

Strategic Cooling Enhancement through Trench-Shaped Injection Holes - Two-Phase Numerical Study

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Abstract - A two-phase numerical investigation on trench-shaped injection holes for film cooling is being conducted using Reynolds-averaged Navier-Stokes simulations to assess their practical effectiveness. The study is focusing on the cooling performance of a novel type of film holes, known as straight transverse trenches attached to a flat plate, using a mixture of water mist and air in the coolant fluid. The examination considers film cooling with varying turbulent intensities (3%, 10%, and 20%) and mist concentrations (4%, 7%, and 10%). The analysis involves adiabatic laterally averaged film cooling effectiveness parameters, with two blowing ratios ranging from 0.6 to 1.0. The findings indicate that trenched cooling holes exhibit a more uniform coolant distribution laterally at all positions, resulting in enhanced cooling performance, particularly at high blowing ratios.

Keywords: Film-cooling, effectiveness, mist, trench-hole, turbulent.

1. Introduction

Trenched holes, employed in diverse applications such as industrial furnaces and gas turbine components, epitomize an advanced film cooling technique [1–3]. In this method, the coolant injection hole opening is intricately linked to a longitudinal groove carved into the surface exposed to the hot mainstream gas [4,5]. The primary objective of incorporating the trench is to alleviate vortex formation in close proximity to the injection orifice, effectively diminishing coolant film separation and averting lift-off from the surface [6–8]. In the current scenario, the presence of a trench mitigates severe coolant jet separation, resulting in elevated film cooling efficiency [9,10]. Conversely, without a trench, the coolant's penetration into the mainstream flow renders a significant portion ineffective. The absence of a trench triggers downstream vortex formation, a phenomenon well-documented by previous researchers, which is notably alleviated with the incorporation of a trench, leading to improved effectiveness. Turbulence levels were examined, revealing isotropy for all turbine blades. Film cooling induces higher turbulence compared to conventional cylindrical holes. Trenched holes consistently exhibit superior adiabatic effectiveness, even at low blowing ratios, suggesting enhanced gas turbine performance [11]. Sunderam and Thole [5] conducted a study focusing on the leading-edge region of a stator vane in a gas turbine, where high heat transfer rates occur due to horseshoe vortices. This research delved into the flowfield dynamics in the presence of an interface slot and film-cooling jets. The study considered conditions with no film-cooling holes or upstream slot, film-cooling holes with an interface slot, and film-cooling holes in a trench with an interface slot. Results unveiled the formation of a second vortex in the presence of film-cooling holes and a slot, accompanied by heightened turbulence intensity. In the same study, measurements at a high blowing ratio (BR=2.5) disclosed variations in flowfield characteristics at the leading edge. In a separate investigation, Kalghatgi and Acharya [6,8] introduced an innovative design featuring a round film cooling hole embedded in a contoured crater, aiming to amplify film cooling effectiveness. The contour is strategically designed to weaken the kidney-pair vortex generated by the film cooling jet. He et al. [12] explored film cooling and aerodynamic losses in a turbine guide vane, evaluating three structures: standard cylindrical film holes, traverse trench, and segmented trench. The study revealed that coolant momentum in trenches rose with mainstream momentum, yet misalignment resulted in lateral spread, particularly at high BR and low Mach regions. Trenched holes exhibited higher discharge coefficients at low BR, with comparable total pressure losses to cylindrical holes. Harrison et al. [13] previously investigated net heat flux reduction and found it to be significantly higher for trenches compared to cylindrical holes. The study examined four reference conditions, highlighting distinct distributions of heat transfer coefficients. Additionally, the researchers studied film cooling and heat transfer in a simulated turbine vane with cylindrical holes placed in a transverse trench on the suction side. The results showed increased heat transfer augmentation for untripped approach flow, especially with upstream heating, reaching up to 180% for the trench. In contrast, tripped approach flow exhibited lower levels of augmentation. Zhang et al. [2] investigated the impact of serrate structure on the film cooling performance of trenched holes using a mist/air mixture. Results revealed that the included angle of serrate significantly influenced WFCE, showing an initial increase and subsequent decrease with rising BR's. Mist concentration and droplet diameter played crucial roles in affecting Wet-

bulb-temperature-based film cooling effectiveness (WFCE). The study affirmed that the serrated transverse trench structure, combined with mist injection, significantly improved film cooling performance under practical conditions for a turbine vane. Baheri et al.[14] conducted a comparative-numerical investigation on film cooling from simple and compound-angle holes injected at 35° on a flat plate with four configurations: (1) cylindrical film hole; (2) 15° forward diffused film hole; (3) trenched cylindrical film hole; (4) trenched 15° forward-diffused film hole. The study, performed at a fixed density ratio of 1.6, BR of 1.25, length-to-diameter (L/D) of 4, and pitch-to-diameter ratio of 3.0, also explored the impact of L/D in the range of 1–8. Computational solutions using the finite volume method for the steady, Reynolds-averaged Navier–Stokes equations revealed that the hole shape and trenched holes significantly influenced film cooling over the protected surface. Trenched shaped holes exhibited higher film cooling effectiveness than other configurations, both spanwise and streamwise, particularly downstream of the injection. In a study by Zhang et al.[3], the effectiveness of Serrate-type trenched cooling holes for film cooling in turbine vanes using coolant air was explored. The research also investigated the potential of mist-assisted film cooling for efficient vane cooling. By examining mist/air mixture flow and heat transfer characteristics, the research demonstrated a noteworthy improvement in cooling performance through the introduction of a serrated structure. The serrated trench design generated anti-counter-rotating vortices, preventing coolant film separation from the wall [5,13]. The study suggests potential benefits for mist/air film cooling under engine-representative conditions, offering valuable insights for turbine vane cooling. The span-wise average film cooling effectiveness of the serrated trench with double-curvature lips surpassed traditional transverse trench designs. While the serrated trench performed well with mist/air coolant, its behavior in scenarios with numerous film cooling holes requires further investigation, especially considering mutual jet interactions.

The exploration of lateral film cooling effectiveness under varying turbulence intensity has been neglected in prior studies. Additionally, the synergistic use of water mist and trench injection holes to augment film cooling, surpassing the capabilities of air-only film cooling, has not been previously examined. In this paper, we conduct a three-dimensional numerical investigation, delving into the interplay of Blowing ratio = $[\frac{\rho_c V_c}{\rho_\infty V_\infty}]$, turbulent intensity, and mist concentration for trench holes. This introduces a novel perspective absent in earlier studies.

2. Injection Patterns and Computational Flow Region

This study utilizes a carefully designed computational domain to comprehend essential geometry and flow details for precise numerical calculations. The analysis occurs in a three-dimensional computer model (depicted in Fig. 1), accurately representing the entire area of interest. A trench is affixed to a cylindrical hole configuration, and this trench is linked to the main flow domain for expelling the coolant into the computational domain. The other end of the injection tube is connected to the settling chamber, where air and mist are blended to form a uniform mixture of the coolant fluid. This arrangement divides the overall flow domain into three distinct zones: the main flow duct, injection tubes, and the settling chamber (outlined in Fig. 1). The visualization offers a comprehensive depiction of measurement data, presenting dimensionless values normalized against the injection hole diameter (D), as elucidated further in Table 1. The trench has a height (H_t) of 0.75D from the injection hole opening and a length (L_t) of 3D, as illustrated in Fig. 1(a). A schematic diagram, as depicted in Fig. 1(b), is provided to offer a geometric understanding of this innovative coolant hole design.

In Fig. 1, a single row of three holes, separated by a specified pitch length, maintains a constant inter-hole spacing (pitch = P) of 3.2D. The injection tube's attachment to the main flow domain through an interconnected straight trench introduces an inclination angle (α) of 30°. By strategically defining a region of interest within the computational domain, demarcated by specific coordinates encompassing Z, the study ensures precision in its analysis.

2.2. Numerical Method and Grid-Independent Test

The numerical model adopts a three-dimensional steady-state incompressible Reynolds-Averaged Navier-Stokes (RANS) framework, including the energy equation. Utilizing the realizable k- ϵ turbulence model with advanced wall treatment enhances predictions of coolant diffusion and boundary layer flow under significant temperature gradients. The realizable k- ϵ model is selected for its closest alignment, ensuring credible and precise flow analysis. A coupled pressure-velocity scheme is employed for stability and consistency, maintaining a constant pressure base solver. To enhance convergence and stability, a higher-order term relaxation factor of 0.25 is applied,

ensuring accurate numerical simulations for a comprehensive analysis of fluid dynamics and turbulence. The study confirms its findings by comparing them to the experimental data of Zhao et al. [15], which pertains to cylindrical holes without trenches (although not shown here due to limitations in paper length). Following this, the results obtained with trenches are validated against the research conducted by Lu et al. [16] as shown in Fig. 3(b). This comprehensive approach systematically evaluates and compares the performance of various turbulence models in capturing the intricate dynamics of film cooling processes. A grid sensitivity analysis (shown in Fig. 3(a)) establishes mesh independence, with minimal temperature disparities among different grid sizes. Grid No. 3 is chosen for subsequent analyses. The numerical model is validated for specified conditions, including a mist-free case, following Lu et al. [16]. Inlet conditions include a turbulent velocity profile with a $0.6D$ boundary layer thickness and a Reynolds number of 15,885. The model's predictions for film cooling effectiveness align well with experimental results over a flat plate along the span-averaged effectiveness.

Table 1: Computational Domain Measurements (Normalized by Hole Diameter)

Parameter	Value
Main Flow Domain Height (H_d)	22D
Main Domain Length (L_d)	145D
Main Domain Width (W_d)	14.2D
Settling Chamber Width (W_p)	14.2D
Settling Chamber Height (H_p)	14.2D

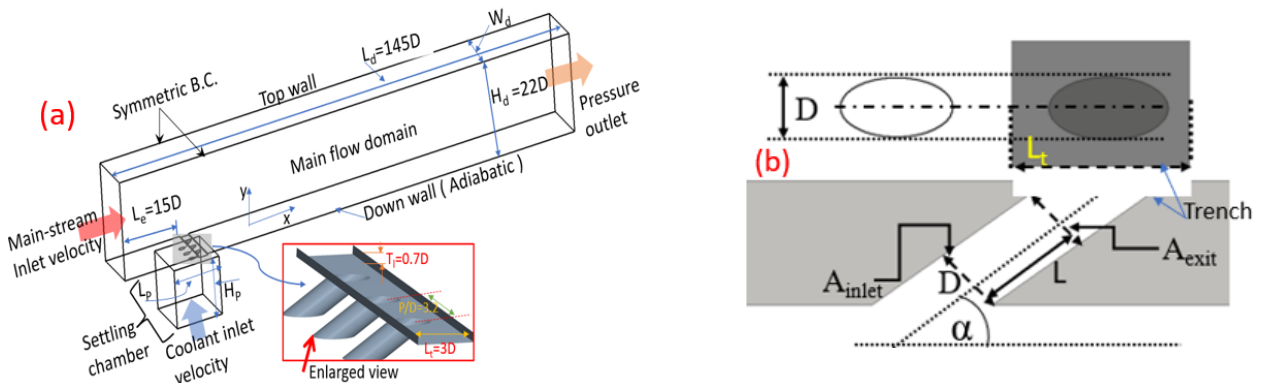


Fig. 1: Comprehensive Overview of the three-Dimensional Computational Domain: (a). Boundary Conditions, and (b). schematic of Trench Injection.

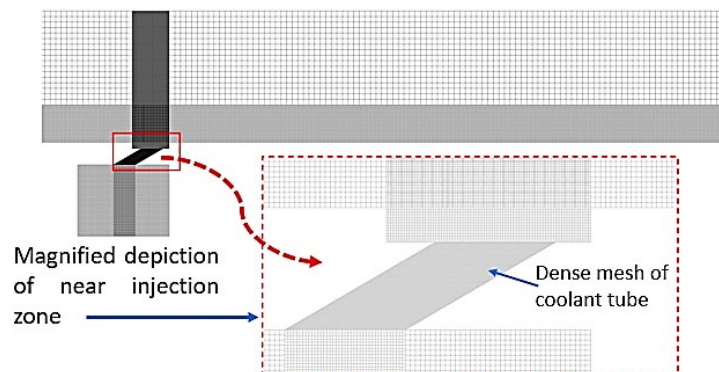


Fig. 2: High-resolution hexahedral meshing for the 3D computational domain with a settling chamber, including a magnified view of the trench-injector tube mesh.

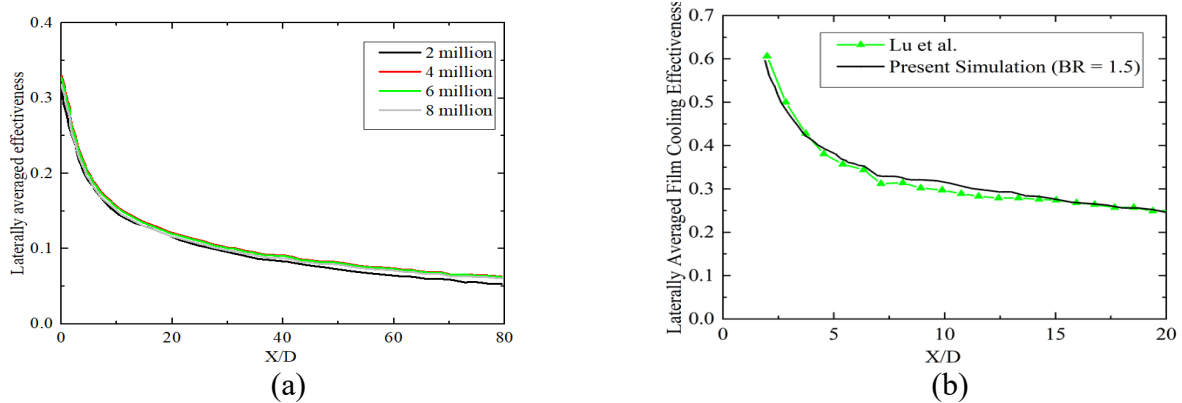


Fig. 3: (a) Grid Independence Study for Film Cooling Effectiveness on a Flat Surface with Varying Grid Sizes and, (b). Validation against Lu et al. [16] Data.

2.1. Computational Domain and Grid Construction

A cooling arrangement incorporates three linearly aligned single-trenched injection orifices to apply a coolant film on a flat surface. These injection ports have an inlet diameter (D) of 6.35mm, and the injection tube extends to a length (L) of $3D$, following the experimental setup of Zhao et al. [15]. Figure 2 provides a close-up view of the meshing around the injection exit, featuring a higher density of mesh nodes near the hole exit for a detailed examination of changes in film performance. In Fig. 2, the mesh structure is intentionally reinforced with inflation layers, increasing density near the flat adiabatic surface and the entry points of the film cooling openings. Additionally, specialized edge meshing is introduced to capture crucial flow phenomena governing turbulence and interactions involving water droplets. The hexahedral computational grid, comprising over 6 million cells, was meticulously generated using ICEM-CFD grid generation software for the cylindrical injection scenario. The geometric configuration of the three distinct computational domains, along with detailed meshing around the injection site, is visually depicted in Fig. 2. The mesh closely interacts with the region around the trench injection outlet, encompassing the internal injection area where significant changes in fluid flow and heat transfer take place. This grid setup aims to facilitate efficient cooling.

2.3. Computational Setup for Turbulence and Governing Equations

The focus of the analysis involves evaluating transport equations for incompressible mass conservation, momentum, and energy. In these assessments, it is reasonable to treat gas properties as constant. Turbulence models are utilized to account for turbulent flows and dissipation. During the interaction between fluid and droplets, significant effects are observed, including the application of drag force on a droplet causing momentum exchange and influencing fluid flow patterns. When employing the Discrete Phase Model (DPM) for particle simulations, the option exists to incorporate these effects. This involves establishing a coupling between Unsteady DPM and steady flow, as discrete particles continuously traverse the system. The analysis, using the Eulerian-Lagrangian (E-L) approach, treats the fluid phase as a continuous medium through the solution of Navier-Stokes equations. Simultaneously, the movement of individual droplets within the calculated flow field is monitored to track the discrete phase. Interactions between dispersed particles, encompassing momentum, mass, and energy exchange with the fluid phase, are considered. A simplification of this approach occurs when particle-particle interactions are disregarded, leading to a more streamlined analysis.

Continuity and Momentum Equations: The numerical resolution of mass and momentum conservation equations is essential for all flows, with an additional focus on energy conservation in cases involving heat transfer. Given the turbulent nature of the flow, additional transport equations are also addressed. The following section outlines the conservation equations specifically tailored for turbulent flow.

$$\frac{\partial(\rho u_i)}{\partial x_i} = S_m \quad (1)$$

$$\frac{\partial(\rho u_i u_j)}{\partial x_i} = \rho g_i - \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\mu_{eff} \left(\frac{\partial(u_i)}{\partial x_j} + \frac{\partial(u_j)}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial(u_i)}{\partial x_i} \right) \right] + F_i \quad (2)$$

Energy Equations: In cases involving heat transfer, an extra energy conservation equation is addressed. Due to the turbulent nature of the flow, additional transport equations are also tackled. This section outlines the conservation equations tailored for turbulent flow in an inertial (non-accelerating) reference frame. Beyond the transport equations governing the continuous phase, a discrete second phase is simulated in a Lagrangian frame. This second phase consists of spherical water droplets dispersed within the continuous phase. The analysis intricately explores the interaction between phases, elucidating its ramifications on both discrete phase trajectories and the dynamics of the continuous phase flow.

$$\frac{\partial}{\partial x_i} (\rho u_i c_p T) = \frac{\partial}{\partial x_j} \left[(k_{eff}) \frac{\partial T}{\partial x_i} \right] + S_h \quad (3)$$

Species Transport Equations:

$$\frac{\partial}{\partial x_i} (\rho u_i C_j) = \frac{\partial}{\partial x_i} \left[(\rho D_{eff}) \frac{\partial C_j}{\partial x_i} \right] + S_j \quad (4)$$

$$\mu_{eff} = \mu_{eff} + \mu_t = \mu + \rho C_\mu \frac{\kappa^2}{\epsilon} \quad (5)$$

$$k_{eff} \frac{\partial T}{\partial x_i} = k \frac{\partial T}{\partial x_i} - \rho C_p (\overline{u_i' T_j'}) \quad (6)$$

$$\rho D_{eff} \frac{\partial C_j}{\partial x_i} = \rho D_i \frac{\partial C_j}{\partial x_i} - (\overline{\rho u_i' C_j'}) \quad (7)$$

The trajectory of a droplet is predicted by integrating a force balance, comparing the droplet's motion strength to the acting forces, as shown in the following equation:

$$\frac{\partial u_p}{\partial t} = F_D (u - u_p) \quad (8)$$

$$F_D = \frac{18 \mu_{eff} C_D Re_D}{24 \rho_p d_p^2}, \quad \text{where } C_D = a_1 + \frac{a_2}{Re_D} + \frac{a_3}{Re_D}$$

$$m_p c_{pp} \left[\frac{\partial T_p}{\partial t} \right] = F_D (T_\infty - T_p) \quad \text{where } (T_{vap} \leq T_p < T_{bp}) \quad (9)$$

$$m_p c_{pp} \left[\frac{\partial T_p}{\partial t} \right] = h A_p (T_\infty - T_p) - \left[\frac{\partial m_p}{\partial t} \right] h_{fg} \quad (10)$$

where F_D is the drag force per unit particle mass. C_D is drag coefficient for smooth particles. Here, constants a_1 , a_2 , and a_3 vary based on different Reynolds number ranges, as specified by Morsi and Alexander [17]. Droplet vaporization is simulated to forecast the evaporation of a discrete-phase particle. The process initiates when the droplet's temperature reaches the vaporization temperature (T_{vap}) and continues until the droplet reaches the boiling point (T_{bp}) or until all easily evaporating components are depleted. Additionally, terms related to transport phenomena like drag coefficient (C_D) and turbulent Prandtl number (P_{rt}) are defined. Other symbols denote variables such as concentration of species (C_j), Reynolds number for droplet motion (Re_D), and drag force per unit particle mass (F_D). Furthermore, important constants and terms related to energy and mass transfer.

2.4. Boundary Conditions

Figure 1(a) presents a comprehensive visualization of the computational domain, highlighting two distinct velocity inlets, a single pressure outlet, symmetric walls, and an adiabatic down wall boundary condition. In the main-stream flow domain's initial segment, a prescribed hot-gas velocity is enforced with an input magnitude of 20.23 m/s, maintaining reference parameter values from prior work [15]. The velocity inlet boundary condition for the coolant (mist-air mixture) is introduced from the bottom of the settling chamber, as illustrated in Fig. 1(a), with a specified blowing ratio (BR) considered as a study variable in this context. A crucial aspect of the study involves maintaining a

specific BR at the lower settling chamber face, enabling the precise determination of the coolant-mist input BR. Thus, the momentum is precisely controlled by manipulating the coolant's mass flow rate at the settling chamber inlet. Throughout the investigation, a consistent density ratio of 1.14 and mist droplet diameter of $5\mu\text{m}$ is maintained across all examined cases, connecting the study with previous research [15] for model reliability. The computational domain is delineated into various zones each with specific boundary conditions. For the Hot Air Inlet zone, the conditions entail a velocity of 20.23 m/s and a reference temperature of $T_\infty = 327.88\text{K}$. Moving to the Coolant Inlet zone, the BR varies between 0.6 to 1.0 while maintaining a temperature of 288.35K. Symmetry conditions are applied to the Side Walls zone (as shown in Fig. 2), ensuring no alteration in flow across these boundaries. The Other Walls zone is governed by a no-slip condition, indicating that fluid velocity at these walls is nil. Finally, the Outlet zone is defined with a pressure condition set to 1 atm.

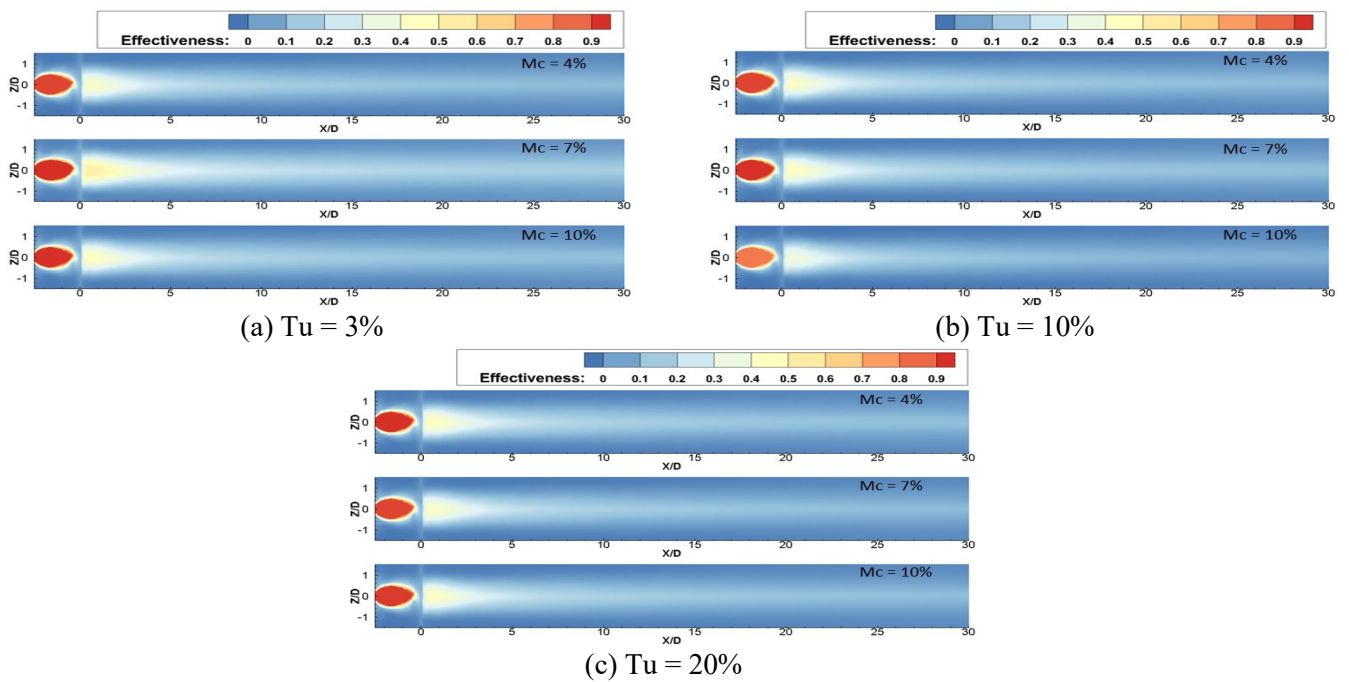


Fig. 4: Contours depicting Film Cooling Effectiveness at a Blowing Ratio (BR) of 0.6 with varying turbulent intensities and mist concentrations (4%, 7%, and 10%).

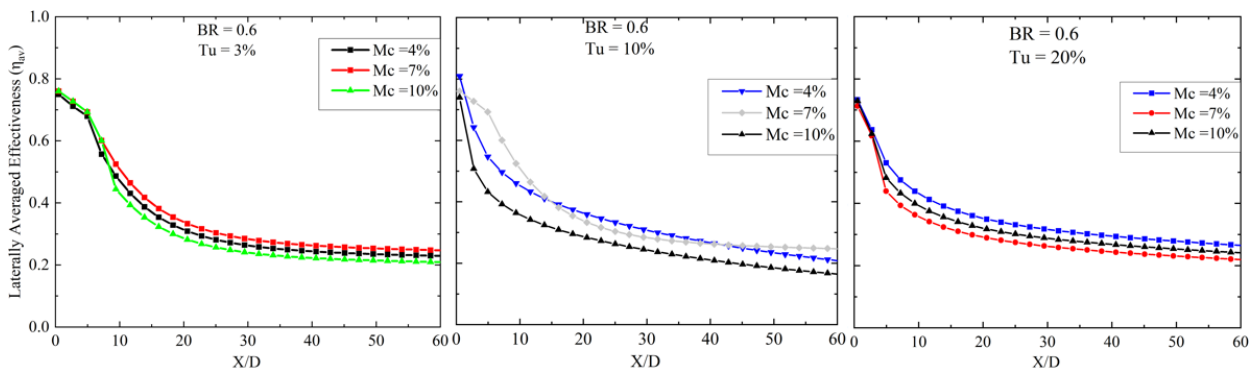


Fig. 5: Variation of Laterally Averaged Film Cooling Effectiveness (η_{av}) with Mist Concentrations (4%, 7%, and 10%) at a Blowing Ratio (BR) of 0.6 under Changing Turbulent Intensities.

3. Results and Discussion

3.1 Effect of Mist Concentration

Figures 4 and 5 illustrate the impact of Mist Concentration (Mc) on spanwise-averaged film effectiveness, correlating with two Blowing Ratios (BR) of 0.6 and 1.0. The results are presented against the normalized streamwise distance (X/D) downstream of the film hole's edge at constant turbulent intensity (Tu). For a lower BR value of 0.6 and a Turbulent Intensity (Tu) of 3%, the peak of Laterally Averaged Effectiveness of mist film (η_{av}) occurs at a mist concentration (Mc) of 7%. As Tu increases to 10%, a subsequent reduction in η_{av} becomes evident for the 7% mist concentration case at a streamwise location between $15 < X/D < 40$. From Fig. 5, it is apparent that with a low Tu of 3% and $BR = 0.6$, the η_{av} initially rises as the Mc increases from 4% to 7%. However, beyond a 7% mist concentration, it declines. Additionally, it is noted that at locations near the injection hole, under the same flow conditions ($Tu = 3\%$, $BR = 0.6$), the laterally averaged effectiveness exhibits marginal disparity, which is less than 5% for each concentration case. This phenomenon can be attributed to the lower mass flow rate of the mist coolant mixture at $BR = 0.6$. Additionally, the minimal turbulence present prevents the film layer from diffusing into the mainstream hot gas. Consequently, the film remains attached to the surface for an extended X/D length without any disturbance. However, the 10% mist concentration exhibits minimal effectiveness for the remaining portion of the surface away from the injection hole. This could be attributed to the increase in relative humidity with higher Mc 's, which impedes evaporation. Additionally, droplets ejected from the circular section without evaporation possess higher momentum compared to the remaining coolant mixture. As a result, these droplets penetrate the mainstream flow and disperse from the surface without providing substantial cooling effects. The contours of film cooling effectiveness for mass flow ratios $BR = 0.6$ and $Tu = 10\%$ are depicted in Fig. 4(b) for various Mc , illustrating how the film covers the vane surface with increased turbulence. Graphical representation is also provided in Fig. 5 for the same Tu and BR . It has been observed that from $X/D = 3$ to $X/D = 15$, $Mc = 7\%$ yields the highest laterally averaged cooling effectiveness, representing an optimal parametric choice for near injection hole cooling, as highlighted in several previous studies [18,19]. However, within the region between $X/D = 15$ to $X/D = 40$, the laterally averaged effectiveness of $Mc = 7\%$ is surpassed by $Mc = 4\%$. The maximum disparity, observed at $X/D = 25$, is 9% when comparing $Mc = 4\%$ and 7% , which may be attributed to the sharp nature of the trench edge. Similar to $Tu = 3\%$, $Tu = 20\%$ exhibits a similar degradation trend of laterally averaged effectiveness with X/D distance, as depicted in Fig. 5. However, $Mc = 7\%$ yields the lowest laterally averaged effectiveness compared to other Mc cases. Additionally, contours of film cooling effectiveness at mass flow ratios $BR = 0.6$ and $Tu = 20\%$ are shown in Fig. 4(c), where almost no difference is identified between $Mc = 4\%$ and 10% . It means that the variation in Mc does not affect the coolant film structure over the surface. Now, at higher Tu of 20% and at $BR = 0.6$, it has been found that with the increase in Mc from 4% to 7%, the laterally averaged effectiveness decreases. If the Mc is increased further, the η_{av} gain enhances. The reason behind the enhancement in the case of high Mc is that the availability of water content in the coolant air is comparatively higher, and due to high turbulence intensity, the mist is mixed with hot gas more effectively, causing enhanced evaporation of the mist. Thus, the surrounding temperature is also dropped.

On the other hand, a high Blowing Ratio (BR) of 1.0, as depicted in Figs. 6 and 7, highlights the influence of Mc on spanwise-averaged film effectiveness analyzed across several Tu . In Fig. 6, the effectiveness distribution illustrates an expanded longitudinal spread at $BR = 1.0$ for each Tu and Mc compared to $BR = 0.6$. Notably, a long triangular region of higher film cooling effectiveness is easily identifiable in Fig. 6 and can be compared to Fig. 4 for the $BR = 0.6$ case. For $BR = 1.0$ and $Tu = 3\%$, the peak of Laterally Averaged Effectiveness of mist film (η_{av}) is observed at a Mc of 7%, near the trench edge. However, there is less than a 5% disparity in the 7% mist content case when compared with an Mc of 4%, and the maximum disparity in magnitude is observed between $X/D = 7$ to $X/d = 25$. Further increasing Mc leads to a reduction in η_{av} , as seen in Fig. 7 for $Mc = 10\%$ under the same $Br = 1.0$ and $Tu = 3\%$. As Tu increases to 10% at the same BR of 1.0, a similar trend of higher laterally averaged film cooling effectiveness is observed for the 7% mist concentration case, as previously seen for $BR = 0.6$ and $Tu = 10\%$ for downstream locations between $X/D = 0.5$ to $X/D = 13$. However, for the rest of the X/D locations, the effectiveness for $Mc = 7\%$ is lower than the 4% case, and in some places, it is even lower than $Mc = 10\%$, as depicted in Fig. 5. This indicates that at low Tu of 3% and BR of 1.0, an increase in Mc leads to a reduction in effectiveness at higher BR, similar to $BR = 0.6$. This observation is also consistent with the $Tu = 10\%$ case. However, the lateral spread of the film over the surface at $BR = 1.0$ remains consistent across all Mc cases, as shown in contour Fig. 6; the spreading of the film is wider than the $BR = 0.6$ case. With the increase of BR, the spreading of the film improves, thus enhancing effectiveness. Moreover, at a higher Tu of 20% and

BR = 1.0, it is observed that a higher Mc of 10% provides the highest η_{av} compared to Mc cases of 4% to 7%. This enhancement with high Mc is attributed to the higher water content in the coolant air, allowing it to travel a greater distance without immediate evaporation at near injection locations. Additionally, due to the high turbulence intensity, the mist is mixed with hot gas more effectively at far downstream locations, leading to enhanced evaporation of the mist.

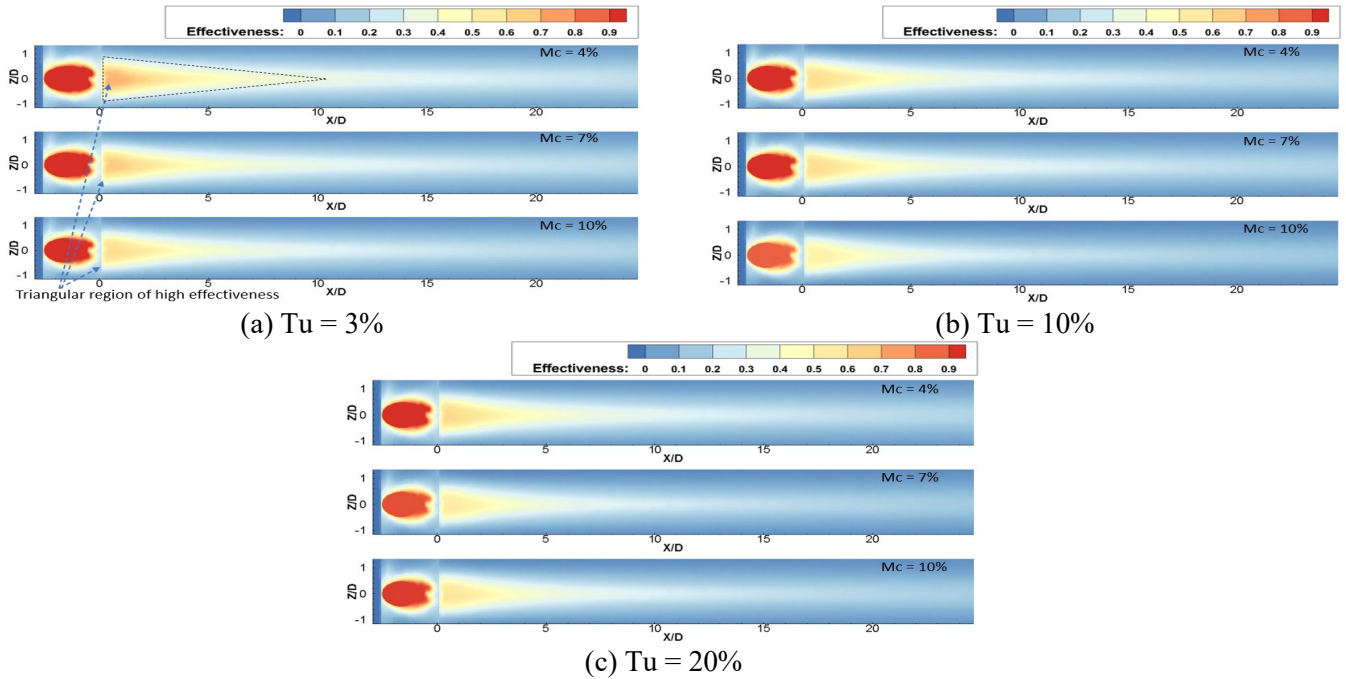


Fig. 6: Contours depicting Film Cooling Effectiveness at a Blowing Ratio (BR) of 1.0 with varying turbulent intensities and mist concentrations (4%, 7%, and 10%).

3.2. Effect of Turbulent Intensity

In this section, the impact of turbulent intensity on the film cooling performance of the coolant ejection trench on the flat plate is explored while keeping the mist concentration (Mc) constant for two Blowing Ratios (BRs) of 0.6 and 1.0. The analysis focuses on the variation in laterally averaged effectiveness, as depicted in Fig. 8. Results reveal that at BR = 1.0 and Mc = 4%, the η_{av} is highest across all X/D locations for Tu = 3%. However, at the same Mc and Tu, lower BR = 0.6 exhibits enhanced lateral film cooling effectiveness within the region spanning from X/D = 4 to X/D = 8, as illustrated in Fig.8(a). The enhanced effectiveness observed at lower BR = 0.6 can be attributed to the formation of cold spots created by non-evaporated mist droplets. These cold spots result in localized areas of high effectiveness. At a low Mc of 4%, the laterally averaged film cooling diminishes up to X/D = 15 with a low Blowing Ratio (BR) of 0.6. However, beyond this point (X/D < 15), the η_{av} exhibits a different trend, being higher for higher Turbulent Intensity (Tu) values of 10% and 20%, and lower for a Tu of 3%. Conversely, at BR = 1.0, the η_{av} decreases with increasing Tu at Mc = 4%. With a slight increase in Mc, the impact of changing turbulent intensity closely resembles that observed with a 4% mist concentration. Like the 4% mist concentration scenario, there exists a downstream region between X/D = 3 to X/D = 10 where Tu = 3% and 10% result in the highest η_{av} . Conversely, for the remaining surface area, Tu = 3% at BR = 1.0 demonstrates enhanced η_{av} compared to all other cases, as shown in Fig. 8(b). The only distinction is observed at Tu = 20% and BR = 0.6, where the η_{av} steeply decreases near the downstream location of X/D < 10. At the highest mist concentration, the variation of Turbulent Intensity (Tu) affects the cooling effectiveness in a slightly different manner. Here, the η_{av} is highest for the surface region from x/D = 10 to the end of the plate when BR = 1.0 and Tu = 20%. Additionally, for the region near the trench, a low BR of 0.6 and low Tu of 3% are beneficial, as observed in slightly lower mist concentration cases. However, at Tu = 10% and BR = 0.6, the laterally averaged film cooling effectiveness is lower compared to all other cases. The above discussion leads to the conclusion that for low mist concentration, high BR, and low Tu are optimal for achieving greater η_{av} . Conversely, for high mist concentration, attaining higher effectiveness requires both high Tu and BR.

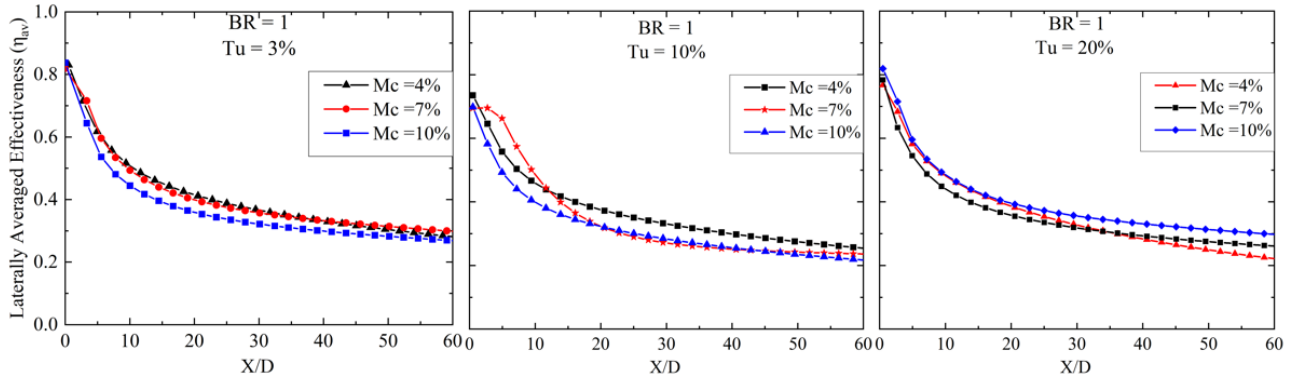


Fig. 7: Variation of Laterally Averaged Film Cooling Effectiveness with Mist Concentrations (4%, 7%, and 10%) at a Blowing Ratio (BR) of 1.0 under Changing Turbulent Intensities.

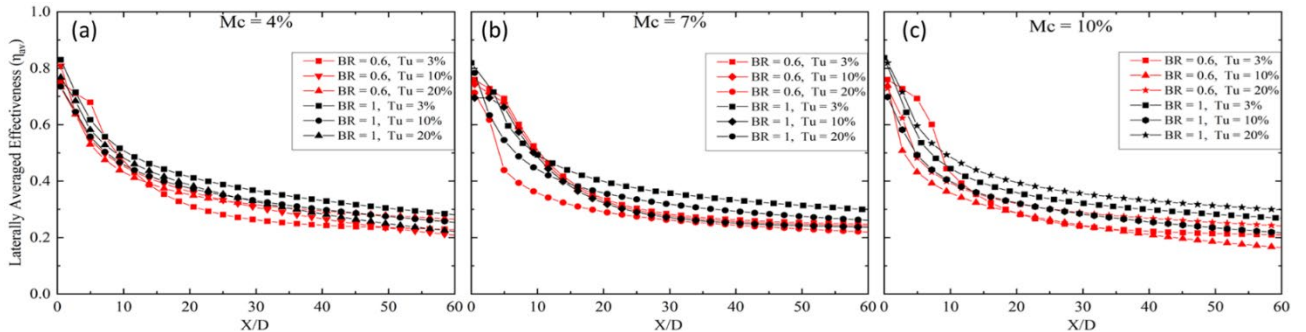


Fig. 8: Variation of Laterally Averaged Film Cooling Effectiveness with Turbulence intensity (3%, 10%, and 200%) under Changing Mist Concentrations.

3.3. Effect of Blowing Ratio

In this section, an examination of the laterally averaged effectiveness (η_{av}) and its connection to variations in the Blowing Ratio (BR) is presented, specifically focusing on two scenarios: BR = 0.6 and 1.0. Analysis of Fig. 8 distinctly reveals that, at low mist concentrations, an increase in BR heightens η_{av} . This phenomenon occurs because the coolant-air mixture gains increased momentum, thereby enhancing film cooling efficiency. However, noteworthy regions near the trench coolant ejection exhibit a different trend. Here, lower blowing ratios result in higher effectiveness, particularly dependent on Turbulent Intensity (Tu). These regions of elevated effectiveness for low blowing ratios are concentrated near the injection hole location, owing to the establishment of a cold spot due to mist accumulation. Additionally, the low mass flow ratio causes mist droplets to be trapped near the trench downstream surface, further enhancing localized cooling. The presence of turbulence intensifies mixing and improves cooling efficiency near the injection hole. Interestingly, despite increases in mist concentration, BR = 1.0 consistently demonstrates superior cooling capacity, attributed to its higher mass flow rate, enabling more effective heat transfer, especially away from the trench. Furthermore, higher mist concentrations amplify effectiveness, particularly at BR = 1.0, due to turbulence promoting better coolant mixing with the mainstream flow, thereby enhancing convective heat transfer and film cooling effectiveness.

5. Conclusions

This study is distinctive in exhibiting the stagnation plane flow field with mist-film cooling injection, considering the influence of coolant flow from an upstream slot. The results highlight a significant enhancement in adiabatic effectiveness when film-cooling holes in a trench are coupled with added mist.

- Higher mist concentrations enhance effectiveness due to improved mist evaporation, particularly under elevated turbulence.
- Maintaining constant parameters, the study observes that higher turbulent intensity boosts film cooling effectiveness. Optimal conditions include low turbulence, high BR, and low Mc, emphasizing the delicate balance.
- Higher blowing ratios generally enhance effectiveness, but specific regions near the injection hole favor lower ratios. A consistent superiority of BR = 1.0 is observed, particularly evident at higher mist concentrations.

- Spatially, the examination reveals distinct regions of effectiveness concentration near the injection hole, influenced by mist concentration, Tu, and BR. Optimal conditions for effectiveness vary across mist concentrations, Tu levels, and BR values, providing valuable insights into achieving superior film cooling performance in specific operational scenarios.

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