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Influence of the internal structures on the performance of sweeping jet film cooling

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ABSTRACT

As sweeping jet film cooling exhibits a remarkable improvement in lateral film effectiveness, research on fluidic oscillators to provide better film coverage is gaining interest. However, compared with the other type holes, the traditional fluidic oscillator is too large for applying well on film cooling. To modify the size of the oscillator, three geometrical parameters are chosen to be analyzed (e.g., the width of the inlet wedge $W_i$, the length of the mixing chamber $W_m$, and the width of the feedback channel $W_b$, normalized by the width of the exit throat $D$). Three-dimensional unsteady RANS simulations are conducted to evaluate the influence of each parameter on the adiabatic film cooling effectiveness, frequency, and flow structures. The simulated results show that the width of the feedback channel has little influence on frequency, whereas the length of the mixing chamber has a significant impact on film cooling effectiveness and frequency. Based on the above results, a modified fluidic oscillator has been proposed, which has a more compact structure and shows higher film cooling effectiveness at the same blowing ratio.

INTRODUCTION

Over the past few decades, the turbine inlet temperature has gradually increased to well above the melting point of the manufacturing materials at the high-temperature parts of gas turbines. It is particularly important to develop efficient cooling technology to ensure the stable work of gas turbines. One of the most advanced methods of engine cooling is film cooling. It means that the coolant is injected from the surface of the high-temperature part at a certain angle through the seam or discrete hole. Then, the coolant can form a protective film between the high-temperature flow and the wall surface, to separate the high-temperature mainstream from the surface.

In the early stage, cylindrical holes were widely used in film cooling. However, it was found that the cylindrical film holes had the defects of small spread coverage area, concentrated jet momentum, strong penetration to the mainstream, and leading to very low cooling efficiency between the holes. Goldstein (Goldstein, Eckert and Burggraf, 1974) first proposed the fan-shaped hole to replace the cylindrical hole to achieve better cooling performance. Therefore, different shapes can impact film cooling effectiveness significantly. A series of studies were carried out on the influence of different shaped holes on film cooling, such as arrowhead-shaped holes (Okita and Nishiura, 2007), console holes (Sargison et al., 2001), waist-shaped slot holes (Liu et al., 2011), Round-to-slot-shaped holes (Huang, Zhang and Wang, 2018)(Huang, Zhang and Wang, 2020), horn-shaped holes (Zhu, Simon and Xie, 2018), tripod holes (Ramesh et al., 2016), Sister Holes (Ely and Jubran, 2008)(Ely and Jubran, 2009)(Khajehhasani and Jubran, 2015), crescent holes (Dai and Lin, 2011)(Zhou et al., 2020). These holes provide a better cooling effect than cylindrical holes.

The above cooling holes had a major problem in that the footprint of the film was too narrow to protect the wall. Since fluidic oscillators can generate a sweeping motion, the sweeping jet film cooling has a notable improvement in lateral film effectiveness. Research on fluidic oscillators to provide better film coverage is gaining interest. As shown in Figure 1, the traditional fluidic oscillator consists of an inlet nozzle, a mixing chamber, two feedback channels, and an outlet nozzle. This device has no moving parts, but it can produce oscillating flow. Fluid oscillators have been used in many ways, i.e., flow separation control (Jentzsch, Taubert and Wygnanski, 2019)(Kara, 2017)(Koklu and Owens, 2017), combustion control.
Thurman (Thurman et al., 2016) used infrared thermography, thermocouple measurement, hot wire measurement, and PIV data to measure the jet oscillator, and compared the jet hole with the diffusion hole (777-shaped hole and square hole). They found that the fluidic oscillator had lower centerline effectiveness but better uniform flow distribution compared to the other holes. Hossain (Hossain et al., 2017) studied the performance of sweeping jet holes, and the results showed that the film cooling effectiveness of the sweeping jet is better than the shaped hole.

Because the geometry of the fluidic oscillator can impact the flow pattern sensitively, it is necessary to investigate the influence of the internal structure of a fluidic oscillator on film cooling. Bobusch (Bobusch et al., 2013) used a validated numerical model to parameterize the internal geometry of the oscillator, including the length of the feedback channel, the width of the inlet wedge, and the inlet of the feedback channel. The results revealed the influence and sensitivity of these geometric features and provided ideas on how to influence the frequency and jet deflection angle. Tomac (TOMAC, 2020) studied fluid oscillators of various lengths with the same main geometric dimensions. The results showed that oscillation stops when the length of the fluidic oscillator is shorter than the threshold value. Wen (Wen et al., 2020) conducted an experimental study by using TR-PIV to obtain the time-resolved flow fields of the fluidic oscillator. The inlet wedge width, mixing chamber width, and exit throat width were considered to examine the jet spreading angle, frequency of the external jet, the feedback flow, and momentum distribution.

Though previous work had studied the impact of the geometry on flow fields of the fluidic oscillator, there is still a lack of the relation between the geometry and the film cooling effectiveness. Besides, compared with the other type holes, the size of the traditional fluidic oscillator is too large for applying well on film cooling. Therefore, in this study, a validated numerical model was employed to investigate the impact of the internal structure of a fluidic oscillator on film cooling. Three geometrical parameters are chosen to be analyzed (e.g., the width of the inlet wedge $W_1$, the length of the mixing chamber $W_2$, and the width of the feedback channel $W_3$, normalized by the width of the exit throat $D$), and Table 1 shows the numerical simulation of operating conditions. At different values of these parameters, film cooling effectiveness, frequency of the external jet, and flow structures are examined. Moreover, a more compact design based on the above results has been proposed and the film cooling effectiveness is better than the traditional hole at $M = 1.5, 2.0, \text{ and } 3.0$.

### Table 1: Operating conditions

<table>
<thead>
<tr>
<th>Geometry part</th>
<th>Baseline</th>
<th>Case1</th>
<th>Case2</th>
<th>Case3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet wedge</td>
<td>$W_1/D = 2.1$</td>
<td>$W_1/D = 1.8$</td>
<td>$W_1/D = 1.5$</td>
<td>$W_1/D = 1$</td>
</tr>
<tr>
<td>Mixing chamber</td>
<td>$W_2/D = 6.1$</td>
<td>$W_2/D = 5.5$</td>
<td>$W_2/D = 5$</td>
<td>$W_2/D = 4.5$</td>
</tr>
<tr>
<td>Feedback channel</td>
<td>$W_3/D = 1.6$</td>
<td>$W_3/D = 1.4$</td>
<td>$W_3/D = 1.2$</td>
<td>$W_3/D = 0.9$</td>
</tr>
</tbody>
</table>

**NUMERICAL SETUP**

To uncover the flow physics of the sweeping jet, a numerical study was conducted for all cases. The following figures show the numerical setup and validation at $M = 2.0$ and 3.0. The domain was extended $40D$ in the streamwise direction,
12D in the spanwise direction, and 9D along the wall-normal direction. The mainstream inlet is located at 10D upstream of the hole. And the angle between the mainstream and the coolant inlet is 35 degree. Velocity inlet was set for mainstream inlet, while the mainstream outlet was set as a pressure outlet with zero-gage pressure boundary conditions. The coolant inlet was set as a velocity inlet condition. The turbulence level was set as 1.5% for both inlets. Two sides of the domain were set as periodic boundary conditions. Both inlets were set to be ideal gas with a total temperature ratio of 0.97 to simulate the density ratio of nitrogen.

A view of the grid for the traditional fluidic oscillator is shown in Figure 2. A structured grid was used in this study which consists of boundary layer meshes with a non-dimensional wall distance $y^+$ close to unity. Due to the flow field of a fluidic oscillator being unstable, a three-dimensional unsteady Reynolds-averaged Navier-Stokes simulation (URANS) was employed in this study using commercial code FLUENT to gain the time-averaged and unsteady film cooling effectiveness and flow fields. The time steps were ensured by the CFL number close to unity. A second-order upwind scheme was used on the flow equations. The unsteady calculation was discretized by a second-order implicit discretization.

![Figure 2. Computational setup.](image)

Here, the film cooling effectiveness ($\eta$) was applied to evaluate the coolant coverage, region with $\eta = 1.0$ indicating a 100% coverage while with $\eta = 0$ meaning no film. It was determined by using the following non-dimensional definition to quantify the performance:

$$\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_c}$$

where $T_\infty$ is the mainstream temperature, $T_c$ is the coolant temperature, and $T_{aw}$ is the adiabatic surface temperature.

![Figure 3. Grid independent test.](image)

Figure 3 shows the mesh independent comparisons of the traditional fluidic oscillator at $M = 3.0$, and finally with a fine mesh of 5 million cells.

Figure 4 shows the time-averaged film cooling effectiveness and the laterally averaged effectiveness of the numerical results and the experimental one (Zhou et al., 2022) at $M = 2.0$ and 3.0. The contour of the film cooling effectiveness shows a similar film footprint and trend. Generally, the numerical effectiveness is consistent with the experimental results. However, compared with the experimental effectiveness, the predicted one is slightly higher at $M = 2.0$ and 3.0. This can be explained by the k-\omega SST model, which predicts a weaker penetration of coolant into the mainstream, resulting in a better cooling effect compared to the experimental study. Besides, the error of the predicted frequency of the jet and the experiments is within 2% for these two blowing ratios. Since there are only a few geometric variations were conducted, this numerical study can provide reliable results.
RESULTS AND DISCUSSION

**Inlet wedge width.** Bobusch (Bobusch et al., 2013) found that with a fixed internal geometry, the inlet wedge has a distinct impact on the feedback flow, which then influences the external flow fields. Since the external flow fields are directly related to the application of film cooling, there is necessary to examine the film cooling effectiveness of different inlet wedge widths. Here, $W_1/D = 1.8$, 1.5, and 1 were proposed, as shown in Figure 1, $W_2/D$ and $W_3/D$ are fixed at 6.1 and 1.6, respectively.

![Figure 4](image1.png)  
*Figure 4. The time-averaged film cooling effectiveness and the laterally averaged effectiveness, (a) $M = 2.0$, (b) $M = 3.0$.***

![Figure 5](image2.png)  
*Figure 5. The film cooling effectiveness at $M = 3.0$, (a) Baseline, $W_1/D = 1.8$, 1.5, 1 (from top to bottom), and (b) laterally averaged effectiveness.*

Figure 5 (a) shows the contour of the time-averaged film cooling effectiveness at $M = 3.0$ with different inlet wedge widths and the Baseline. With the decrease of $W_1/D$, the coverage of the film in both streamwise and spanwise directions...
gradually shrinks except the case of \( W_1/D = 1.8 \), which also means that the film cooling effectiveness decreases significantly. When \( W_1/D = 1.8 \), the film covers more uniformly in the spanwise direction and extends farther along the streamwise than the Baseline. To quantitatively compare the cooling performance, the laterally averaged effectiveness of the above four structures was determined by averaging the effectiveness in the spanwise direction within the \(-3 \leq Y/D \leq 3\) regions, which is shown in Figure 5 (b). It can be seen that the value of the cooling effectiveness of the \( W_1/D = 1 \) shows \(-45\%\) and \(-82\%\) lower than the Baseline.

\[
\sqrt{\frac{\nu^2 + v^2 + \frac{w^2}{U_c}}{U_c^2}}
\]

![Flow fields comparison](image)

**Figure 6.** The time-averaged flow fields at \( M = 3.0 \), (a) Baseline, \( W_1/D = 1.8 \), 1.5, 1 (from left to right), and (b) oscillation frequency.

Figure 6 shows the time-averaged flow fields in fluid oscillators with different inlet wedge widths and the oscillation frequency of each size at \( M = 3.0 \). It can be seen from the velocity contour and streamline that the inlet wedge width not only affects the flow rate but also affects the flow direction in the feedback channel. When the width of the inlet wedge is \( W_1/D = 1.8 (W_1 > W_2) \), most of the main jet flow at the throat of the inlet enters the control chamber, and a small portion enters the outlet of the feedback channel. Therefore, the feedback flow in the feedback channel is dominant, and the jet at the outlet produces obvious oscillation, but it is still less than the oscillation frequency of the Baseline. When the inlet wedge width is reduced to \( W_1/D = 1.5 (W_1 \approx W_2) \), it is obvious that a large number of vortices appear in the feedback channel, and continuous feedback flow cannot be seen. Hence, the frequency of the exit flow gets lower. Finally, when the size of the inlet wedge is reduced to \( W_1/D = 1 (W_1 < W_2) \), a large number of main jets are separated and flow into the feedback channel before entering the control chamber, and the remaining fluid directly exits from the inlet of the control chamber to the outlet throat. Since there is no flow in the direction of backflow in the feedback channel, the frequency of the main jet is greatly reduced. And this phenomenon was found in Wen (Wen et al., 2020). For the Baseline \( (W_1 > W_2) \), the feedback flow is the strongest of all, and the oscillation frequency is naturally the highest.
Figure 7. The velocity distribution inside the oscillators at $M = 3.0$, (a) Baseline, (b) $W_1/D = 1$.

Figure 7 shows the instantaneous velocity of Baseline and $W_1/D = 1$, to uncover the flow pattern inside the oscillators with two different inlet wedge widths. In Figure 7 (a), when the fluid enters the mixing chamber and is attached to one wall because of the Coanda effect (which can make the fluid change its original direction and flow along the surface of the object in contact, here is the left wall at $t_0$). Then, at $t_0 + 0.25T_0$, the fluid passes through the feedback channel, and enters the mixing chamber again, starting to generate a recirculation bubble. At $t_0 + 0.5T_0$, the recirculation bubble pushes the main jet to another wall. With a portion of the jet returning to the mixing chamber, another bubble is generating at $t_0 + 0.75T_0$. However, with the decreasing of the inlet wedge width, the flow pattern inside the oscillator is changing. At $W_1/D = 1$, we can find that there is no obvious feedback flow in Figure 6 (a), which still shows the sweeping motion, and the instantaneous velocity in Figure 7 (b) can explain it. At $t_1$, a portion of the main jet is introduced to the mixing chamber and feedback channel, and the fluid in the right channel is dominated, which generates a recirculation bubble in the mixing chamber, while the flow in the left channel is directed out the hole. Later, the flow on the left side of the channel is becoming stronger, pushes the main jet to another side wall, and generates a recirculation bubble in the mixing chamber (at $t_1 +$...
At $t_1 + 0.5T_1$, the generated bubble makes the fluid in the right channel sweep out with the main jet. Finally, the fluid inside the oscillator repeats the process and turns to the left (at $t_1 + 0.75T_1$).

Figure 8. (a) Non-dimensional temperature and (b) vorticity distribution at $X/D = 5$ with $M = 3.0$. Baseline, $W_1/D = 1.8, 1.5, 1$ (from left to right).

Figure 8 shows the dimensionless temperature and vortex distribution at $X/D=5$. From the temperature distribution, with the decrease of $W_1/D$, the penetrating effect of coolant on the mainstream is gradually obvious, which is embodied in that the coolant is farther from the wall, and finally exceeds the position of $Z/D = 3$. At $W_1/D = 1.8$, the coolant penetration to the mainstream is close to that of the Baseline, but the coolant contact at the wall position ($Z/D = 0$) is better than that of the Baseline. This also explains that in Figure 5 (a) above, the Baseline presents two narrow coolant footprints, and there is no coolant covering in the middle area, while film coverage at $W_1/D = 1.8$ is more uniform. By decreasing the inlet wedge width to $W_1/D = 1.5$, the coolant is in contact with the wall only in the range of $Y/D = \pm 1$. Therefore, the coverage of the film shown in Figure 5 (a) becomes narrower in the spanwise direction. By decreasing $W_1/D$ to 1, it’s obvious that there is no coolant attached to the wall, which has been completely separated, finally leading to almost no film covering on the wall, and the cooling effect is significantly reduced. Figure 8 (b) depicts the flow structures of all configurations. The existence of anti-CRVP can be observed at the case of $W_1/D = 1.8, 1.5$ and Baseline. However, the size of the vortex keeps shrinking until it completely disappears in the case of $W_1/D = 1$, and only CRVP plays a role. CRVP can take coolant away from the wall, making it less protected by coolant, which is shown in inclined jet-in-crossflow (Fric and Roshko, 1994), anti-CRVP can drive the coolant to the wall, resulting in better coverage.

Mixing chamber length. The length of the mixing chamber can impact the size of the recirculation bubble, which can affect jet spreading angle and frequencies, resulting in different external flow fields and film cooling effectiveness. Therefore, different lengths of the mixing chamber are considered here to study how this parameter influences the flow structures and film cooling effectiveness. In this section, $W_2/D = 5.5, 5$, and 4.5 were selected as shown in Figure 1, while $W_1/D$ and $W_3/D$ are fixed at 2.1 and 1.6, respectively.

Figure 9. The film cooling effectiveness at $M = 3.0$, (a) Baseline, $W_2/D = 5.5, 5, 4.5$ (from top to bottom), and (b) laterally averaged effectiveness.
Figure 9 (a) depicts the contour of the time-averaged film cooling effectiveness at $M = 3.0$ with different lengths of the mixing chamber and the Baseline. It is obvious that compared with the Baseline when the $W_2/D$ decreases, the coverage of the film becomes wider and the extension in the streamwise direction becomes longer, which also means that the protection on the wall becomes better. Figure 9 (b) shows the laterally averaged film cooling effectiveness by averaging the effectiveness in spanwise direction within $-3 \leq Y/D \leq 3$ range. By decreasing $W_2/D$, the film cooling effectiveness is getting higher. Especially, the value of the cooling effectiveness of the $W_2/D = 4.5$ shows ~8% and ~100% higher than the Baseline.

![Figure 9](image)

Figure 9. (a) Non-dimensional temperature and (b) vorticity distribution at $X/D = 5$ with $M = 3.0$. Baseline, $W_2/D = 5.5, 5, 4.5$ (from left to right).

Figure 10 shows the dimensionless temperature and vortex distribution at $X/D = 5$. As can be seen from the temperature distribution, with the decrease of $W_2/D$, the contact width between the coolant and the wall gradually increases until it completely covers the range of $-3 \leq Y/D \leq 3$. The vortex distribution in Figure 10 (b) describes the flow field structure of the fluidic oscillator at $X/D = 5$ downstream of the holes. All structures generate an anti-CRVP structure, which effectively inhibits CRVP. With the decrease of $W_2/D$, the anti-CRVP structure becomes larger and larger, resulting in better film coverage, which is shown in Figure 9 (a).

![Figure 10](image)

Figure 10. (a) Non-dimensional temperature and (b) vorticity distribution at $X/D = 5$ with $M = 3.0$. Baseline, $W_2/D = 5.5, 5, 4.5$ (from left to right).

Feedback channel width. The feedback channel is generally expected to be the most important parameter that impacts the frequency of the external jet and may have a significant influence on the coverage of the film. Here, $W_3/D = 1.4, 1.2, \text{ and } 0.9$ were selected, as shown in Figure 1, while $W_1/D$ and $W_2/D$ are kept constant as 2.1 and 6.1, respectively.

![Figure 11](image)

Figure 11. The film cooling effectiveness at $M = 3.0$, (a) Baseline, $W_3/D = 1.4, 1.2, 0.9$(from top to bottom), and (b) laterally averaged effectiveness.

Figure 11 (a) shows the contour of the time-averaged film cooling effectiveness at $M = 3.0$ with different lengths of feedback channel and the Baseline. With the gradual decrease of $W_3/D$, the overall trend of film coverage is close, while the range of film diffusion is getting narrow. From Figure 11 (b), the film cooling effectiveness of all structures decreases...
along the streamwise direction, which is caused by the gradual mixing of coolant with the mainstream and separation from the wall.

Figure 12. The time-averaged flow fields at $M = 3.0$, Baseline, $W_3/D = 1.4$, 1.2, 0.9 (from left to right).

Figure 12 shows the time-averaged flow fields in fluid oscillators with different feedback channel widths at $M = 3.0$. It can be seen from the velocity contour and streamline that the feedback channel width affects little the flow pattern, which can explain the slight difference in Figure 11.

**Modified configuration.** The main purpose is to propose a more compact structure the effectiveness is better than the traditional fluidic oscillator. However, with the decreasing of the inlet wedge, the cooling performance is becoming dramatically bad. From the study on the flow fields inside the case of the $W_3/D = 1$, the primary reason that influences the flow tendency and the recirculation bubble is the size of $W_4$, which makes the bad cooling effectiveness. And the effect of the $W_4$ can be seen in Figure 6. So, a modified fluidic oscillator structure ($W_3/D = 1$, $W_3/D = 4.5$, $W_3/D = 0.9$, and $W_4/D = 0.8$) with the smallest sizes were selected from the three geometric parameters, and the inlet throat $W_4$ is smaller than the inlet wedge width $W_3$. Figure 13 shows the geometry of the Compound SJ.

Figure 13. Internal geometrical structures of the Compound SJ.
Figure 14. The film cooling effectiveness at $M = 1.5$, 2.0, and 3.0, (a) Baseline and (b) Compound SJ.

Figure 14 shows the time-averaged film cooling effectiveness at $M = 1.5$, 2.0, and 3.0 with the Baseline and the Compound SJ. The film coverage of the Compound SJ is much larger than the Baseline in the contour of the cooling effectiveness, which means better cooling performance.

Figure 15. The laterally averaged effectiveness of the Baseline and the Compound SJ at (a) $M = 1.5$, (b) $M = 2.0$, (c) $M = 3.0$ and the area-averaged effectiveness (d).

Figure 15 shows the laterally averaged effectiveness and the area-averaged effectiveness of the Baseline and the Compound SJ. The laterally averaged effectiveness was averaged in spanwise direction within $-3 \leq Y/D \leq 3$ range and the cooling effectiveness of the Compound SJ is better than the Baseline at $M = 1.5$, 2.0, and 3.0. The area-averaged effectiveness was averaged in the region of the $-3 \leq Y/D \leq 3$ and $0 \leq X/D \leq 15$. Figure 15 (d) shows that the cooling
performance of the Compound SJ is much better than the Baseline at the same blowing ratios, especially at $M = 3.0$, the Compound SJ shows a 50% higher value than the Baseline.

**Conclusion**

A numerical study is conducted to investigate the impact of the internal structure of a fluidic oscillator on film cooling. The oscillation frequency and flow fields both inside and outside the oscillator are considered to uncover the mechanisms of the SJ. The widths of the inlet wedge $W_1$, the length of the mixing chamber $W_2$, and the widths of the feedback channel $W_3$ are studied, normalized by the width of the exit throat $D$. The major findings are as follows:

- The inlet wedge width $W_1$ controls the size of the recirculation bubble, the direction of the feedback flow, and external flow fields, which influence the film coverage. At $W_1/D = 1.8$, the flow fields inside the oscillator are close to the traditional one, have obvious feedback flow and recirculation bubble, and show a stable frequency. However, this structure has better cooling effectiveness than the traditional hole due to the role of separation and reattachment of coolant. By decreasing $W_1/D$ to 1.5, the feedback flow is not clear. Thus, the oscillation frequency is reduced. And with higher jet momentum, the film cooling effectiveness is decreased. When decreasing the inlet wedge width to $W_1/D = 1$, there is no feedback flow but still, a small recirculation bubble which leads to a reduced frequency and a sweeping motion. Because of the highest jet momentum compared with the above cases, this configuration shows the worst film cooling effectiveness.

- The length of the mixing chamber $W_2$ can impact the size of the recirculation bubble, which determines the jet spreading angle and frequency, then influences the external flow and the effect of cooling. When the length of the mixing chamber is decreased, the frequency is becoming higher and making the film coverage more uniform. With higher frequency, the momentum of the jet is becoming lower, resulting in better cooling effectiveness. Besides, the size of the anti-CRVP is bigger, which decreases the CRVP, and drives the coolant to stay closer to the wall.

- The feedback channel is generally expected to be the most important parameter that impacts the frequency of the external jet, but the results only show a slighter difference. With the decreasing of the width of the feedback channel, the film footprint and trend are almost approximate.

- The width of the inlet throat $W_4$ is an important parameter that can influence the flow pattern inside the oscillator, and to make the structure of the oscillator more compact, the Compound SJ has been proposed. Compared with the traditional hole, the Compound SJ shows improved adiabatic effectiveness at $M = 1.5, 2.0, and 3.0$.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Acronyms</th>
<th>X</th>
<th>Streamwise direction</th>
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<tbody>
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<td>CRVP</td>
<td>Y</td>
<td>Spanwise direction</td>
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<tr>
<td>SST</td>
<td>Z</td>
<td>Wall normal direction</td>
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<td>Film cooling effectiveness</td>
<td>$W_1$</td>
<td>Inlet wedge width</td>
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<td>$W_2$</td>
<td>Length of the control chamber</td>
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<td>$W_3$</td>
<td>Feedback channel width</td>
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<td>$W_4$</td>
<td>Inlet throat width</td>
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<td>$T_c$</td>
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<td>$T_m$</td>
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**References**


