Influences of Pressure ratio and Fluid Temperature on overall Cooling Performances of Ribbed Channel with Film Holes

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Abstract. The previous experiments of overall cooling performances were most conducted using simplified models and under the similar temperature ratio of mainstream to cooling air with real gas turbine operations, and ambient outlet pressure. To discuss the reliability of this type of experimental data, this paper exhibits two series of numerical simulations. Using a real E3 blade as model, which has two-pass rib-roughened channel with inclined film holes, numerical simulations are carried out at the same temperature ratio and pressure ratio, but different fluid temperatures including mainstream and cooling air, and different outlet pressure. The numerical results reveal two important conclusions: 1) At the same outlet pressure, the overall cooling effectiveness on PS is not sensitive to the fluid temperatures, but on SS in the region between two rows of film holes, a higher fluid temperature corresponds to a higher cooling effectiveness. 2) At the same pressure ratio of inlet to outlet, the overall cooling effectiveness on PS and SS is not sensitive to the outlet pressure and fluid temperature.

1. Introduction
To increase thrust and efficiency of aero-engine, it is an effective approach to increase turbine inlet temperature (TIT). Today, advanced gas turbines are operated at the temperatures much higher than the allowable temperature of turbine blade materials, and therefore it is necessary to develop more effective cooling techniques. In general, the current blade cooling technique can be divided into internal convective cooling and external film cooling. The internal convective cooling effect can be enhanced by adding ribs on channel wall, and the film cooling effect can be also enhanced by the cooling air passing through the ribbed channel.

There have been a lot of investigations on singular film cooling performances, and the main topic focuses on the shape, angle and position of film holes, blowing ratio, momentum ratio and density ratio of mainstream to cooling air. Bogard & Thole [1] indicted that the influence factors of film cooling characteristics should also include the spacing ratio of hole length to diameter, the arrangement of holes, turbulence intensity, and the surface curvature and roughness of film hole location. Han & Chandra [2] studied heat transfer performances and pressure loss of different ribbed channel, and indicated that the channel with 60 degree ribs can provide the best heat transfer performances, but its pressure loss is also the largest.
To study the coupled performances of film cooling with ribbed channel, Wilfert & Wolff [3] conducted a series of experiments, and deemed that the film cooling effectiveness is influenced by ribbed channel, because the cooling air injected from ribbed channel can increase the cooling effectiveness averaged over transverse distance 56%. Through experiment and simulation, Kissel et al. [4] compared the film cooling effects between three cooling air transport methods, i.e. from smooth or ribbed channel, and from a plenum chamber with a constant static pressure. The comparisons indicated that the cooling air injected from the ribbed channel can get higher average heat transfer coefficients. Through numerical simulations, Wang et al.[5] analyzed the influences of rib shape on film cooling effect, and indicated that in comparison square ribs and pin ribs, continuous round ribs can obtain higher cooling effectiveness. Bunker & Bailey[6] measured the influences of the ribs on the mass flow rate of cooling air. Chanteloup & Bölcs[7,8] studied the film cooling characteristics of fluid flow and heat transfer of a two-pass ribbed channel, and revealed that the asymmetrical vortices in film holes can enhance the local heat transfer near the film holes.

Although there have been a large number investigations on the heat transfer and fluid flow performances of film cooling combined with ribbed channel, therein numerical investigations were most conducted by simplified channel models, and experimental investigations under very low temperature ratio and pressure ratio. Using simplified models, Greiner et al.[9] analyzed the conclusion differences
between the experimental conditions of high or low temperatures, large or small density ratio and blowing ratio. Up to now, there is still a lack of the investigations on the fluid flow and heat transfer performances of the film cooling coupled with ribbed channel in the real gas turbine operation environment and using real turbine blade model.

This paper exhibits a numerical investigation on the coupled effect of film cooling with ribbed channel, under the same temperature ratio and pressure ratio, numerical simulations are carried out using a real turbine blade model, and the influences of fluid temperature and outlet pressure on the cooling effect are discussed. The aim of this work is to provide the turbine blade designers with a more valuable reference.

2. Geometric Model
The two-pass channel used in this work is from a high pressure turbine blade of Pratt & Whitney Energy Efficient Engine (E3) program reported by Thulin et al. [10]. There are 54 blades on the annular cascade. The axial chord \((C_{ax})\) of the blade is 29.46mm, the pitch of the annular cascade is 42.64mm. Other detailed parameters of the blade and cascade are given in Table 1 and Fig. 1. The rib-roughened channel consists of an inlet section, a round turn and an outlet section. On PS and SS, there are 11 ribs with 60° in the inlet pass, 10 ribs with 60° in outlet pass, and the space between two adjacent ribs is 4mm. In the middle of the two neighbouring ribs, there is a film hole with an incline angle of 60°, and detailed parameters of the channel, rib and film hole are shown in Tab. 2.

![Figure 1. Schematic diagram of the blade with two pass ribbed channel.](image)

| parameter | value |
|-----------|-------|
| \(P/mm\) | 42.64 |
| \(C_{ax}/mm\) | 29.46 |
| \(\alpha/deg\) | 33.7 |
| \(\beta/deg\) | 16.9 |
| \(D_t/mm\) | 1.52 |
| \(D_l/mm\) | 3.93 |
| \(H/mm\) | 58.14 |

| parameter | value |
|-----------|-------|
| \(p/mm\) | 4 |
| \(e/mm\) | 0.6 |
| \(d/mm\) | 0.5 |
| \(D_p/mm\) | 6.7 |
| \(N_{i,PS}\) | 11 |
| \(N_{i,SS}\) | 11 |
| \(N_{o,PS}\) | 10 |
| \(N_{o,SS}\) | 10 |

3. Numerical Model

3.1. Computational domain and boundary conditions
To reduce the influences of mainstream on the inlet and outlet flow, the fluid region is extended 1.5 \(C_{ax}\) upstream from the leading edge of the blade and 3.5 \(C_{ax}\) downstream from the trailing edge, respectively.
The solid domain contains the blade, channel wall and endwall, and the fluid domain includes cooling air and mainstream.

The mainstream and cooling air are deemed as ideal gas, and their viscosity is calculated by the Sutherland law, and specific heat capacity and thermal conductivity are polynomial functions of temperature given by Holman [11]. The solid blade is nickel-based single crystal alloy CMSX-4. The density of the alloy is a linear function of temperature, and its specific heat capacity and thermal conductivity are functions of temperature, which are fitted from the data provided by Matsushita et al. [12] and Mills et al. [13], respectively.

The numerical algorithm of fluid-solid conjugated heat transfer is used in this work. To validate the numerical method, the experimental data and the same condition of Thulin et al. [10] are used, as shown in Table 3 Case 1. To analyze the influences of mainstream outlet pressure on the cooling effect, numerical simulations are carried out at the same temperature ratio of 1.81, and same blowing ratio of 1.19, but different outlet pressures. To keep the same mainstream Reynolds number of 2.02E+05 and blowing ratio, the mass flow rates of the mainstream and cooling air are changed as shown in Table.3

| boundary condition | Case 1 | Case 2 | Case 3 | Case 4 | Case 5 |
|-------------------|--------|--------|--------|--------|--------|
| T∞                | 1500k  | 873.15k| 494.4k | 873.15k| 494.4k |
| TC                | 829.15k| 482.4k | 273.15k| 482.4k | 273.15k|
| m∞                | 0.5686kg/s | 0.4135kg/s | 0.2865kg/s | 0.1435kg/s | 0.2865kg/s |
| mC                | 0.010kg/s | 0.00728kg/s | 0.00504kg/s | 0.00728kg/s | 0.00504kg/s |
| Po                | 3atm   | 3atm   | 3atm   | 2atm   | 1atm   |
| Pi                | 5.4atm | 4.2atm | 3.5atm | 3.6atm | 1.8atm |
| Pi/Po              | 1.8    | 1.4    | 1.2    | 1.8    | 1.8    |

3.2. Computational settings
The three-dimensional steady-state turbulent flow is solved by using the commercial software FLUENT 15.0, and the shear stress transport k-ω model is used to solve the Reynolds-averaged Navier-Stokes equations. In order to speed up the solution process, the second-order discretization scheme is applied to determine the equation solutions of momentum, energy and turbulent transport, while the PRESTO! scheme is selected for the pressure term in order to speed up the solution process. The simulation is thought to be converged when the residuals of the continuity, momentum, and turbulence equations are less than 1.0×10⁻⁴, and 1.0×10⁻⁶ for the energy equation. At the same time, check that the unbalanced mass is less than 0.5%.
3.3. **Mesh independence test**

The commercial software ANSYS Fluent Meshing 15.0 is selected to generate the meshes in fluid and solid domains. To acquire a high resolution of flow characteristics with low costs, the solid domain consists of only polyhedral cells, the fluid domain is meshed with polyhedral cells for the main region, but with prismatic cells near the region adjacent to the walls, and there are 10 layers of prismatic cells in the near-wall regions to ensure the averaged value of $y^+$ close to 1.0.

To prove the mesh independence of numerical results, three meshes are used. They are coarse, medium and fine grids with element numbers of 19471043, 23437203, and 30014644. Numerical simulations are carried out under the condition Case 1 in Tab. 3 using the three grids, and the wall temperature distributions on the middle cross section of the blade are compared. The maximal deviation between the coarse and medium meshes is about 5% but less than 2% between the medium and fine meshes. Hence, the medium mesh is used for the subsequent numerical simulations, and Figure 3 illustrates the mesh.

![Figure 3. Numerical mesh in solid and fluid domains.](image)

3.4. **Validation of the numerical method**

Using the experimental condition of Wang et al. [14], numerical calculations are conducted, and the numerical results are compared with the experimental data. In the validation test use the same mesh method, turbulence model, and residual criteria. Fig.4 shows the temperature distribution in a mainstream oriented line. As is shown in Fig. 4, a similar temperature trend exists, the maximum temperature difference between the calculation results to the experiment data in the entire surface is less than 10%, and therefore the numerical method used in the following simulations can be confirmed.

![Figure 4. Numerical method validation using experimental data[14].](image)
4. Numerical Results and Discussions

4.1. Influences of temperatures on cooling effectiveness

To analyze the influences of the fluid temperatures including mainstream and cooling air on the cooling effectiveness, numerical calculations are conducted at the same outlet pressure 3atm and the same temperature ratio 1.82, but different fluid temperatures (Case 1, 2, and 3 in Table 3). The numerical results of the cooling effectiveness distributions on PS and SS are shown in Figure 5. It is very interesting phenomenon, i.e., although the fluid temperatures are different, under the same temperature ratio and outlet pressure, the overall cooling effectiveness contributed by film cooling and ribbed enhanced heat transfer still keeps very similar variation trend, only in region 1 of PS, one can find that a lower temperature correspond to lower cooling effectiveness. On SS surface in region 2, especially between two rows of film holes, the cooling effectiveness with the highest mainstream temperature of 1500K is significantly higher than the other two cases.

To provide a quantitative comparison, Figure 6 exhibits the averaged cooling effectiveness over the total blade height. From Figure 6, one can find that on PS surface, the averaged cooling effectiveness are almost independent on the fluid temperatures, but on SS surface, higher fluid temperatures correspond to higher cooling effectiveness, especially in the region between two rows of film holes, and the largest different between Case 1 and Case 3 reach 28.3%, and between Case 1 and Case 2 21.5%.
4.2. Influences of outlet pressure on cooling effectiveness

Under different outlet pressures, numerical simulations are carried out under the conditions of Case 1, 4, and 5 in Table 3, while the pressure ratio and temperature ratio keep unchanged. Figure 7 exhibits the cooling effectiveness distributions on PS and SS. It is clear that with a decrease of outlet pressure, the cooling effectiveness between two rows of film holes increases slightly. To describe quantitatively the differences of the cooling effectiveness between the different outlet pressures, Figure 8 shows the cooling effectiveness averaged over the blade height. This is another interesting phenomenon, the cooling effectiveness is almost independent on the outlet pressure, because the largest difference between Case 1, 4, and 5 is only 4.8%.

Figure 6. Averaged cooling effectiveness over the height of the blade.

Figure 7. Cooling effectiveness distributions in Case 1, 4, and 5.
5. Conclusions
The previous experimental investigations on overall cooling performance of the ribbed channel adding film cooling were carried most under the same temperature ratio and ambient outlet pressure. To estimate the reliability of the experimental data, this paper exhibits a numerical investigation on the influences of the fluid temperature and outlet pressure on the cooling performances. The simulations are conducted at the same temperature ratio, which is close to real gas turbine operation conditions, but the fluid temperatures including mainstream and cooling air are changed, and the outlet pressure is also changed. Through the analysis of the numerical results, the following important conclusions can be drawn:

1) At the different pressure ratio of mainstream inlet to outlet and the same temperature ratio of mainstream to cooling air, when the outlet pressure is consistent, the effects of pressure ratio on PS cooling performances are not significant, but on SS between two rows of film holes, the effects are obvious, because the largest difference in the averaged cooling effectiveness can be larger than 28%.

2) At the same pressure ratio of mainstream inlet to outlet and the different inlet temperature of cooling air, when the temperature ratio of mainstream inlet to outlet is consistent, the inlet temperature is not sensitive to the cooling effectiveness. The largest difference in the averaged cooling effectiveness is less than 5%.

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