

Study of Gas Film Cooling under Large Eddy Simulation: Effect of Forward and Backward Injection on Gas Turbine Cooling Efficiency and Stability

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Abstract: In the design of advanced gas turbines, improving thermal efficiency and power output is crucial to achieve energy saving and emission reduction goals. This necessitates raising the inlet temperature, thereby highlighting the importance of developing efficient cooling technologies to ensure the longevity of hot components. Gas film cooling, as a widely used traditional cooling method, is subject to the influence of fluid dynamic characteristics, especially in unstable operating environments such as pressure fluctuations in the upstream combustion chamber and interactions between stators and rotors. This study utilized large eddy simulation to analyze the effectiveness of trench film cooling under such unstable conditions, with a specific focus on the time-averaged and transient characteristics of forward and backward injection of cooling air. The study investigated the efficacy of film cooling under pulsating flow conditions, with a blowing ratio of 1.5 under steady-state conditions, utilizing cosine and square waves as pulse boundaries, and a Strouhal number of 0.254. The research findings reveal that: 1) Contrary to the results under steady-state conditions, pulsating reverse injection led to a reduction of over 15% in time-averaged cooling efficiency and a noticeable decrease in the film coverage area compared to forward injection; this phenomenon is determined by the different flow mechanisms under steady-state and pulsating states. 2) Under a certain blowing ratio, the instability of gas film cooling is induced by the temporal fluctuations of near-wall vortices, which could result in rapid temperature changes and component failures. For pulsating forward injection, additional high film cooling unsteadiness outside the slot membrane orifices was observed. In the case of backward injection, pulsations did not lead to a significant increase in instability. These research findings provide a deeper understanding of the impact of gas film cooling on the cooling efficiency and stability of gas turbines, enabling gas film cooling technology to meet the inlet temperature requirements of advanced gas turbines, thereby reducing carbon emissions and contributing to the achievement of China's dual carbon goals.

Keywords: Large Vortex Simulation; Grooved Air Film Cooling; Pulsed Reverse Injection; Film Cooling Non-stationary Level.

1. Introduction

In September 2020, China announced its goal at the United Nations to strive to peak carbon emissions by 2030 and achieve carbon neutrality by 2060, highlighting the demand for energy efficiency and advanced technologies. Heavy-duty gas turbines play a critical role as high-efficiency clean energy sources, necessitating continuous development to meet these targets, which require temperatures exceeding 1700°C, surpassing current material limitations. This calls for the advancement of improved gas film cooling technologies to ensure turbine performance and support China's carbon reduction commitments. However, from a fluid dynamics perspective, the fundamental nature of gas film cooling falls within the realm of jet in crossflow (JICF) phenomena. Past studies on JICF have revealed a series of high-turbulence unsteadiness induced by jet shear layer vortices, counter-rotating vortex pairs (CRVP), horseshoe vortices, and other vortex structures.

Furthermore, in the actual operating environment of gas turbines, the injection velocity of cooling air is challenging to stabilize due to main flow pressure fluctuations in the upstream combustion chamber and interactions between stators and rotors [1]. This instability could lead to rapid temperature fluctuations on the cooling structure's surface, resulting in unexpected turbine component failures.

In conclusion, for a novel gas film cooling design, time-

averaged film efficiency can provide a general assessment of its cooling performance. However, the highly unsteady nature of gas film cooling characteristics demands additional requirements for unsteady analysis, which previous studies often overlooked by assuming steady-state conditions. Thus far, the practical cooling performance of trench film/backward injection film cooling (F/BIFC) in pulsating environments remains unresolved [2].

In order to examine the flow and heat transfer processes of cooling air injection under steady, cosine, and square wave conditions, this work uses a number of large eddy simulations. The cosine wave represents continuous pressure changes brought on by stator/rotor interactions, while the square wave represents transient full-on and full-off cooling air injections brought on by main flow pressure fluctuations. This study aims to clarify the effect of pulsating cooling air injection on the time-averaged and transient cooling performance of trench F/BIFC design, with a Strouhal number of 0.254 and a blowing ratio of 1.5 under stable conditions. The study will also provide a comprehensive reference for researchers and gas film cooling designers.

2. Calculation Methods

2.1. Geometric Model, Computational Domain and Boundary Conditions

The front/back injection grooved air film cooling model is

shown in Figure 1.

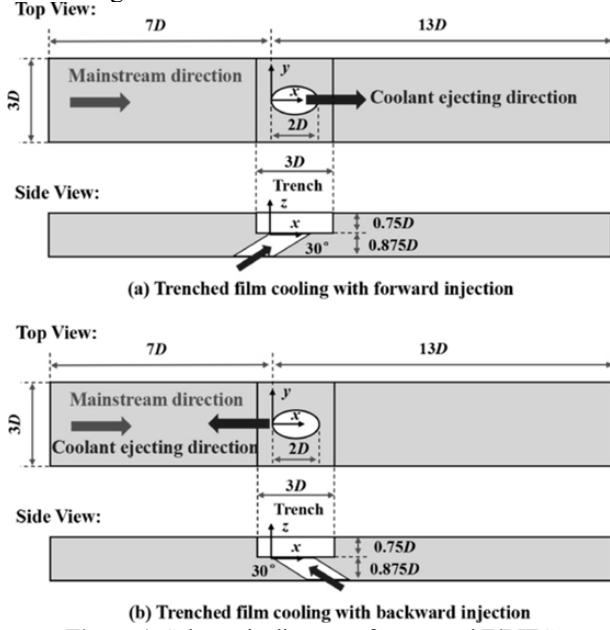


Figure 1. Schematic diagram of a grooved F/BIFC

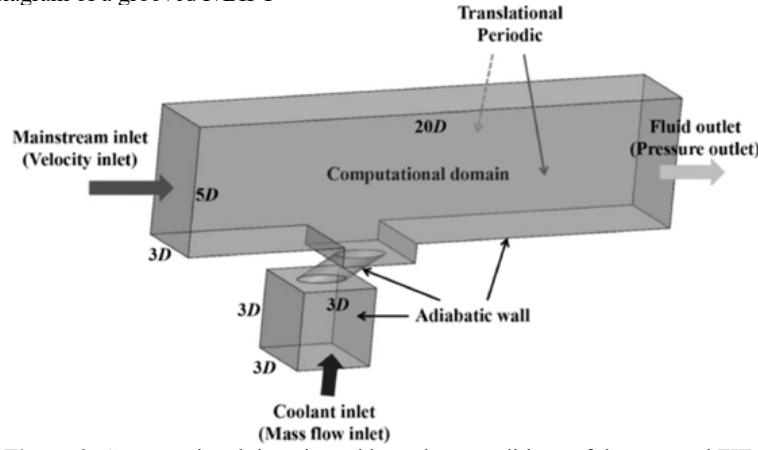


Figure 2. Computational domain and boundary conditions of the grooved FIFC

The primary flow channel, the grooved membrane orifice, and the coolant gas chamber make up the computational domain of the grooved FIFC. The coolant chamber's length, breadth, and depth are all equal to $3D$, whereas the main flow channel's height and width are $5D$ and $3D$, respectively [3]. The fluid in the computational domain is set to air, which is an ideal gas. One atm is the operating pressure used in this experiment. The gas's specific heat capacity (C_p) and thermal conductivity (λ) are determined using a polynomial function of temperature, while Sutherland's equation is used to compute the dynamic viscosity. The following illustrates the precise thermal characteristics of air.

$$C_p = 1.0151 \times 10^3 - 3.5186 \times 10^{-1}T + 9.0458 \times 10^{-4}T^2 - 5.8145 \times 10^{-7}T^3 + 1.2665 \times 10^{-10}T^4 \text{ (J} \cdot \text{K}^{-1} \cdot \text{kg}^{-1}) \quad (1)$$

$$\lambda = -9.3851 \times 10^{-4} + 1.0241 \times 10^{-4}T - 4.6304 \times 10^{-8}T^2 + 1.3228 \times 10^{-11}T^3 \text{ (W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}) \quad (2)$$

$$\mu = 1.716 \times 10^{-5} \left(\frac{T}{273.11} \right)^{1.5} \left(\frac{384.07}{T+110.96} \right) \text{ (N} \cdot \text{s} \cdot \text{m}^{-2}) \quad (3)$$

The steady-state boundary conditions were matched to the rest of the experiment, as shown in Fig. 2. Periodic conditions are imposed on both sides of the main flow channel. The other walls are adiabatic. At the mainstream inlet, a non-uniform Dirichlet velocity profile in the z -direction without velocity uptake, i.e., $u = u_\infty \left(\frac{z}{\delta} \right)^{\frac{1}{5}}$, is used, taking into account the

A standard TFC design with the same geometrical dimensions and the coordinates' origin at the membrane aperture's upstream edge. The membrane aperture measures $0.875D$ in height, 30° inclination angle, and $D = 12.7$ mm in diameter. The model has a thickness of $1.625D$, a total length of $20D$, and the same width in three dimensions. The model's inlet and outflow are located 7 and $13D$, respectively, from the origin. The membrane aperture in question is set inside a transverse trench that measures $3D$ in length and $0.75D$ in height. The model is $0.625D$ in length and $0.75D$ in height. The direction of cooling air injection differs between the grooved F/BIFC; it is forward in one and backward in the other, as seen in Figure 2.

effect of the boundary layer, where δ is the thickness of the boundary layer, and u_∞ is the mean velocity of the mainstream, which is a given value (20 m/s). The main flow Reynolds number (Re_∞) is based on $u_\infty = 20$ m/s and the hydraulic diameter of the main flow entrance. This result is consistent with earlier research. At the intake, the cooling air is configured as a mass inflow port. With the mainstream and cooling air intake temperatures of 300 K and 150 K, respectively, the genuine gas turbine air film cooling environment was found to have a more realistic density ratio ($DR = 2.0$), in line with the experimental circumstances [4]. The findings of matching BR and DR often mirror the features of the membrane cooling design based on prior research work, even if these lower temperatures cause the working fluid to have different thermal properties and the radiation impact is ignored at higher temperatures. Under non-reflective boundary circumstances, the fluid outlet is configured to be the pressure outlet. According to Fig. 3.

The mass flow rate in the steady cooling air injection scenario is 4.47×10^{-3} kg/s, meaning that a BR of 1.5 is achieved. The square and cosine waves at the same 400 Hz frequency are provided in the pulsed situation, which translates to a Strouhal number (St) of 0.254 . For a single period of time, the relative coolant mass ratios for the steady state, cosine, and square waves are $2:1:1$. This indicates that the quantity of coolant air injected is the same in both pulsed

situations, but only half of that in the steady state case [5].

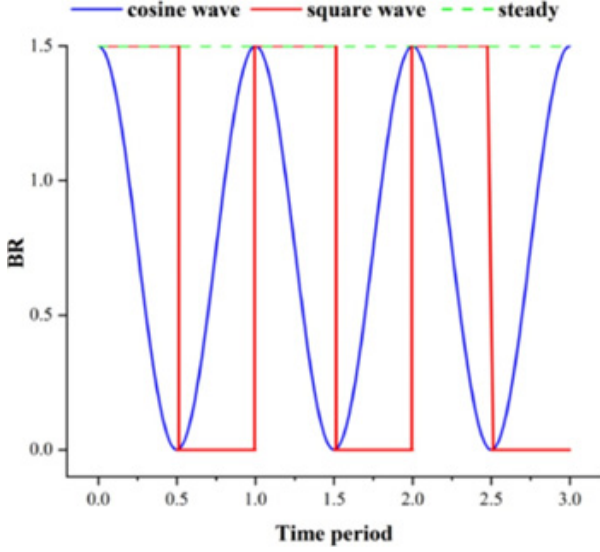


Figure 3. Variation of blowing ratio for steady and pulsed cases

2.2. Control Equations for the LES

By solving the transient problem, a large vortex simulation was carried out, corresponding to the Navier-Stokes governing equations for mass conservation.

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho \bar{u}_i) = 0 \quad (4)$$

Conservation of momentum

$$\frac{\partial}{\partial t}(\rho \bar{u}_i) + \frac{\partial}{\partial x_i}(\rho \bar{u}_i \bar{u}_j) = \frac{\partial}{\partial x_j}(\sigma_{ij}) - \frac{\partial \bar{p}}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j} \quad (5)$$

where σ_{ij} is the stress tensor and τ_{ij} is the subgrid scale stress.

Conservation of energy.

$$\frac{\partial \rho \bar{h}_s}{\partial t} + \frac{\partial \rho \bar{u}_i \bar{h}_s}{\partial x_i} = \frac{\partial \bar{p}}{\partial t} + \bar{u}_j \frac{\partial \bar{p}}{\partial x_j} + \frac{\partial}{\partial x_i} \left(\lambda \frac{\partial \bar{h}_s}{\partial x_i} \right) - \frac{\partial}{\partial x_j} [\rho (\bar{u}_i \bar{h}_s - \bar{u}_i \bar{h}_s)] \quad (6)$$

where λ and h_s denote the thermal conductivity and sensible enthalpy, respectively. In Eq. (5), the subgrid scale stress τ_{ij} is calculated as follows.

$$\tau_{ij} - \frac{1}{3} \tau_{kk} \delta_{ij} = -\mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \quad (7)$$

where τ_{kk} is the isotropic part of the subgrid-scale stress and μ_t is the subgrid-scale turbulent viscosity. The Wall Adaptive Local Eddy Viscosity (WALE) subgrid-scale model is used in this study to forecast the unresolved stresses because of its strong asymptotic wall behavior prediction for wall confined flow. Prior research has shown that for blade suction-side flow and heat transfer, the WALE model outperforms the Smagorinsky and KEM models. The flow and heat transfer performance for the air film cooling issue with high likelihood of flow separation was also effectively predicted using the WALE closure model. Furthermore, in the sections that follow, experimental data will be used to further confirm the numerical findings from the LES using the WALE model. inside the WALE framework.

$$\mu_t = \rho L_s^2 \frac{(s_{ij}^d s_{ij}^d)^{\frac{3}{2}}}{(\bar{s}_{ij} \bar{s}_{ij})^{\frac{5}{2}} + (s_{ij}^d s_{ij}^d)^{\frac{3}{2}}} \quad (8)$$

where L_s and S_{ij}^d are defined by the following equation.

$$L_s = \min(kd, C_w V^{1/3}) \quad (9)$$

$$S_{ij}^d = \frac{1}{2} (\bar{g}_{ij}^2 + \bar{g}_{ji}^2) - \frac{1}{3} \bar{g}_{ij} \bar{g}_{kk}^2, \bar{g}_{ij}^2 = \frac{\partial \bar{u}_i}{\partial x_j} \quad (10)$$

In the WALE model, C_w is set to a constant 0.325, k to a constant 0.4187, d to the distance to the nearest wall, and V to the volume of the computational cell.

2.3. Discretization Method and Solution Setup

Using the least squares cell-based approach in the commercial program AN-SYS FLUENT, the velocity gradient and temperature gradient at the cell's center are spatially discretized in order to solve the control equations. The continuity and momentum control equations were written in the bounded centre difference format, while the energy equations were written in the second order windward format [6]. Bounded second-order implicit interpolation is used for the time term and second-order implicit interpolation is used for the pressure term. In order to ensure a high and appropriate temporal resolution, the physical time step is elaborately set to $2.5 \times 10^{-6} s$ according to $\Delta t = \frac{0.004D}{u_\infty}$. At each time step, the maximum number of iterations is set to 30 in order to ensure the convergence of the solution, and the relative (scaled) residuals should be less than 10^{-4} for the continuity, and less than 10^{-7} for the velocity and the energy. At the same time, the residuals for all the variables should be reduced by an order of magnitude 104. The initial conditions were obtained using numerical results from the RNG turbulence model at steady BR.

During the computations, the fluctuation in instantaneous area-averaged pressure at the bottom of the trench was observed in order to identify the data sampling procedure for correct time-averaged statistical results. The area-averaged gauge pressure for the pulsed cosine wave FIFC is shown in Figure 4.

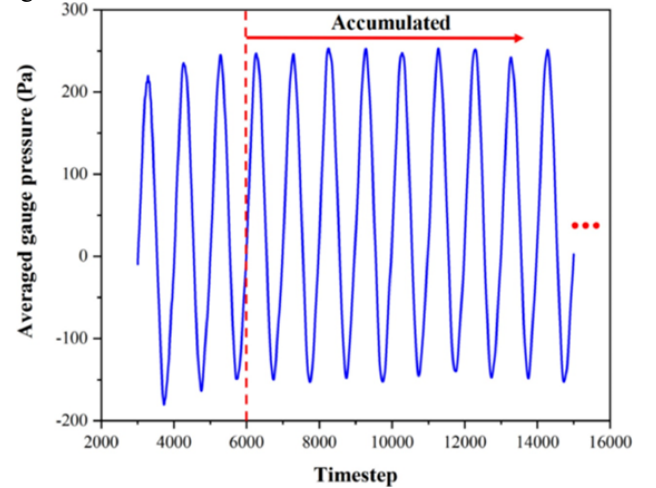


Figure 4. Transient area-averaged pressure at the bottom of the groove in cosine-pulsed FIFC

Figure 4 shows that after 6000 time steps, the average pressure varies on a regular basis. As a result, after 6000 time steps, the time-averaged data was collected. As seen in Fig. 5, two temporal and lateral averaged mem-brane cooling efficiencies may be derived by collecting the subsequent 15,000 and 16,000 data after 6,000 time steps.

The following formula defines the effectiveness of air film cooling.

$$\eta = \frac{T_\infty - T}{T_\infty - T_c} \quad (11)$$

where T is the local temperature and T_∞ and T_c are the mainstream and cooling air inlet temperatures, respectively. It is evident that for the course of the two cumulative time steps, the membrane cooling effect is almost constant. Therefore, gathering 16,000 time steps may provide trustworthy time-averaged data.

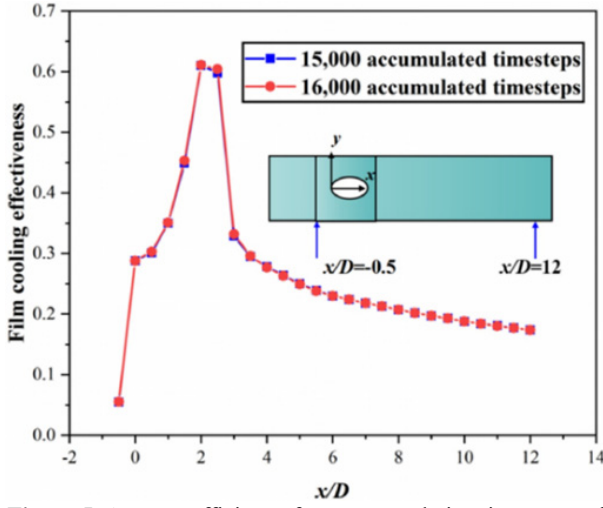


Figure 5. Average efficiency for two cumulative time steps of cosine-pulsed FIFC

2.4. Grid Generation Strategy and Grid Independence Study

The trench FIFC grid is shown in Fig. 6.

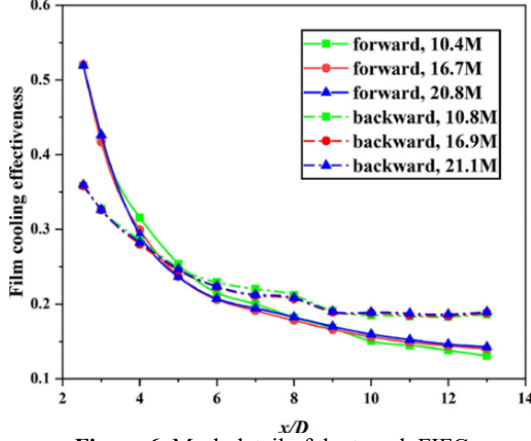


Figure 6. Mesh detail of the trench FIFC

Using ANSYS ICEM for mesh generation, a structural mesh comprising hexahedral cells was meticulously constructed. The strategy employed for the dissection of membrane holes was a tailored type grid approach, ensuring precision in the resultant mesh. Proximity to the boundary walls necessitated a delicate adjustment of the grid spacing for z^+ , which was calibrated to be under the threshold of 1 to accommodate the near-wall modeling requirements. Concurrently, in the streamwise and spanwise directions, the

grid spacings were respectively designated as $x^+ \approx 5$ and $y^+ \approx 7$, reflecting a strategic gradation to suit the flow physics. The mesh generation protocol adopted for the boundary interface flow condition (BIFC) mirrored this method, signifying a consistent application of meshing strategies across different aspects of the computational domain [7].

The fidelity and robustness of the numerical results were substantiated through a systematic grid independence study. This entailed a comparative analysis of the time-averaged and lateral-averaged air-film cooling efficiencies, denoted by 'a', across three distinct sets of grid strategies. The evaluations were conducted at a steady-state condition characterized by a blowdown cooling ratio (BR) of 1.5. The comparative outcomes of this study are illustrated in Figure 7, which consolidates the findings and underscores the negligible variance attributable to the refinement in grid resolution, thereby confirming the reliability of the numerical simulations.

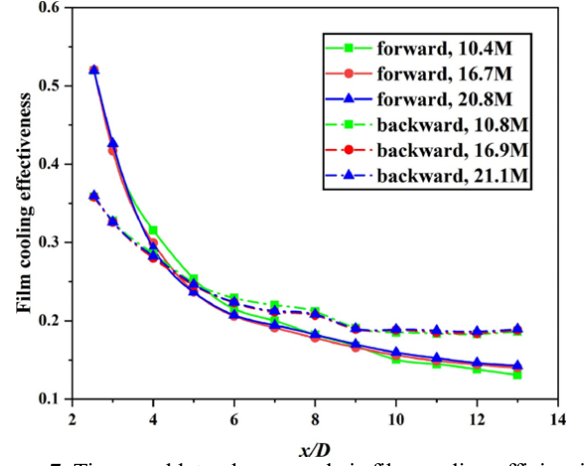


Figure 7. Time- and lateral-averaged air-film cooling efficiencies obtained for the three grid strategies

For FIFC, the difference in average effectiveness between the 16.7 million and 20.8 million grids is negligible, but there is a significant overprediction in the $3 < x/D < 7$, 10.4 million element grid, but an underprediction in the $x/D > 10$ region. For BIFC, the 16.9 million-element grid and the 21.1 million-element grid yield almost identical results. In order to balance the computational cost and accuracy, F / BIFC used a grid cell count of 16.7 million and 16.9 million in the following simulations, respectively.

3. Validation of the Numerical Strategy

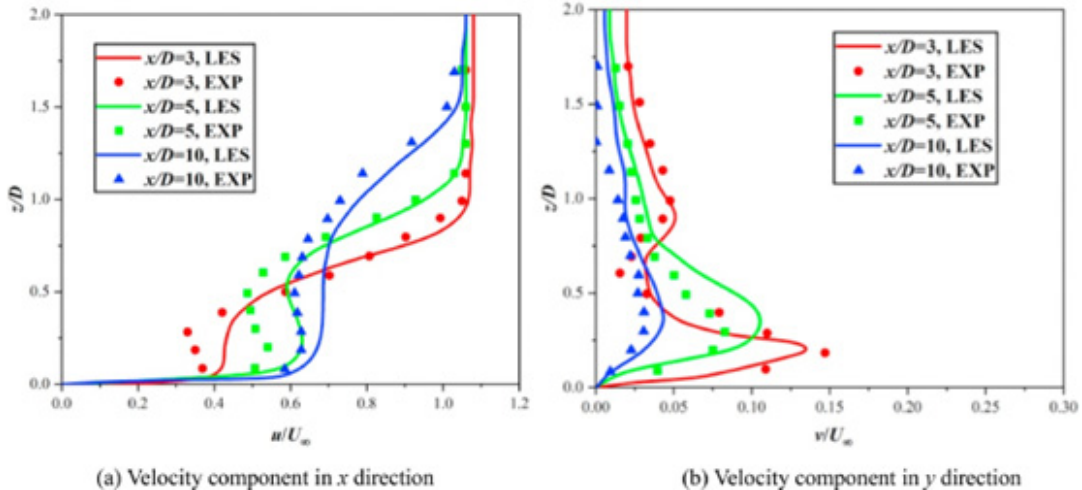


Figure 8. Time-averaged speed difference between experimental and LES data

To validate the above numerical strategy, we used the time-averaged velocity field obtained from the experimental data. Fig. 8(a-b) compares the flow and lateral velocity components (u/u_∞ and v/u_∞) at three locations ($x/D = 3, 5$ and 10) with the numerical results predicted by the LES method. In general, the numerical calculations agree well with the experimental data. Larger discrepancies are observed near $z/D = 0.3$, which may be due to velocimetry perturbations in the near-wall region. However, these differences are limited to an acceptable range [8]. Consequently, velocity field prediction may be accomplished using the current numerical method.

4. Results

4.1. Time-averaged Air Film Cooling Characteristics

Employing the numerically validated strategy, the study proceeded to prognosticate the time-averaged attributes of the trench film cooling, designated as F/BIFC, under a variety of conditions. These conditions included a steady-state blowdown cooling ratio (BR) of 1.5, as well as two distinct

pulsed cooling air injection modalities, specifically those following a cosine and a square wave pattern, both set at a frequency of 400 Hz, corresponding to a Strouhal number (St) of 0.254. The resultant air film cooling effectiveness on the target wall is depicted in Figure 9, which provides a visual representation of the thermal mitigation achieved through the trench cooling technique.

Analysis of Figure 9 elucidates several notable phenomena regarding the trench film cooling efficiency. Firstly, under a regime of steady cooling air injection, the grooved Blade Integrated Film Cooling (BIFC) configuration exhibits enhanced cooling efficacy. This is attributed not only to the augmented cooling efficiency within the groove itself but also to the expanded and more homogenous film coverage observed downstream of the groove. Secondly, when considering Full Film Cooling (FIFC), it is observed that, downstream of the groove, the film coverage under pulsed blowdown ratios (BRs) surpasses that of the steady-state BRs, resulting in a more homogenized temperature distribution as cited in the literature [9]. In stark contrast, within the BIFC scenario, the cooling performance under a stable BR exhibits superiority over that under pulsed BR conditions.

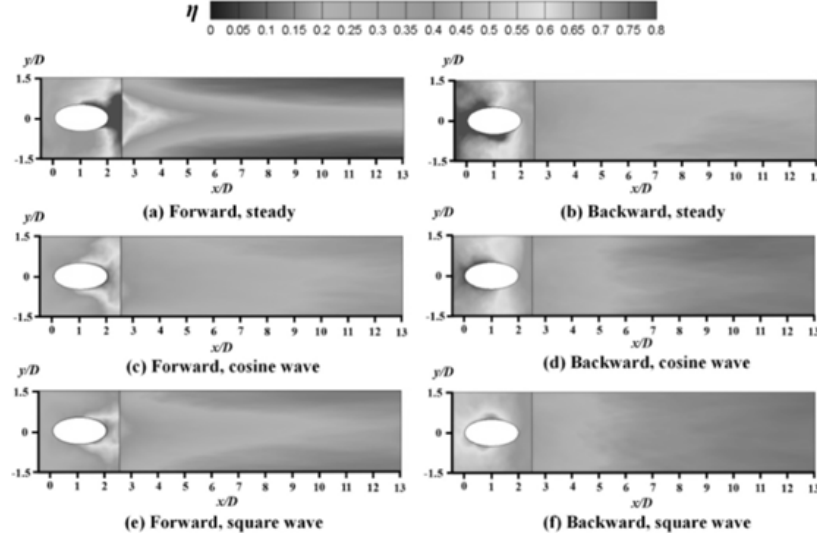


Figure 9. Time-averaged cooling performance in both pulsed and steady state BR

Thirdly, under pulsed conditions, FIFC generally outperforms BIFC; this is characterized by a higher degree of cooling and more extensive film coverage in the area following the trench, which stands in direct opposition to the observations made under steady BR conditions. To facilitate a quantitative comparison of the cooling performance exhibited by trench F/BIF under both steady and pulsed BRs, Figure 10 presents the time-averaged and spanwise-averaged cooling along the direction of the main flow.

From Fig. 10, the following phenomena can be observed: 1) Inside the trench, i.e., $-0.5 < x/D < 2.5$, in both cases of FIFC and BIFC, the membrane cooling effect of stabilised BR is higher than that of two-pulse injection. 2) Outside the trench, especially in the downstream region. $x/D > 4.5$, for FIFC, the cooling efficiency is the highest under cosine-pulsed BRs and the lowest under stabilised BRs. 3) For BIFC, both in and out of the trench, the cooling efficiency of stabilised BR is higher than that of two-pulse injection [10]. Fig. 11 (a-b) shows the time-averaged cooling efficiencies for cooling the wall on both lines $x/D = 3$ and $x/D = 6$.

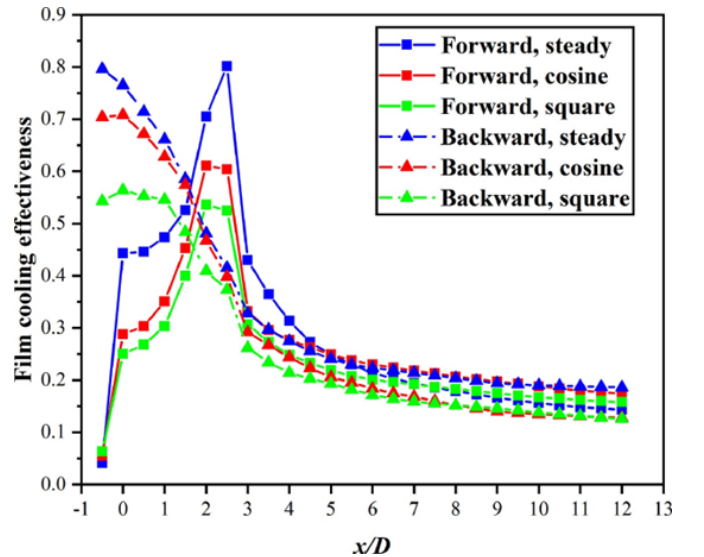


Figure 10. Temporal and lateral average cooling efficiencies of trench F/BIFC along the main flow direction under stabilised and pulsed BRs.

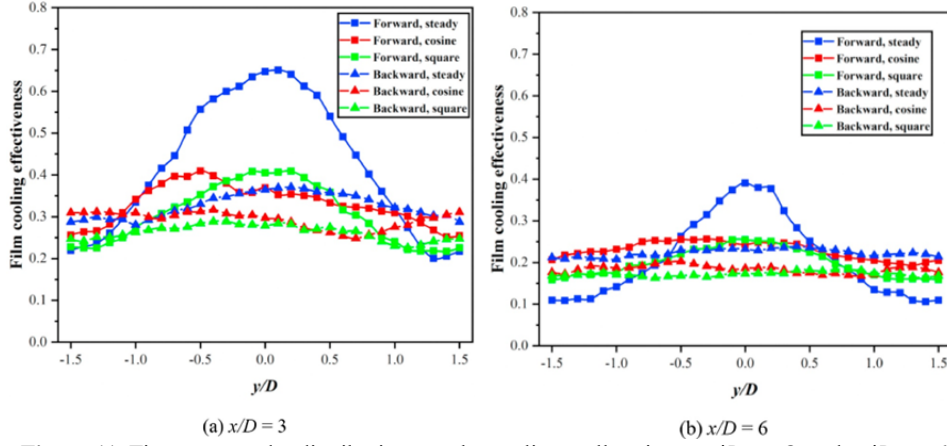


Figure 11. Time-averaged η distributions on the cooling wall at times $x/D = 3$ and $x/D = 6$

Under stable FIFC, the cooling efficiency peaks at the centre of $y/D = 0$ and a minimum occurs on both sides of $y/D = \pm 1.5$. The cooling efficiency is more uniform under stable BIFC. Under stable BIFC, the air film cooling effect is more uniform. The membrane cooling efficiency values of both pre-pulse spray methods are significantly higher than those of the post-pulse spray methods, indicating that the pre-pulse spray methods have better lateral membrane coverage performance under pulsed environments [11].

In elucidating the phenomena observed in Figure 9, an in-depth analysis was conducted on the flow dynamics and mixing properties of the cooling air with the mainstream. This

examination is crucial for understanding the mechanisms driving the differences in cooling performance between steady and pulsed injections. Figure 12 (a-f) presents a detailed representation of the flow structures and the corresponding time-averaged cooling efficiencies on the e-centre surface, comparing the effects of steady and pulsed injection methods. The visualizations within these figures are instrumental in discerning the intricate interactions between the injected cooling air and the high-temperature mainstream flow, thereby providing insights into the thermal protection efficacy of the film cooling technology employed.

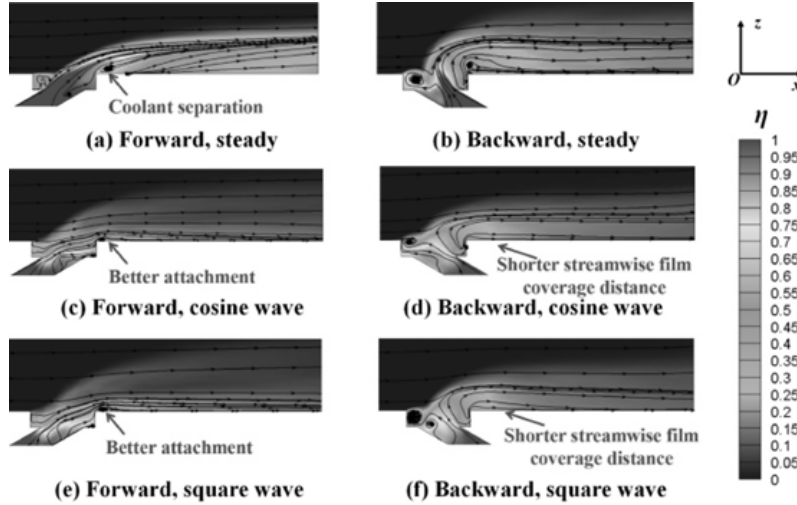


Figure 12. Flow and time-averaged efficiencies for profile $y/D = 0$

Figure 12(a-f) facilitates the observation of several critical phenomena regarding the interaction between cooling air and the mainstream under different film injection conditions. Firstly, it is evident that in Forward Injection Film Cooling (FIFC) scenarios, the trajectory of the cooling air largely aligns with the direction of the mainstream across all cases. Notably, under steady Blowing Ratios (BR), an ascension of coolant is observed downstream of the trench, a phenomenon which is markedly diminished in the pulsed injection scenarios, resulting in an enhanced attachment of the cooling air to the surface.

Additionally, concerning Backward Injection Film Cooling (BIFC), irrespective of the injection being stabilized or pulsed, the cooling air bifurcates upon exiting the film hole. At the trench corner, a portion of the air ascends to form a vortex, while the remainder descends, effectively filling the trench

with cooling air. This dynamic suggests a superior trench cooling performance in BIFC configurations.

Furthermore, the assessment reveals that downstream of the trench, the pulsed FIFC configuration benefits from a prolonged trajectory towards the film coverage distance, indicative of a more cooling air adherence when compared to the pulsed BIFC. This observation points towards the nuanced differences between pulsed injection strategies and their impact on film cooling.

The spatial distribution of flow lines and cooling efficiency for both sections $x/D = 3$ and $x/D = 6$ is shown in Fig. 13.

Under constant forward injection, a counter-rotating vortex pair (CRVP) appears downstream of the groove in Fig. 13(a), and the vortex pair becomes larger from $x/D = 3$ to $x/D = 6$. The vortex pair is not a large vortex pair, and the

vortex pair is not a large vortex pair. This phenomenon leads to severe cooling air rise and poor side film coverage. In the steady state, in Fig. 13(b), an asymmetric vortex pair exists, and the larger vortex carries the cooling air to diffuse along

the lateral direction, resulting in a better lateral film coverage and a higher average cooling efficiency as compared to the forward injection.

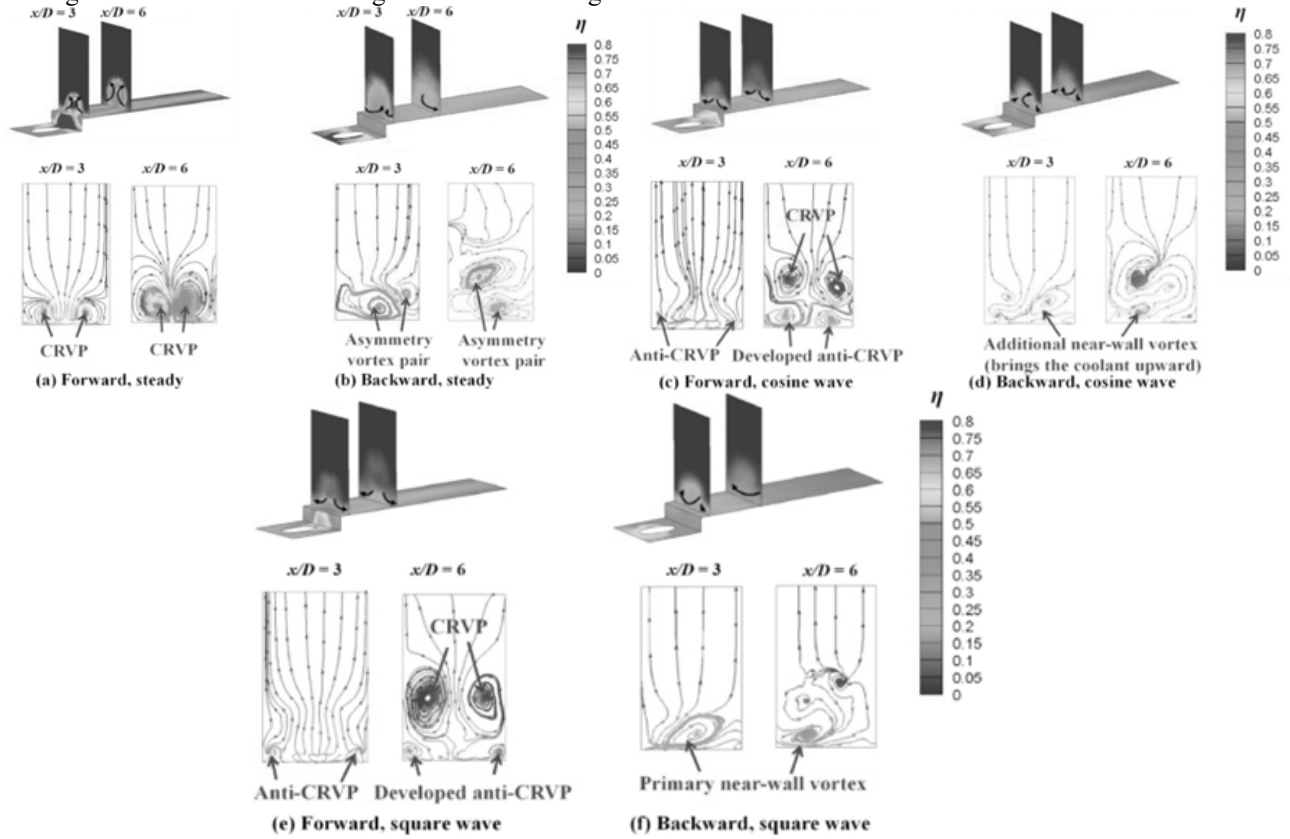


Figure 13. Time-averaged efficiency and streamline distribution at $x/D = 3$ and $x/D = 6$

Under the cosine pulse condition of FIFC, as shown in Fig. 13(c), the small anti-CRVPs at two corners gradually evolve into two pairs of large vortices: CRVPs and anti-CRVPs from $x/D = 3$ to 6. At $x/D = 6$, the anti-CRVPs occupy the entire near-wall region. As a result, the cooling air is suppressed down and adheres to the cooling wall. For the pulsed square forward injection in Fig. 15(e), the size of the anti-CRVP is smaller than that of the cosine forward injection at $x/D = 6$; therefore, the relatively low time-averaged film cooling efficiency is justified.

In the cosine pulse condition with BIFC, two asymmetric vortices appear in embryonic form in the $x/D = 3$ -section. When using cosine-pulse injection, Fig. 13(c) and (d) reveal a second near-wall vortex that pulls the cooled air upward and combines it with the main flow, indicating that the average efficiency of BIFC is lower than that of FIFC. When comparing the F/BIFC example of square pulse wave, as shown in Figs. 13(e) and (f), the average cooling efficiency of reverse injection is also lower than that of forward injection. The near-wall vortex in Fig. 15(f) is the cause of this, since it raises the cooling air upward.

A comparison of cooling wall time and area-averaged cooling efficiency ($2.5 < x/D < 12$) for the F/BIFC case under steady/pulse injection is shown in Figure 14.

In a comparative analysis, the data illustrates that the peak area-averaged film cooling efficiency is attained at 0.229 for the cosine forward injection approach. When operated under constant injection parameters, the reverse injection technique marginally enhances the area-averaged efficiency, elevating it from 0.225 to 0.227. Contrastingly, the Backward Injection Film Cooling (BIFC) design experiences a notable decrement

in efficiency under pulsating conditions. Specifically, the area-averaged efficiency undergoes a reduction of 20.9% and 15.7% for pulsed cosine and square Blowing Ratios (BRs), respectively. This suggests that while reverse injection can improve efficiency in steady-state conditions, the BIFC configuration is more susceptible to efficiency losses when subjected to pulsed injection, particularly with varying temporal profiles of the BR.

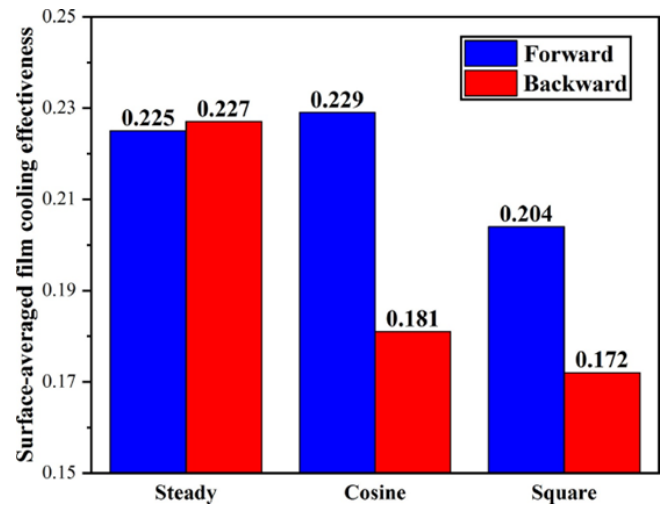


Figure 14. Comparing the target wall's area averaged η values over time and in various scenarios

4.2. Variable Properties of Temperature and Flow Fields

In the context of turbulent flow and the complexities

introduced by pulsed cooling air injection, an in-depth examination of the transient flow and temperature distributions within the pulsed Forward/Backward Injection Film Cooling (F/BIFC) process is imperative. The coherent vortex structures that manifest at the termination of the pulse cycle are effectively discerned using the Q-criterion method, with Figure 15 elucidating these with clarity. The intricate turbulent structures are depicted with temperature gradations for enhanced visualization.

Figure 15(a) delineates the conventional air film cooling vortex regime, characterized by a prominent horseshoe vortex proximate to the leading edge of the air film aperture, a shear layer vortex formed at the juncture of the mainstream and cooling air, and an array of hairpin vortices trailing downstream of the grooves. This configuration stands in stark contrast to the scenario presented in Figure 15(b), where the influence of cooling air impingement at the upstream groove edge leads to a marked thickening of the hairpin vortices and a subsequent dissolution of the horseshoe and shear layer vortices.

Diverging significantly from the aforementioned structures, the vortex formations under cosine and square pulse conditions manifest unique characteristics. As depicted in Figures 15(c) through 15(f), dual trails spawned by hairpin vortices emerge downstream of the groove, suggesting a complex interplay between pulse injection patterns and the resulting aerothermal field. This nuanced understanding of vortex dynamics is crucial for optimizing film cooling performance under varied pulsing conditions.

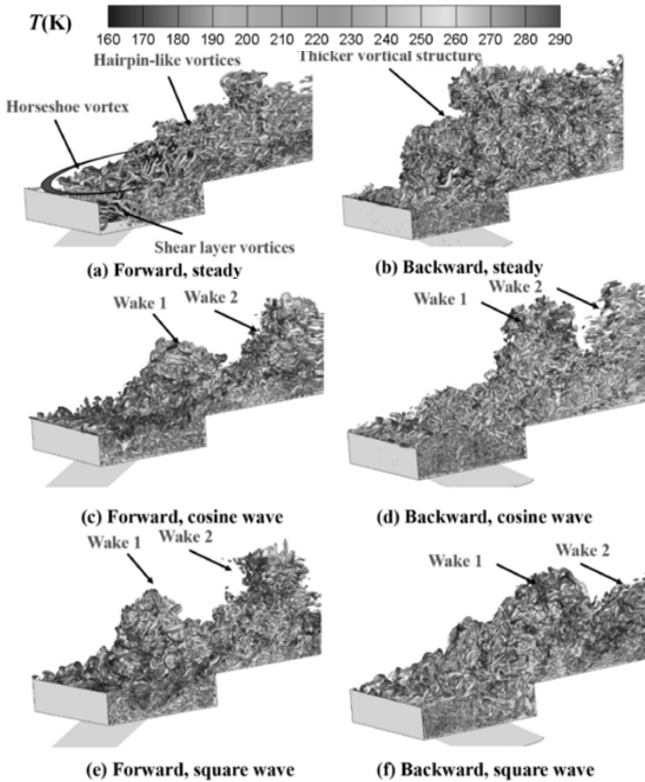


Figure 15. Coherent vortex structure identified at the end of the pulse cycle with a criterion coloured by the local temperature

This phenomenon is caused by the pulsed cooling air injection. When the cooling air volume is reduced, vortex valleys are generated. Figure 16(a,b) shows the temperature distribution in the centre plane at time $y/D = 0$ for the stable F/BIFC case. It can be seen that the transient mixing process between the cooling air and the main flow is very

irregular due to the presence of shear layer vortices, i.e., $K-H$ instability. For FIFC, the low-temperature cooling air is injected directly into the main stream, which reduces the cooling capacity. For BIFC, the coolant jet is split into two branches: one impinges on the upper edge of the groove and the other flows downstream.

Figure 16 (c-f) shows the time evolution of the coolant-mainstream mixture throughout the pulsation cycle. In Figs. 16 (c-d), discontinuous cooling flows are seen in both F/BIFC cases due to the effect of cosine pulse injection. At the beginning of the pulse cycle ($t = 0$), two separated coolant clouds A and B are observed. With the passage of time, cloud A moves along the main flow and spreads out towards the main flow. At the end of the cycle ($t = t_4$), cloud A occupies approximately the position of cloud B at $t = t_0$, and a new coolant air cloud emerges to replace the initial position of cloud A, forming a complete cycle. As shown in Fig. 18 ($t = 0$), for the cosine pulse FIFC, there is a slight entry of the main flow into the membrane pore at time c ; this then disappears at time $t = t_1$ and reappears at time $t = t_3$. The main flow of air into the membrane pore at time c is a very small amount. However, for the square-pulse FIFC shown in Fig. 16(e), the main stream ingress phenomenon is more severe; it is not observed only at time $t = t_2$. For the BIFC, it is noted that in the square case, the cooling air is rapidly dissipated downstream of the grooves, resulting in low cooling efficiency. In addition, the mainstream ingestion phenomenon is less severe for the BIFC compared to the FIFC.

Figure 17(a-f) shows snapshots of the transient film cooling effect at the cooling wall for $t = t_4$ -stabilised and pulsed F/BIFC conditions.

The transient behavior of cooling efficiency within the context of pulsed cooling air injection reveals a markedly less predictable pattern relative to the time-averaged outcomes, as visualized in Figure 10. An examination of 17(a) illustrates that, following the trench, an expanse is subjected to the continuous influence of a forward cooling air jet, which is conspicuously narrow in its spatial coverage. This contrasts with Figure 17(b), where the film cooling footprint expands substantially in the lateral dimension.

When analyzing pulsed Film Injection Film Cooling (FIFC) dynamics, as depicted in Figures 17(c) and (e), it becomes apparent that the transient coverage and cooling efficiency of the membrane are superior to those observed in the steady state condition. Conversely, the pulsed Backward Injection Film Cooling (BC) transient efficiencies, as shown in Figures 17(d) and (f), generally underperform relative to their steady state counterparts. Notably, the square wave injection yields the least results, with both the lowest efficiency and the minimal area of membrane coverage.

To encapsulate, the transient findings corroborate the cooling performance implications suggested by the time-averaged results; the method of injection exerts a consistent impact on cooling effectiveness. This accentuates the importance of selecting an appropriate pulsing strategy to optimize transient cooling performance, a critical consideration in advanced thermal management systems.

The temperature rms deviation (T_{RMSD}) is defined as in order to assess the cooling wall's non-stationary level in more detail.

$$T_{RMSD} = \sqrt{\frac{\sum_{i=1}^n (T_i - T_{ave})^2}{n}} \quad (12)$$

where T_i is the local temperature at each time step, T_{ave} is the time-averaged temperature, and n is the total number of

samples ($n = 16000$ in this work). A high T_{RMSD} corresponds to drastic wall temperature variations and strong air film cooling instability. Figure 18(a-f) shows the unsteady level of

T_{RMSD} at the cooled wall surface under steady and pulsed F/BIFC.

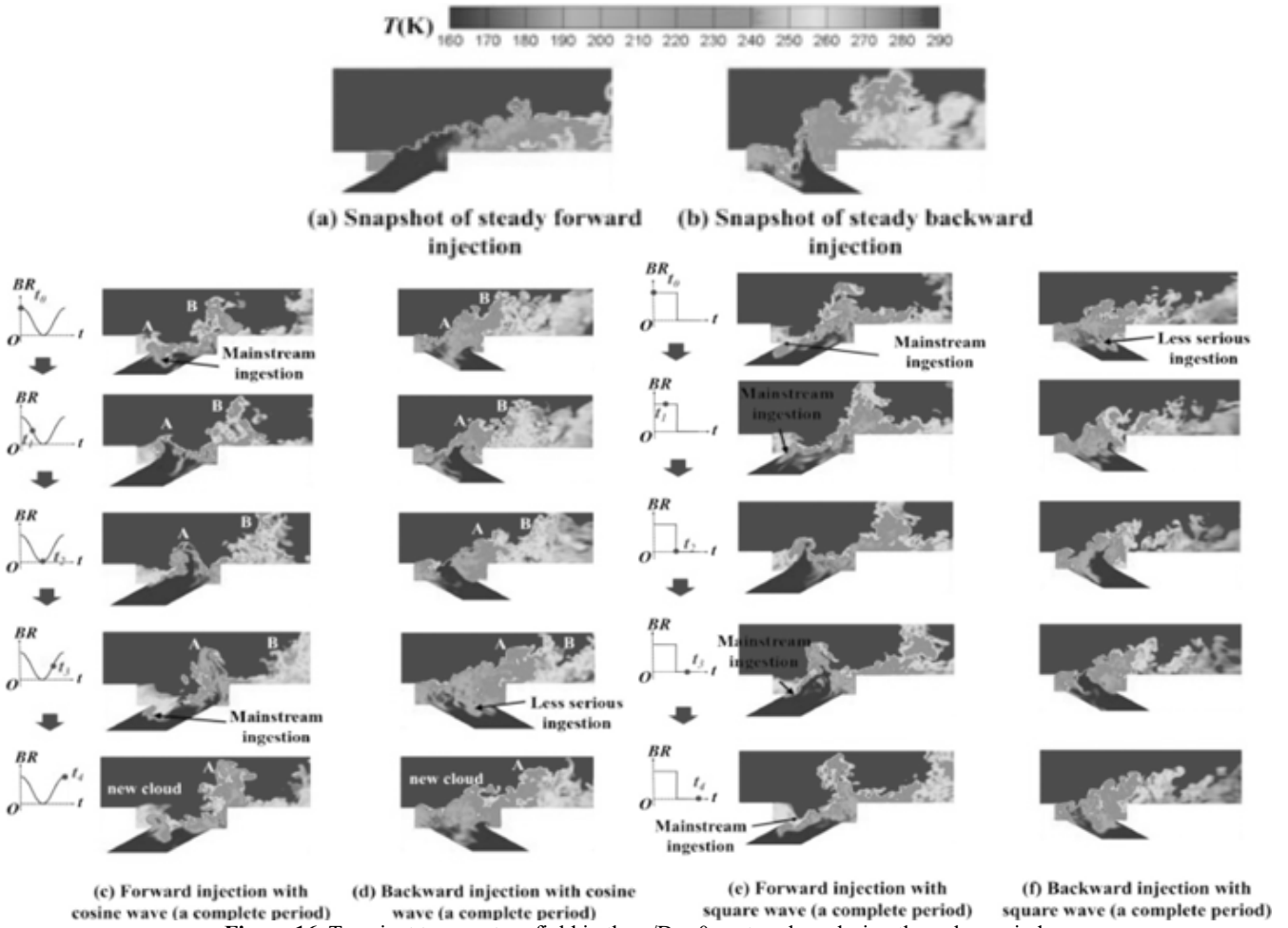


Figure 16. Transient temperature field in the $y/D = 0$ centre plane during the pulse period

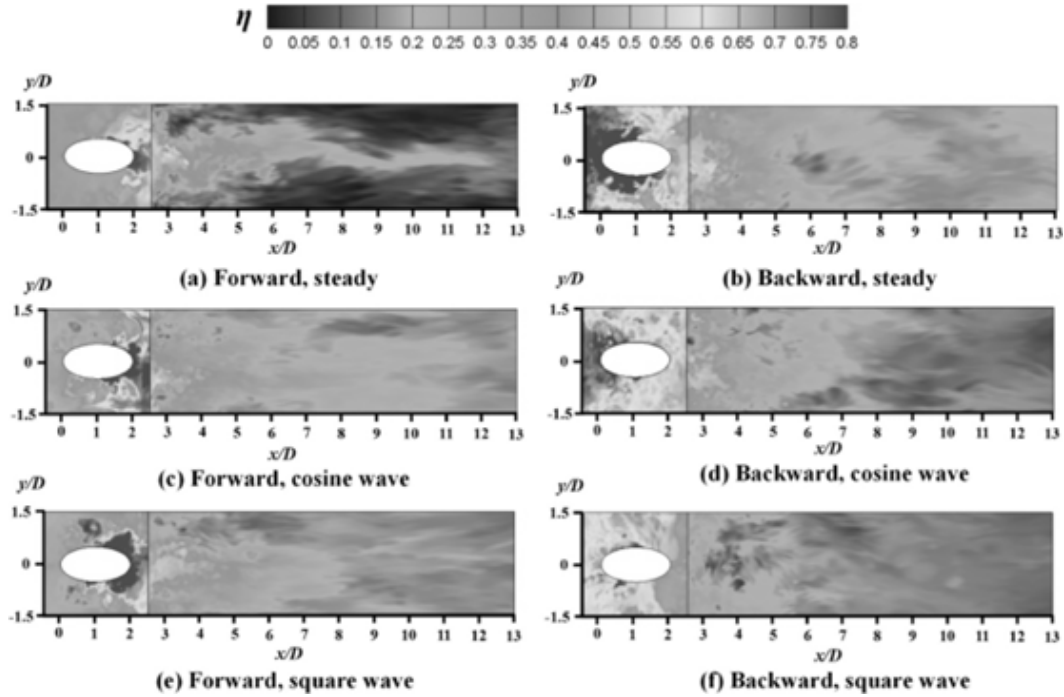


Figure 17. Transient efficiency of the cooling wall at the end of pulse period ($t = t_4$)

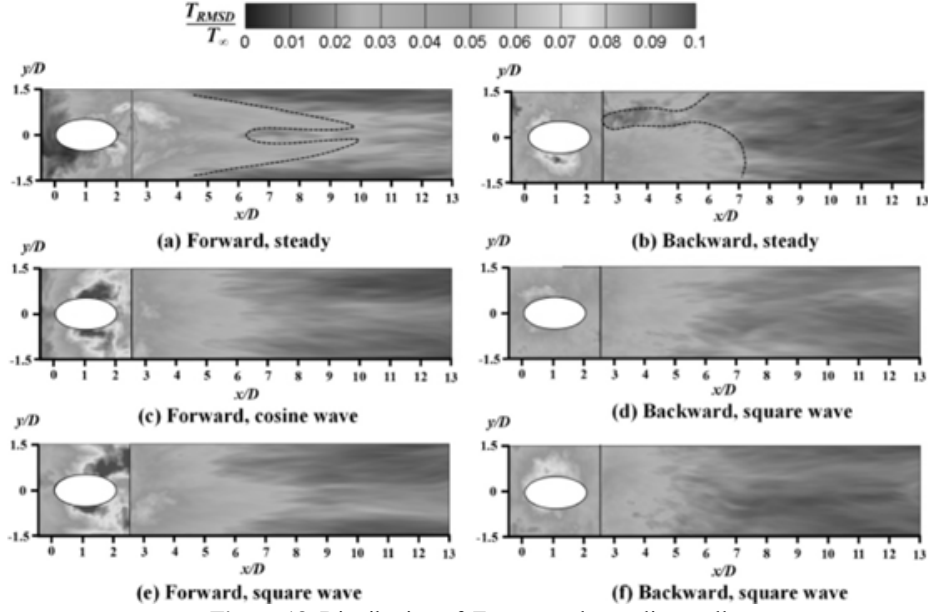


Figure 18. Distribution of T_{RMSD} on the cooling wall

Turbulence at steady state BR is the source of the instability in gas film cooling, as Fig. 18(a) and (b) illustrate. After the membrane pores, there are two comparatively high instability zones that are affected by the CRVP shown in Fig. 13(a-b). In FIFC, the two bands are often symmetrical, whereas in BIFC, they are asymmetrical. In Fig. 18(c) and (e), the T_{RMSD} distributions of the cosine-pulsed FIFC and the square-pulsed FIFC are similar. Inside the trench ($0.5 < x/D < 2.5$), the nonconstant level on both sides of the membrane pore is significantly increased due to the injection of pulsed cooling air. Whereas downstream of the trench, i.e., $x/D < 2.5$, the non-stoichiometric injection is at the same level as the stoichiometric injection; in region $x/D < 7$, the instability of the pulse injection is much lower. For BIFC, comparing the results in Fig. 18(b), (d) and (f), it can be found that the highest level of non-stereotypicality is observed for stable injection, while the lowest level of non-stereotypicality is observed for square pulse injection. In summary, for FIFC, the high nonconstant region caused by pulse injection is confined to the slot, and for BIFC, pulse injection does not lead to an increase in membrane cooling nonconstant.

5. Conclusion

Using both forward and backward cooling air injections, extensive large eddy simulations have been performed in this work to examine the time-averaged and transient features of trench film cooling under the influence of constant and pulsating blowing ratios of square and cosine waveforms. The thorough investigation produced significant findings about the effectiveness of various cooling techniques.

Time-averaged analysis reveals that, in comparison to the traditional trench-style forward injection film cooling (FC), the backward injection film cooling (BIFC) configuration greatly increases the film cooling efficiency under steady conditions, expanding the film-covered domain and producing a more uniform temperature field. However, the benefits of BIFC are lessened when blowing ratios are high. In particular, compared to FIFC, BIFC exhibits a decline in time-averaged and spatial film cooling effectiveness of 20.9% and 15. %, respectively, along with a notable contraction in the effectively cooled area. Divergent time-averaged flow

dynamics are the cause of this BIFC performance disparity between steady pulsating circumstances. The BIFC's asymmetric vortex forms aggressively increase the lateral film coverage at constant blowing ratios. On the other hand, FIFC gains from the formation of near-wall counter-rotating pairs (CRVPs) with pulsing ratios because they prevent coolant ejection in the direction of the wall, enhancing lateral film coverage. On the other hand, the cooling effectiveness is decreased when BIFC pulses because some of the coolant is swept upward by the near-wall vort.

The study of trench film cooling stability presents a new standard for evaluating cooling efficacy, suggesting that higher instability is associated with higher thermal danger. In two different zones of marked unsteadiness, the temporal oscillations of CRVPs primarily cause variability in gas film cooling under stable circumstances. The use of pulsing BR with FIFC increases the instability around the trench's film holes considerably. But in both constant and pulsating circumstances, the instability is equivalent or reduced downstream of the trench. In contrast, pulsing injection in the BIFC regime does not increase cooling film instability; in fact, the amount of instability is less than that seen in FIFC under comparable circumstances.

Moreover, it is shown that the impact of pulsation waveforms on the performance of film cooling is independent of the direction of injection. Compared to their cosine wave equivalents, FIFC and BIFC under square wave BRs both show reduced cooling efficiency and a smaller coverage area. Moreover, square wave pulsations amplify the mainstream flow's transient mixing into the film holes, indicating a crucial element to take into account while designing and analyzing film cooling systems.

6. Discussion

In the realm of gas turbine engineering, the enhancement of cooling efficiency and stability remains a critical concern. A recent study undertakes an in-depth examination of trench film cooling through large eddy simulations, probing the nuanced performance under various blowing ratios (BRs) and the directional influences of forward (FIFC) and backward (BIFC) cooling air injections. This discourse synthesizes the

key findings of the study, underscoring the implications for gas turbine cooling efficiency and stability.

The study's cornerstone is its exploration of the time-averaged effects of steady and pulsating BRs on film cooling. Under static conditions, BIFC emerges as a superior method, enhancing gas film cooling efficiency, extending the film coverage, and achieving a more uniform temperature distribution. These advancements are credited to the formation of asymmetric vortex structures that promote lateral film expansion. This is juxtaposed with the pulsating BR scenario, wherein BIFC's superiority is inverted. Both cosine and square pulsating BRs incur a decline in BIFC's cooling efficiency, with area-averaged reductions of 20.9% and 15.7%, respectively, relative to FIFC. This reversal is attributed to the transient flow dynamics; pulsating BRs foster the development of near-wall counter-rotating vortex pairs in FIFC that hinder coolant ejection toward the wall, enhancing side film coverage. In contrast, pulsating BIFC suffers as a portion of the cooling air is drawn upward by near-wall vortices, compromising its effectiveness.

A novel aspect of this study is the correlation between the instability of trench film cooling and the risk of thermal damage. The study posits that a higher degree of instability translates to greater thermal risk. Temporal fluctuations of CRVPs induce unsteadiness in gas film cooling under steady conditions, creating zones of pronounced instability. Notably, pulsating BRs amplify this instability in FIFC, especially on the sides of the film holes within the trench, whereas the instability downstream remains stable or decreased when compared to steady conditions. In the case of BIFC, pulsating BRs do not exacerbate film cooling instability, which remains lower than FIFC under equivalent circumstances.

The study also elucidates that the impact of pulsation waveforms on gas film cooling performance is independent of the injection direction. A pivotal finding is that square wave BRs are less effective than cosine waves for both FIFC and BIFC, resulting in lower cooling effectiveness and reduced film coverage. The marked characteristic of square wave pulsations is their enhanced transient mainstream intake into the film holes, an effect that must be considered in the optimization of film cooling methodologies.

In summary, this comprehensive analysis of trench film cooling through large eddy simulations reveals that the direction of air injection and the nature of BRs are critical determinants of cooling efficiency and stability. Under steady conditions, BIFC outperforms FIFC, but this advantage is negated under pulsating conditions. The study's insights into the instability of gas film cooling provide a new perspective on evaluating the performance and designing more resilient cooling systems. The findings also highlight the nuanced effects of pulsation waveforms, with square waves presenting specific challenges that must be addressed. As such, this work contributes significantly to the ongoing development of advanced cooling strategies in gas turbine technology, with the potential to influence future design and operational protocols, ultimately improving turbine efficiency and longevity.

The comprehensive study of gas film cooling under large eddy simulation with emphasis on forward and backward injection has paved the way for future research that could significantly advance the domain of gas turbine cooling efficiency and stability. The findings that backward injection film cooling (BIFC) generally surpasses forward injection film cooling (FIFC) under steady blowing ratios (BRs) but

underperforms during pulsating conditions, alongside the observed impact of pulsation waveforms, raise several avenues for further exploration.

Future studies could delve deeper into the material and structural integrity of the turbine components under varying cooling conditions. The interplay between cooling efficiency and component longevity needs to be understood better, particularly under the dynamic conditions of pulsating BRs. Moreover, the thermal fatigue induced by temperature fluctuations warrants a closer look, as it could lead to microstructural degradation over time. Furthermore, the role of advanced materials that can withstand higher temperatures could be evaluated. These materials could potentially reduce the reliance on cooling systems, thus allowing for higher operating temperatures and improved overall efficiency. Investigating alternative pulsation waveforms and their effects on film cooling could reveal new strategies to optimize film cooling effectiveness. The less effective square wave BRs identified in the study suggest that there may be an optimal waveform or a combination of waveforms that could maximize cooling while minimizing the coolant required.

Additionally, the effects of varying geometric configurations of film cooling holes, trenches, and their orientations could be explored. This could include studying the impact of compound angles and shaped holes on the cooling efficiency and film stability. The interaction between film cooling and other cooling techniques, such as internal cooling channels or transpiration cooling, is another potential research area. Combining film cooling with these methods could yield a more robust and efficient overall cooling strategy. Moreover, the study's findings on the instability induced by pulsating BRs call for a detailed investigation into control strategies for mitigating the impact of these instabilities. Active control of the coolant flow could be one such method, dynamically adjusting to the engine's operating conditions to maintain optimal cooling. Real engine conditions, including the effect of engine rotation, pressure gradients, and the realistic thermal boundary conditions, should also be addressed. Simulating these conditions can help bridge the gap between experimental studies and real-world applications. Lastly, computational advancements could enable more detailed and accurate simulations. Future research could leverage even more sophisticated computational methods or hybrid models that combine large eddy simulation with other approaches to improve predictions for film cooling performance.

In conclusion, the study of gas film cooling under large eddy simulation has revealed critical insights into the cooling efficiency and stability of gas turbines, but it has also highlighted the complexity of the problem and the need for ongoing research. By building on the knowledge gained, future research can lead to more effective cooling strategies, enhancing the performance and durability of gas turbines in the field.

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