colored zone. As the Reynolds number rises, the strength of the vortex and the reattachment of the flow downstream increase, which leads to an expansion of the lower temperature regions (green to blue areas) and a corresponding decrease in the high-temperature zones.

In contrast, the configuration with 1CPV introduces an additional jet flow through the circular opening. At lower Reynolds number, this jet emerges from the hole and impinges directly onto the downstream wall, breaking up the stagnant region and improving local cooling around the recirculation zone. When the Reynolds number is raised to 10,000 and 15,000, the jet flow becomes more energetic and interacts strongly with the main vortex. The combined effect leads to more uniform flow patterns, weaker dead zones, and a broader area of relatively low sur-

face temperature. The comparison of these two cases shows that the perforated V-ribs consistently reduce the extent of high-temperature regions and generate more efficient mixing, which improves the overall cooling effectiveness throughout the tested Reynolds number range.

5.3 Local wall Nusselt number

At a Reynolds number of 10,000, the local Nusselt number distribution along the heated wall on the bottom wall exhibits pronounced variations among the different rib configurations was shown in Fig. 7. In the case of the VR, the highest heat transfer rates are concentrated around the rib apex, forming distinct red regions at the leading edge and near the rib tips. In contrast, the central areas between ribs remain characterized by noticeably lower Nusselt numbers, indicating the presence of a stagnant flow zone that limits convective heat exchange in those regions. Introducing a 1SCPV through the rib changes this behavior markedly. The impinging jet emerging from the opening disrupts the recirculating flow behind the rib, reduces the dead zone, and enhances local mixing. As a result, the distribution of Nusselt number becomes more uniform, and the low-value regions behind the ribs are considerably reduced when compared with the solid rib case.

In the cases of 2CPV and 3CPV, multiple jets are generated, producing additional flow paths that help spread the coolant over a wider surface area. This effect further decreases the wall temperature and increases the Nusselt number across most of the heated wall. However, because the jet strength from each hole diminishes as the number of perforations increases, the heat transfer enhancement in these cases, although more uniform, is slightly less in-

tense than that of the 1CPV.

For ribs with semi-circular perforations 1SCPV, 2SCPV and 3SCPV, the flow pattern resembles that of the circular perforations, but the jets emerging from the semi-circular openings remain closer to the wall surface. This promotes stronger shear near the wall and leads to an extended region of high heat transfer downstream of each rib. Moreover, the flow between ribs becomes better mixed, particularly for the configurations with two and three semi-circular holes, resulting in a broader area of elevated Nusselt number. Overall, the results confirm that perforated ribs can effectively weaken dead zones, intensify mixing, and lead to a significant increase in the average heat transfer performance relative to the solid rib channel.

5.4 Comprehensive assessment of thermal performance

5.4.1 Heat transfer evaluation

The distributions of the averaged Nusselt number and the Nu/Nu_0 ratio obtained from the simulations clearly illustrate the heat transfer trends for each rib configuration. As shown in Fig. 8(a), the Nusselt number increases consistently with rising Reynolds number for all cases. For VR, Nu grows from about 77.35 at $Re = 5,000$ to 93.03 at $Re = 15,000$. In comparison, the 1CPV produces Nu values ranging from approximately 77.35 to 87.55 over the same range, while the 1SCPV yields Nu values from about 75.45 to 86.00. These results highlight the stronger turbulence and mixing generated at higher flow rates, which leads to more efficient convective heat transfer. It is evident that perforated ribs—both circular and semi-circular—achieve higher Nu than the solid configuration, with the single-hole designs showing the greatest enhancement. The

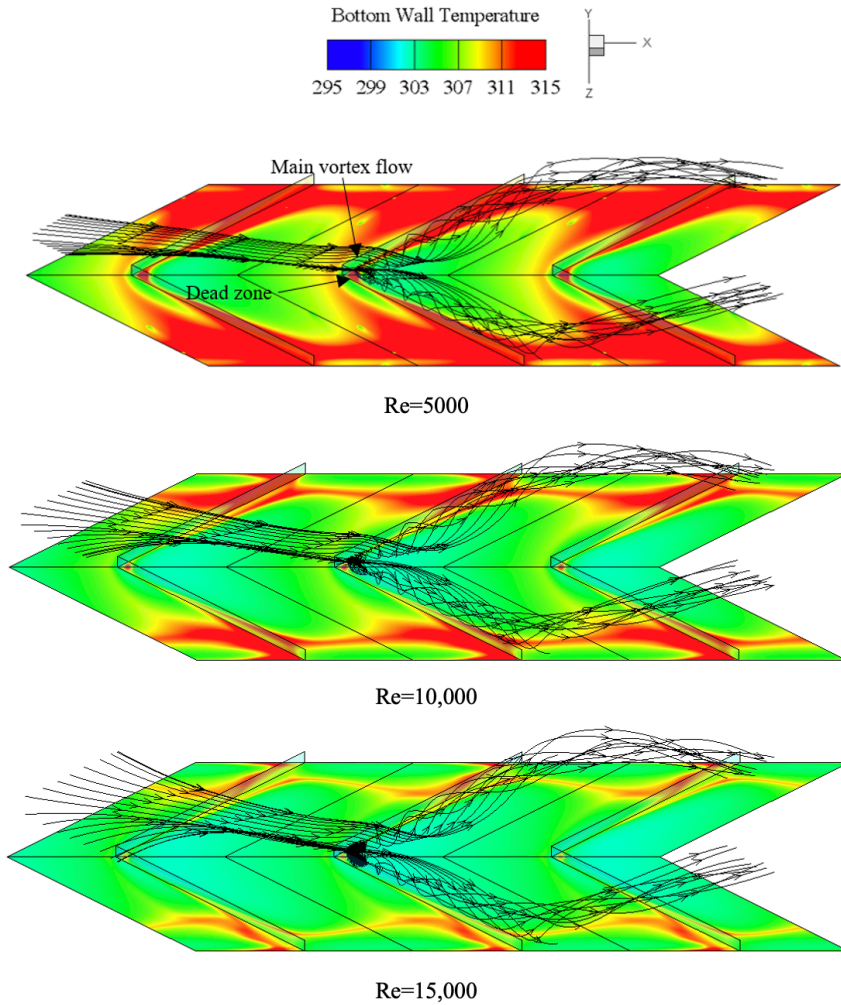


Fig. 5. The temperature distribution vs Re, V ribs (VR).

flow through the opening generates a jet that disturbs the wake region behind the rib, thereby reducing recirculation and enhancing fluid mixing.

The results presented in Fig. 8(b) show the Nu/Nu_0 ratio, which compares the thermal performance relative to a smooth channel. For all cases, Nu/Nu_0 remains significantly greater than that of a smooth wall, although the ratio decreases slightly as Reynolds number increases. For instance, in the solid rib configuration, Nu/Nu_0 decreases from 2.43 at $Re = 5,000$ to about

1.94 at $Re = 15,000$. In contrast, 1CPV decreases from 2.50 to 2.00 and 1SCPV from 2.48 to 1.99 over the same range. This decline occurs because the effect of the ribs becomes less dominant as the core flow velocity increases. Even so, perforated ribs consistently maintain higher Nu/Nu_0 than solid ribs. Configurations with two and three holes spread the cooling effect over a broader area but reduce the intensity of each jet, which results in a more uniform but slightly less effective enhancement compared with single-hole ribs. Over-

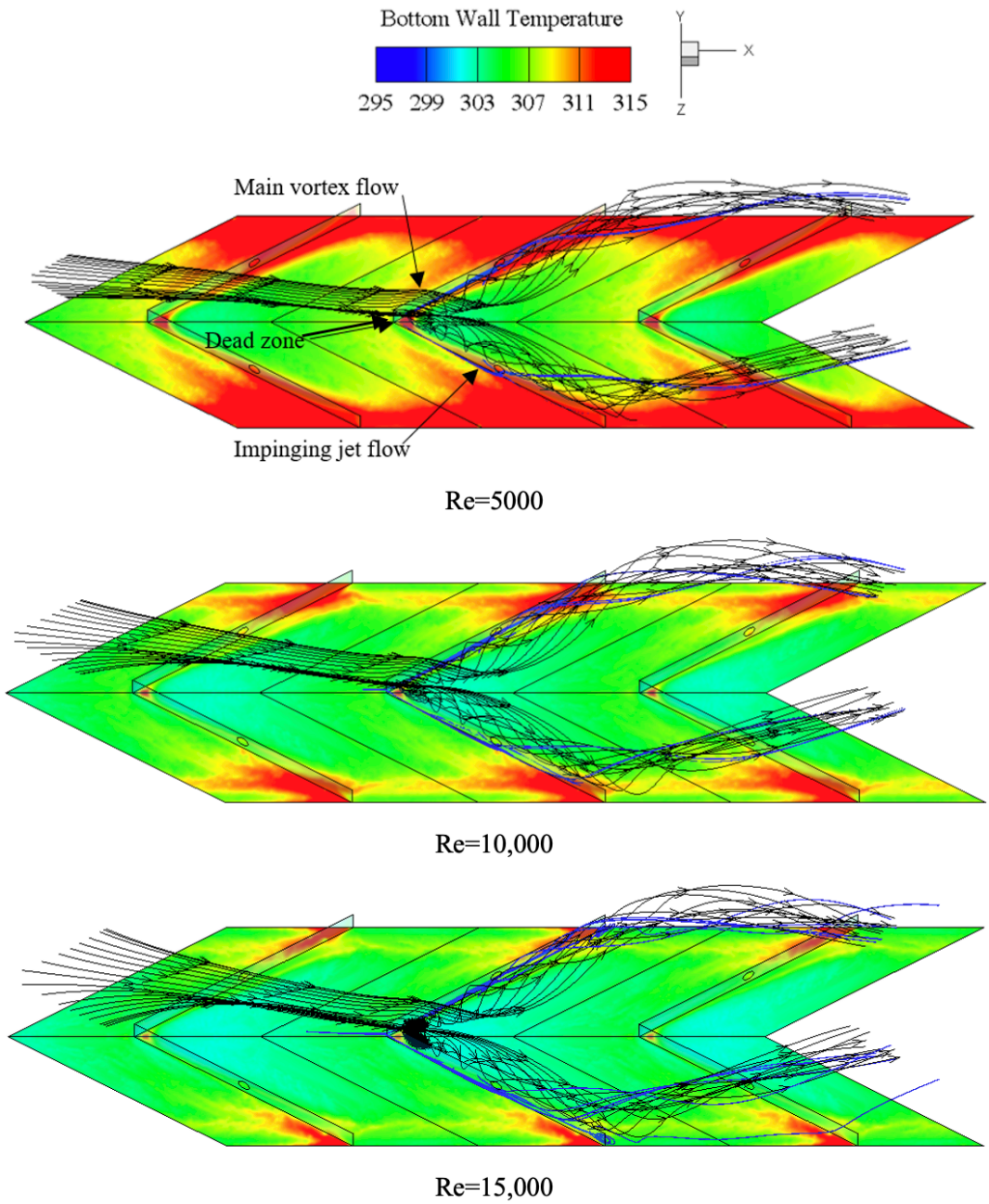


Fig. 6. The temperature distribution vs Re, 1 hole circular perforated V ribs (1CPV).

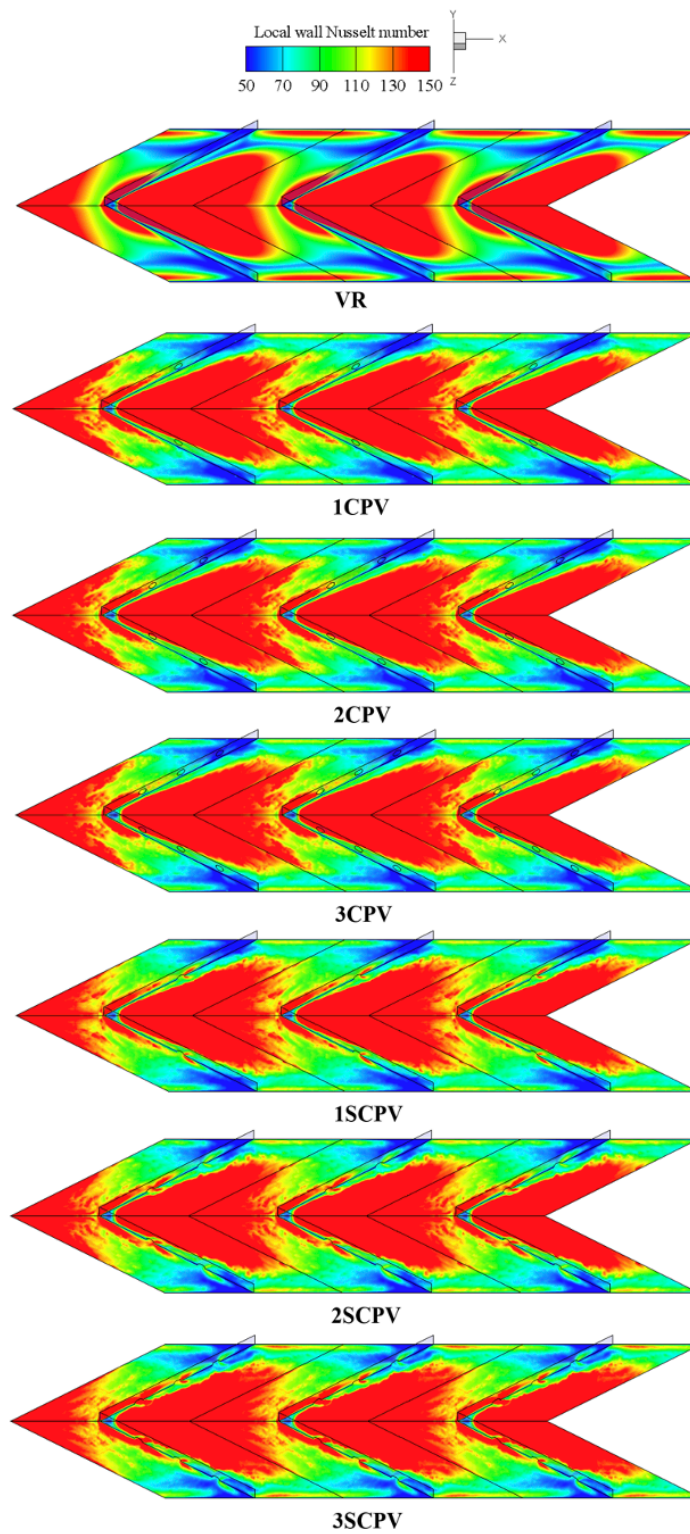


Fig. 7. The local wall Nusselt number contour of $Re=10000$ for the influence of model configuration.

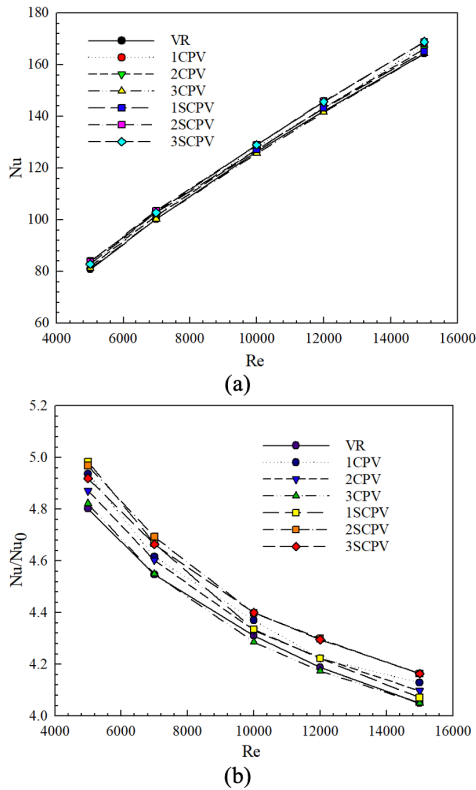


Fig. 8. Distributions of (a) Nu and (b) Nu/Nu_0 various Re.

all, these findings confirm that introducing perforations into ribs promotes stronger flow mixing, suppresses dead zones, and significantly improves the heat transfer performance of the ribbed channels.

5.4.2 Friction factor evaluation

The computed friction factor and its normalized ratio are illustrated in Fig. 9, where the rib configuration is shown to have a strong influence on flow resistance inside the channel. As the Reynolds number increases, the friction factor for all cases gradually decreases; however, the solid rib consistently produces the highest values throughout the examined range. Introducing perforations through the ribs allows a portion of the flow to pass through the openings, which weakens the wake region and

consequently lowers the pressure drag behind the ribs.

The variation of the f/f_0 ratio indicates that, at low Reynolds numbers, all configurations begin with values above 7.5 and rise further as the flow rate increases. The solid rib exhibits the largest increase, with the ratio ranging approximately from 8.0 to 9.0 as Re grows from 5,000 to 15,000. These results demonstrate that modifying the rib geometry by adding perforations does not only promote heat transfer, as discussed earlier, but also mitigates frictional losses within the channel. The benefit becomes more evident for multi-hole ribs, particularly under high-Reynolds-number conditions.

5.4.3 Thermal enhancement factor evaluation

The evaluation of the Thermal Enhancement Factor (TEF) for different rib configurations clearly demonstrates the significant influence of rib perforation on the overall thermal-hydraulic performance of the system. As illustrated in Fig. 10, the TEF of VR starts at approximately 2.43 at $Re = 5,000$ and gradually decreases as the Reynolds number increases. In contrast, ribs with circular and semi-circular perforations consistently maintain higher TEF values across the entire range. At $Re = 5,000$, the TEF for the 1CPV and 1SCPV cases exceeds that of VR by about 2.8% and 3.3%, respectively, and at $Re = 15,000$, the improvement remains at around 3.1% and 3.6%. Specifically, the TEF decreases from 2.50 to 2.00, 2.48 to 1.99, and 2.48 to 1.99 for 1CPV, 2CPV, and 3CPV, respectively. For the semi-circular perforations, the corresponding TEF values range from 2.51 to 2.01, 2.51 to 2.01, and 2.49 to 2.02 for 1SCPV, 2SCPV, and 3SCPV, respectively, all of which remain clearly higher

than those of the solid rib configuration.

When comparing the two perforation shapes, semi-circular perforations consistently yield slightly higher TEF values than their circular counterparts throughout the entire Reynolds number range. At $Re = 5,000$, this difference is about 0.4–0.6%, and at $Re = 15,000$, the difference increases slightly to 0.5–1%. This improvement can be attributed to the flow pattern induced by semi-circular holes, where jets tend to stay closer to the wall, intensifying wall shear and providing a more continuous enhancement of heat transfer downstream of the rib. The main mechanism behind these results lies in the jets passing through the perforations, which reduce recirculation zones and form drag while enhancing the mixing of the coolant. This combined effect improves heat transfer without a corresponding increase in frictional losses. In summary, all perforated rib configurations achieve higher TEF values than solid ribs, with semi-circular perforated ribs providing a slight but consistent advantage over circular perforations throughout the Reynolds number range considered in this study.

6. Conclusion

The present numerical study demonstrates the benefits of incorporating perforations into V-shaped ribs for internal cooling channels in gas turbine blades. The main findings can be summarized as follows:

1. Heat transfer performance

All perforated rib configurations achieved higher Nusselt numbers and Nu/Nu_0 ratios than solid ribs. Single-hole designs yielded the highest peak values, while multi-hole designs provided more uniform enhancement across the wall surface.

2. Flow structure

Perforations generate jets that inter-

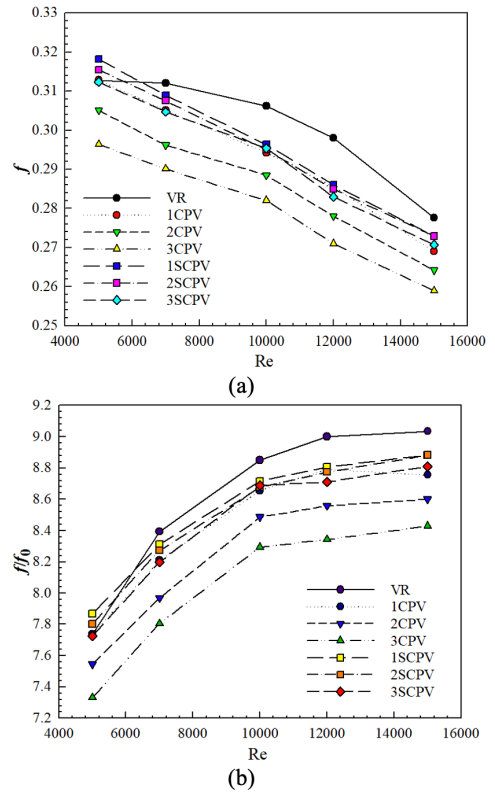


Fig. 9. Distributions of (a) f and (b) f/f_0 versus Re .

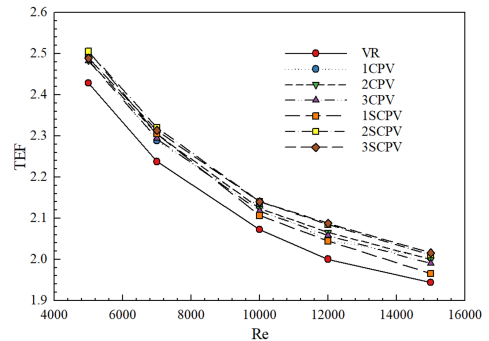


Fig. 10. TEF versus Re with various models.

act with the primary vortex, suppress recirculation zones, and promote mixing downstream of the ribs, which contributes to improved heat transfer uniformity.

3. Frictional effects

While ribbed channels inherently in-

crease frictional losses, the inclusion of perforations alleviates this penalty. Multi-hole configurations in particular produce lower f/f_0 ratios compared with solid ribs.

4. Overall performance

Evaluated using the thermal enhancement factor, perforated ribs consistently outperformed solid ribs across the entire Reynolds number range, with semi-circular perforations showing a 0.5–1% advantage over circular perforations.

5. Perforated ribs are feasible with current AM and laser-drilling technologies; however, future structural assessment is required.

In conclusion, perforated V-shaped ribs provide a balanced approach to increasing convective heat transfer while reducing excess flow resistance, making them a promising cooling design for advanced turbine blade systems.

Acknowledgements

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Appendix

Greek letters

α	attack angle, degree
ρ	density, kg m ³
δ_{ij}	Kronecker delta, m
σ_k	turbulent Prandtl numbers, (–)
σ_ε	turbulent Prandtl numbers, (–)
η	ratio of turbulence mean strain, (–)
ε	rate of turbulent dissipation, m ² s ³
μ	dynamic viscosity, kg m ¹ s ¹

Nomenclature

A	surface area, m ²
A_R	aspect ratio, (–)
B_R	blockage ratio, (–)
b	baffle height, m
$C_{1\varepsilon}$	turbulence model coefficient, (–)
$C_{2\varepsilon}$	constant of the turbulence model, (–)
C_μ	mean strain and rotation rate function, (–)
C_p	specific heat, J kg ¹ K ¹
D_h	hydraulic diameter, m
E_{ij}	tensor of mean strain rate, s ¹
f	friction factor, (–)
G_k	turbulent kinetic energy production by mean velocity gradients, kg s ³ m ¹
H	channel height, mm
h_x	coefficient of the local convective heat transfer, W m ² K ^{−1}
k	turbulence kinetic energy, m ² s ²
k_a	air thermal conductivity, W m ¹ K ¹
Nu	average Nusselt number, (–)
Nu_x	local wall Nusselt number, (–)
p	static pressure, Pa
P_R	pitch ratio, (–)
Pr	Prandtl number, (–)
P	distance between baffles, m
Re	Reynolds number, (–)
TEF	thermal enhancement factor, (–)
T	temperature, K
u	air velocity, W ms ¹
u_i	x_i –direction velocity component
u'_i	x_i –direction fluctuation velocity
u_j	x_j –direction velocity component
u'_j	x_j –direction fluctuation velocity
VG	vortex generator, (–)
W	channel width, mm
x_i, x_j	coordinate direction, m

Subscripts

0	smooth flat channel
a	air
t	turbulence
x	local

Superscripts

–	average
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