Numerical Investigations of Wake and Shock Wave Effects on Film Cooling Performance in a Transonic Turbine Stage, Part 2 – Unsteadiness Effect in a 2D Rotating Passage

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ABSTRACT
Using the validated CFD model and realizable k-ε turbulence model from Part 1, the unsteadiness effects of shock waves and wake flow on the pressure coefficient and film cooling performance in a 2-D, rotating passage is studied in Part 2. Both time-dependent and time-averaged cooling effectiveness distributions on the rotor under the influence of shock waves and unsteady wake flows are presented with blowing ratios of 0.5, 1.0, and 1.5. The results show that (a) the unsteady wake passing caused by the blade rotation significantly influences the time-dependent pressure coefficients, especially in the forward region downstream of the stagnation point on both the suction and pressure sides on the rotor. This effect can be evidenced by the "wrapping behavior" of a strong vorticity field wrapping around the blade leading edge after the wake hit the blade (b) On the suction side, over the crown area of the rotor, the wake-induced pressure fluctuations are augmented by the weak shock wave emanating from the stator's trailing edge, appearing as a primary peak followed by a secondary peak. The weak oblique shock wave can be evidenced by the dense Mach contour curves. The effect of the wake (rotational) is distinguished from that of the shock (inviscid and non-rotational) by comparing the pressure gradient field with the vorticity field. (c) Similar effects of passing wakes and shock waves on the surface pressure on a rotor are observed on the film cooling effectiveness, except that these effects are of the same magnitude. (d) The effect of the blowing ratio on the static pressure distribution at the rotor wall is not significant, whereas its effect on the film cooling effectiveness is more pronounced.

NOMENCLATURE
\( T \) Time, s
\( T_{aw,c} \) Adiabatic wall temperature with film cooling, K
\( T_c \) Film cooling air static temperature, K
\( T_g \) Total temperature of the mainstream, K
\( \Delta t \) Time step, s
\( V \) Velocity, m/s
\( V_{circum} \) Circumferential rotor speed at the mid-span, m/s.

Greek Letters
\( \rho \) Cooling air density, kg/m\(^3\)
\( \eta \) Film cooling effectiveness

Subscripts
c Cooling air
g Mainstream gas

INTRODUCTION
Today, advanced aero-engines are operating at turbine inlet temperatures of around 2200K, which is far beyond the melting limits of existing super-alloy materials. Film cooling is a highly effective cooling technique used to protect temperature-sensitive components from the extremely high temperatures inside gas turbines. Since the 1970s, the investigation on film cooling performance has been making impressive progress in experimental and numerical aspects. Part 1 was focused on validating computational methodology applied to studying the effect of steady shock waves and wakes on turbine airfoils as well as the surface pressure distribution and film cooling performance on isolated stationary stators and rotors. This paper, as Part 2, is focused on studying the effects of unsteady shock waves and wakes on the pressure-flow field and film cooling performance in a rotating stator-rotor passage.

To generate the viscous wake of a relative movement between the stator and rotor, Funazaki et al. [1, 2] (1995) and Womack et al. [3] (2008) placed an array of rotating rods in front of a blunt body to experimentally study the influence of
the viscous wake and pulsating coolant injection on the film cooling characteristics over a stator. Lagrani et al. [4] (2001) detected the interaction between shock waves and film cooling effectiveness with a flat plate under subsonic and supersonic conditions (Mach number 0.8, 1.10-1.12). Their experiments concluded that the shock wave changes the fluid flow behaviors within the film hole as well as the temperature and pressure profiles near the film hole exit, resulting in a significant change in film cooling characteristics downstream of the film holes. Smith et al. [5] (2003) discussed the effect of shock waves on film cooling effectiveness over a transonic stator vane. He claimed that a relatively strong shock wave tends to retard the coolant ejection at the suction side, while it causes excess coolant effusion from the film holes at the pressure side. Didier et al. [6](2002) studied the heat transfer coefficient distribution over the rotor in a transonic turbine stage at a temperature much lower than that of a real turbine. Their experimental results revealed that the shock wave dominates the heat transfer near the leading edge of the rotor.

Miller and Urbassik et al. [7,8](2002, 2004) compared the actions of shock waves, viscous wakes, and potential flows on the heat transfer coefficient through 1.5 turbine stages. The comparisons indicated that the wake effect on the pressure distribution over rotor could be neglected. Loma et al. [9] (2008) studied the heat transfer characteristics of a rotor under different blowing ratios and pressure ratios. Their results indicated that the unsteady heat transfer in the front part of the rotor is mostly dominated by the impingement of the shock wave coming from the trailing edge of a stator, and the fluctuations caused by the shock wave at 15% of the rotor height can double in magnitude at 85% of the rotor height. Although there have been a large number of experimental studies on film cooling characteristics in transonic turbine stages, most of the experiments were carried out on stationary platforms or in simplified facilities due to the limitation of laboratory conditions and the costs.

Numerical approaches can remove some of the limitations of the experimental conditions, and therefore numerical approaches have been widely used to simulate rotating and transonic environments to supplement missing information that is difficult or expensive to obtain. Abhari et al. [10] (1992) used an in-house code UNSFLO and the Baldwin-Lomax (B-L) turbulence model to compare the numerical results of turbine rotor blade heat transfer with their experimental measurements. They concluded that the numerical simulation can very well predict the quantitative time history of heat flux distribution and about 90% of the measured integrated heat load to the blades. The discrepancy was explained due to an underestimation of the local shock strength in the numerical simulation. Abhari [11] (1996) adopted the same code to discuss the impact of stator-rotor interaction on the surface heat transfer of film cooled transonic turbine blades under steady and unsteady conditions. He concluded that the unsteady rotor-stator interaction results in the large-scale pulsations of the coolant injection flow out of the film holes, which lower the local adiabatic film cooling effectiveness on the pressure side the rotor by as much as 64 percent when compared to the steady-state case.

Korakianitis et al. [12](1993) used an inviscid model to numerically analyze the propagation of the pressure disturbances due to the potential flow and wake separately, as well as the combination of both of them. Garg et al. [13] (1999) simulated the heat transfer characteristics using three turbulence models, B-L, k-ω, and q-ω, under different pressure and temperature ratios, and their results indicated that the k-ω model can offer the best agreement with the experimental data overall. Using the B-L model, Nishizawa et al. [14] (2003) compared the film cooling effectiveness obtained by 3-D simulations of a stationary turbine stage with empirical correlations. They claimed that both the experiments and numerical analyses had some weaknesses: 1) rotating effects were not taken into consideration in the stationary rig, 2) the velocity and temperature distributions at the exits of the film holes were assigned with empirical correlations, which could result in a large error of film cooling effectiveness on the suction side of the rotor. Shyam et al. [15] (2011) used the low Reynolds number k-ε model to simulate pressure variations on rotors under the influences of wakes and shock waves. They compared the numerical results with their experimental data and indicated that the viscous wake impinging on rotors should be one of the major factors that contribute to the pressure field changes at the rotor leading edge. However, their simulation was not in good accordance with the experimental data in terms of shock wave and boundary layer transitions.

As mentioned above, in a transonic turbine stage, the unsteadiness caused by the viscous wake and shock wave from the stators as well the potential flow variations caused by rotation are important factors that significantly influence the film cooling behaviors of rotor blades. The objective of Part 2 is to employ the validated computational model with an appropriate turbulence model in Part 1 to investigate the unsteady wake and shock effects induced by the rotating movement between the stator and rotor on the film cooling characteristics over the rotor.

Computational Methodology

Using the validated CFD model and realizable k-ε turbulence model from Part 1, the unsteadiness effects of shock waves and wake flow on the pressure coefficient and film cooling characteristics in a 2-D, rotating passage is performed. A pressure-based solver is applied to solve the coupling conservation equations of continuity, Navier-Stokes (momentum), energy, and turbulence transport using the commercial code, Ansys/Fluent. The second-order upwind scheme is used in the discretization of the convective terms.

Geometry and Mesh Scheme

The geometric parameters of an entire turbine stage presented by Thulin [16] are used to investigate the influence of unsteady effects on the film cooling characteristics. But, in order to reduce computational time in the numerical simulations, the number ratio of stator vanes to rotor blades is changed from the original ratio, 24:56, to 24:48 (1:2). Therefore, the entire stage can be conveniently structured by employing a periodic boundary condition with one stator vane and two rotor blades instead of three vanes with seven blades. The parameters and operating conditions are summarized in Table 1. Three film holes are positioned in the rotor blade, each with one hole on the pressure side, suction side, and leading edge, respectively in this 2D geometry. The
corresponding computational domain and meshes are shown in Fig. 1. Compared to an unstructured grid, a structured grid can offer more precise results in the regions with steep gradients through the same cell number, but less iteration steps, therefore a structured grid composed of quadrilateral cells is adopted in this part. The mesh is generated with ICEM and refined so that the first near-wall cells yield \( y^+ \) values of less than 1 within the boundary layers. To speed up the convergence, an interpolation method is applied through mapping the results from a coarse mesh to a fine mesh. The mesh independence was tested by three progressive finer grids with cell numbers of 34,648; 77,061; and 151,078 in the stator stage. The experimental data of the stator under the subsonic conditions provided by Kopper [17] were used for comparison. The results indicated that the differences of static pressures among these three grids were less than 1%. Therefore, the mesh number of 77,061 in the stator stage is used. When the grid is extended to the rotating blade stage, the total cell number is increased to 251,864.

### Table 1 Mid-span mean line stage parameters from Thulin [16]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vane number</td>
<td>24</td>
</tr>
<tr>
<td>Blade number</td>
<td>48</td>
</tr>
<tr>
<td>Vane axial chord</td>
<td>1.715 in</td>
</tr>
<tr>
<td>Blade axial Chord</td>
<td>1.160 in</td>
</tr>
<tr>
<td>Vane inlet angle</td>
<td>90 °</td>
</tr>
<tr>
<td>Vane outlet angle</td>
<td>10.312 °</td>
</tr>
<tr>
<td>Blade inlet angle</td>
<td>34 °</td>
</tr>
<tr>
<td>Blade outlet angle</td>
<td>16.8 °</td>
</tr>
<tr>
<td>Tip radius</td>
<td>16.149 in</td>
</tr>
<tr>
<td>Hub radius</td>
<td>13.88 in</td>
</tr>
<tr>
<td>Tip clearance</td>
<td>0.0186 in</td>
</tr>
<tr>
<td>Vane leading edge diameter</td>
<td>0.525 in</td>
</tr>
<tr>
<td>Vane trailing edge diameter</td>
<td>0.064 in</td>
</tr>
<tr>
<td>Blade leading edge diameter</td>
<td>0.155 in</td>
</tr>
<tr>
<td>Blade trailing edge diameter</td>
<td>0.06 in</td>
</tr>
<tr>
<td>Stage gap distance</td>
<td>0.6 in</td>
</tr>
<tr>
<td>Inlet total temperature</td>
<td>2614.73 °F</td>
</tr>
<tr>
<td>Inlet total pressure</td>
<td>192.08 psi</td>
</tr>
<tr>
<td>Inlet flow angle</td>
<td>90 °</td>
</tr>
<tr>
<td>Inlet Mach number</td>
<td>0.09</td>
</tr>
<tr>
<td>Rotor angular speed</td>
<td>13866 rpm</td>
</tr>
<tr>
<td>Pressure ratio</td>
<td>4.02</td>
</tr>
<tr>
<td>Reaction level</td>
<td>43%</td>
</tr>
<tr>
<td>Inlet Flow parameters</td>
<td>16.984</td>
</tr>
</tbody>
</table>

The adiabatic film cooling effectiveness is defined as:

\[
\eta = \frac{T_g - T_{av,c}}{T_g - T_c} \quad (1)
\]

Note that in a compressible flow, viscous dissipation will add additional heating load to the wall, so the film cooling has to perform better to achieve the same \( \eta \)-value as in the incompressible flow. The pressure coefficient is defined as:

\[
C_d = \frac{P}{P_{in} V_{circum}^2} \quad (2)
\]

where \( P \), \( \rho_{in} \), and \( V_{circum} \) are static pressure, cooling air density at the film hole inlet, and circumferential velocity at the mid-span, respectively.

The air, treated as an ideal gas, is used as the cooling medium. Its corresponding viscosity, conductivity, and heat capability at different temperatures from 300 to 2100 K are provided by polynomial expressions as used Part 1.

### RESULTS AND ANALYSIS

#### Unsteady Wake Passing

It is very difficult to measure the velocity and temperature within a rotating passage non-intrusively. Thus, experimental investigations on the unsteadiness of fluid flow field due to blade rotating have been rarely presented in detail. Computational simulation can help to describe the unsteady process if the model is adequate. Following the analysis in Part 1, the realizable \( k-\epsilon \) turbulence model is selected to simulate the instantaneous fields of fluid flow and static pressure. The results are first shown in the vorticity distribution in Fig. 2 to...
To capture the vortical characteristics of the wake flows, we demonstrate the dynamic feature of the interaction between the moving blades and the passing wakes, and provide the vorticity field variation over one pitch period in six time instants (each time interval is 30μs). This process begins at Fig. 3a, and a viscous wake from the trailing edge of the stator, manifesting itself as a strong vortex band, starts to enter the rotor (R) passage. It is interesting to note that the vorticity field of the wake shows a band of low vorticity core sandwiched between two strong vortical strings. The band-width of the low vorticity core apparently corresponds to the finite thickness of the stator's trailing edge and can be clearly seen in Figs. 3c to 3d. In Fig. 3b, the rotor moves itself downwards, and impinges the viscous wake with its crown (C). From Figs. 3c to 3e, the wake flows around the crown gradually and the vortex moves itself into the rotor passage. Up to Fig. 3f, as most of the viscous wake passes through and out of the rotor passage, a new wake approaches after 30μs, and the cycle starts over again from Fig. 3a. The corresponding variation in the mid-span static pressure at the rotor wall is shown in Fig. 4. The static pressure's fluctuations can be seen by comparing the pressure curves from Fig. 4(a) to (e). For example, a big pressure drop on the suction side can be seen between (e) and (f); it is associated with the wake's departure from the suction surface as seen in Figs. 3(e) and (f).

When the instantaneous static distributions of consecutive time intervals are stacked up in one figure, it can be represented as a three-dimensional plot as Fig. 4(b). It can be seen that in the forward region from the leading edge to approximately 55% of chord length, non-uniform variation of static pressure is seen in the time coordinate, indicating the impact of wake passing on surface pressure in the forward area. In the rear region, the pressure distribution is more uniform, indicating its insensitivity to wake passing.

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![Fig. 3 Close-up views of dynamic variation of vorticity field corresponding to wake passing through rotor passage during one stator pitch period.](image)

![Fig. 4 Mid-span static pressure variation on the rotor corresponding to wake passing during one stator pitch period of 180 μs: (a) in six 30 μs time intervals (b) in 3-D plot.](image)
Unsteady Interaction between Wake and Oblique Shock Wave Passing

To monitor more specific local information, nine monitoring locations are specified; they are located at 10%, 30%, 65%, 94% of the blade axial chord on the pressure side and 0%, 10%, 35%, 65%, and 99% on the pressure side as shown in Fig. 5. The corresponding fluctuations of surface pressure and temperature due to the unsteadiness caused by the viscous wakes and shock waves from the stator and the potential flow variation introduced by the relative movement between the stator and rotor are presented and analyzed further on.

![Fig. 5 Positions of 9 monitoring points on the rotor.](image)

Static Pressure Characteristics

Figure 6a shows the history of the pressure coefficient at four locations on the pressure side of the rotor over three periods with the blowing ratio 1.0. The corresponding pressure values at six specific instants have been previously shown in Fig. 4. By comparing the peak and valley locations in Fig. 6a, it is interesting to see that the fluctuations in the pressure coefficients at point 1 shift phase through points 2 and 3, and at point 4 the fluctuation phase is almost 180-degree opposite to that at point 1. The phase shift from upstream locations to downstream locations reflects the propagation speed of the wakes. This phase shift can be more clearly seen on the suction side in Fig. 6b. However, in addition to the distinctive primary $C_d$ peaks in Fig. 6, a secondary peak follows each primary peak. It is speculated that these secondary peaks are caused by the oblique shock waves emanating from the upstream stator's trailing edge. To separate the effect of wakes and shock waves, the pressure gradients (shock strength) and vorticity magnitudes at three time instants are plotted and compared in Fig. 7 and Fig. 8, respectively.

![Fig. 6 Unsteady static pressure traces (at nine monitoring points with the blowing ratio $Br = 1.0$.](image)

![Fig. 7 Contour of pressure gradient magnitude (Pa/m) showing the shock wave hitting point 7, 6, and 5, at time instants of (a) 6.768 ms; (b) 6.786 ms; (c) 6.798 ms.](image)

![Fig. 8 Contours of vorticity magnitude showing wake passing at exactly three instants corresponding to shock wave dynamics hitting point 7, 6, and 5 in Fig. 7.](image)

In the pressure gradient strength plot in Fig. 7, both the wake and oblique shocks can be seen as the regions of relatively high pressure gradient. In Fig. 8, however, only the wake region can be seen in the vorticity plot because the oblique shocks are inviscid and non-rotational. The argument that supports the speculation that the secondary peaks are caused by the shock wave is validated in the vorticity plot in Fig. 8, wherein it can be clearly seen that the wake has already hit and wrapped around the rotor's crown region while the shock wave has just hit the crown region in Fig. 7. The hollow arrows in Fig. 7 show the starting point of the shock trace,
while the black arrows mark the impingement points of the trailing edge shocks on the blade wall.

In terms of timeline, the shock wave impinges on point 7 in Fig. 7a, then, 12 μs later, it hits point 6 in Fig. 7b, and another 12μs later it hits point 5 in Fig. 7c. This dynamic sequence is reflected as the corresponding secondary peak shifts from point 7 to point 5 in Fig. 6.

**Effect of Unsteadiness on Film Cooling Effectiveness with Br = 1.0**

The values of the adiabatic film cooling effectiveness ($\eta$) of nine monitoring points at blowing ratio of 1.0 are shown in Figs. 9a and b. The cooling effectiveness of points 1, 3, and 4 is about 0.16 and its value at points 7, 8, and 9 decreases from a mean value of 0.36 to 0.16. Point 2 has a high cooling effectiveness, while points 5 and 6 have an even higher cooling effectiveness, although the effectiveness value itself varies significantly.

The largest fluctuation of the $\eta$-value from 0.88 to 0.68 at point 2 on the pressure side of the rotor translates to a wall temperature swing of 300 K, which exemplifies the potential of inducing a large thermal stress in the leading edge region caused by the passing wake. In the meantime, on the suction side of the rotor, the cooling effectiveness is significantly affected by the interactions of both the passing wake and oblique waves with large fluctuating amplitudes of 40% for point 5 and 20% for point 6. The physics of the dynamic effect of wake passing and shock waves on the static pressure have been explained earlier. The only difference is that their effects on the cooling performance result in two peaks of the same magnitude rather than a primary peak followed by a secondary peak in the static pressure fluctuation (Fig. 9 vs. Fig. 6).

**Fig. 9 Unsteady film cooling effectiveness at nine locations on the stator with the blowing ratio Br = 1.0.**

The temperature and flow fields near the rotor leading edge are shown in Figs. 10 and 11, respectively. At point 1, between two film injection holes, the cooling effectiveness is the lowest because the wall temperature near the leading edge approaches the stagnation temperature and the cooling air ejected from the neighboring two film holes hardly covers this region, as shown in Fig. 10.

**Fig. 10 Static temperature contours near the leading edge at blowing ratio Br=1.0.**

**Fig. 11 Flow field the rotor leading edge at an instant at Br = 1.0.**

**Influence of Blowing Ratio on Aerodynamic Characteristics**

The influences of blowing ratio on aerodynamic characteristics at the monitoring points are investigated at three blowing ratios $Br = 0.5$, 1.0, and 1.5. The static pressure variations at 9 points of the rotor wall over three pitch periods are shown in Fig 12. The result clearly shows that the effect of blowing ratio on the static wall pressure is not significant. Unsteadiness doesn't seem to change this negligible effect.

**Influence of Blowing Ratio on Film Cooling Effectiveness**

Different from the surface pressure response to the blowing ratio changes, the effect of blowing ratio on adiabatic film cooling effectiveness is significant, as shown in Fig. 13. In general, the cooling effectiveness increases with the
blowing ratio at all points except for point 2, where the highest cooling effectiveness appears at \( Br = 1.0 \). This phenomenon can be explained by examining the close-up view of the temperature contours near point 2 in Fig. 14. At a low blowing ratio of 0.5, the momentum of the cooling air ejected from the film holes is not strong enough to effectively transport the cooling air in order to protect point 2. Increasing the blowing ratio to 1.0, the jet flow appears to have sufficient momentum to move downstream, nicely blanketing the region surrounding point 2 with cooling air. However, as the blowing ratio further increases to 1.5, the film cooling jet seems to be too strong, moving with enhanced mixing, which reduces the effectiveness of film cooling.

At point 3, the effect of unsteadiness appears to be amplified by increasing the blowing ratio to 1.5, resulting in a largely enhanced cooling effectiveness accompanied with a large fluctuation with an amplitude of 30\% of the mean value.

In the poorly cooled region, at point 1 on the leading edge and at points 4 and 9 on the trailing edge, increasing the blowing ratio has negligible effect on the cooling effectiveness.

The effect of the passing wake and shock waves on the cooling effectiveness increases with the blowing ratio, but the increased magnitude slows down as blowing ratio further increases to 1.5.

Fig.12 Effect of blowing ratio (0.5, 1.0 and 1.5) on surface pressure variation on the rotor appears negligible
Fig. 13 Effect of blow ratio (0.5, 1.0 and 1.5) on film cooling effectiveness over the rotor

Time Averaged Film Cooling Effectiveness

The distributions of the time-averaged cooling effectiveness on the rotor over one pitch rotating period with blowing ratios of 0.5, 1.0, and 1.5 are plotted in Fig. 15. The cooling effectiveness over the suction side increases monotonically with the blowing ratio, whereas, along the pressure side, the cooling effectiveness of the cases with Br = 1.0 and 0.5 exceeds that of the Br = 1.5 case at 0.5 and 0.2 chord lengths. This can be explained by the fact that at Br=1.5 the highest mass flow rate of cooling air forms a coverage from 0 to 0.2 chord length, up to 0.5 chord length; the cooling air departs from the surface in the region from 0.2 to 0.5 chord
length and thus the film cooling effectiveness falls, as demonstrated in Fig.14(c).

![Fig. 15 Time averaged film cooling effectiveness for Br=0.5, 1.0 and 1.5.]

**CONCLUSION**

This numerical research focuses on investigating the effects of unsteady wakes and shock waves on the aerodynamic and film cooling performances of a transonic rotating stator-rotor stage. The conclusions are:

1. The unsteady wake passing caused by the blade rotation significantly triggers large fluctuations of surface static pressure, especially in the forward region downstream of the stagnation point on both the suction and pressure sides. This effect can be observed from the "wrapping behavior" of a strong vorticity field located near the blade leading edge after the wake has hit the blade.

2. On the suction side over the crown area of the rotor, the wake-induced pressure fluctuations are augmented by the weak shock wave emanating from the stator's trailing edge, appearing as a primary peak followed by a secondary peak. The weak oblique shock wave can be located by observing the dense Mach contour curves. The effect of the wake (rotational) is distinguished from that of the shock (inviscid and non-rotational) by comparing the pressure gradient field with the vorticity field.

3. Similar effects of passing wakes and shock waves on the surface pressure are observed on the film cooling effectiveness, except effects of wakes and shock waves are of the same magnitude.

4. The effect of the blowing ratio on the static pressure distribution at the rotor wall is not significant, whereas its effect on the film cooling effectiveness is more pronounced. The effectiveness typically increases with increased blowing ratio except in the region with highest curvature on the pressure surface.

5. The time-averaged cooling effectiveness along the suction side of rotor increases monotonically with the blowing ratio, whereas, on the pressure side, the cooling effectiveness first increases with blowing ratio, and then decreases just downstream of the film holes as blowing ratio further increases to Br = 1.5.

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