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Dominant flow structure in the squealer tip gap and its impact on turbine aerodynamic performance

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ABSTRACT

Tip leakage loss reduction is important for improving the turbine aerodynamic performance. In this paper, the flow field of a transonic high pressure turbine stage with a squealer tip is numerically investigated. The physical mechanism of flow structures inside the cavity that control leakage loss is presented, which is obtained by analyzing the evolution of the flow structures and its influence on the leakage flow rate and momentum at the gap outlet. The impacts of the aerodynamic conditions and geometric parameters, such as blade loading distributions in the tip region, squealer heights, and gap heights, on leakage loss reduction are also discussed. The results show that the scraping vortex generated inside the cavity is the dominant flow structure affecting turbine aerodynamic performance. An aero-labyrinth liked sealing effect is formed by the scraping vortex, which increases the energy dissipation of the leakage flow inside the gap and reduces the equivalent flow area at the gap outlet. The discharge coefficient of the squealer tip is therefore decreased, and the tip leakage loss is reduced accordingly. Variations in the blade loading distribution in the tip region and the squealer geometry change the scraping vortex characteristics, such as the size, intensity, and its position inside the cavity, resulting in a different controlling effect on leakage loss. By reasonable blade tip loading distribution and squealer tip geometry for organizing scraping vortex characteristics, the squealer tip can improve the turbine aerodynamic performance effectively.

1. Introduction

Due to the existing radial gap between the rotating blades and the stationary casing of a turbine, a tip leakage flow driven by the pressure difference between the blade pressure side and suction side is formed. When the leakage flow inevitably mixes with main flow, the leakage loss occurs, which accounts for about 1/3 of the total blade passage loss in unshrouded turbines [1]. This large loss highlights the importance of controlling tip leakage loss for efficient operation of turbine components and even the whole turbomachinery.

There are two types of blades in turbines, i.e. shrouded blades [2,3] and unshrouded blades. For unshrouded blades, different tip shape designs are commonly utilized for reduction of leakage loss, mainly including winglets [4,5] and squealers [6–8]. Squealer tips have three typical types, tips with a suction side squealer, a pressure side squealer or a double side squealer. The double side squealer tip, which is referred as a squealer tip in this paper, has out of these gained the most attention because of its excellent aerodynamic performance [9]. This squealer tip can separate the flow at the top of the two squealers, meaning it can block the leakage flow twice and enhance the mixing of the leakage flow inside the cavity. These actions bring a smaller discharge coefficient that reduces more leakage loss than that of flat tips and single squealer tips [10,11]. The aerodynamic advantage of the squealer tip is confirmed by many researchers [12,13]. For example, when comparing the squealer tip to the flat tip, Krishnababu et al. [14] noticed a decrease of 8% in leakage mass flow and an obvious reduction of total pressure loss coefficient; Zhou [15] found that the loss produced by the squealer tip was about 13% lower than that of the flat tip in a transonic turbine cascade; Kegalj et al. [16] applied a squealer tip in

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a $1^{1/2}$ stage turbine and improved the isentropic efficiency of a rotor stage by 0.24%.

With deepening study of the squealer tip, researchers have noticed that there are many vortices residing inside the cavity [17–19], such as the cavity vortex, the scraping vortex and the corner vortex. When these vortices' characteristics change, they alter the mixing of leakage flow inside the cavity and the separation bubble at the top of the suction side squealer, affecting the controlling effect of squealer tip on leakage flow [20]. Since becoming aware of the importance of flow structures inside the cavity, researchers have begun to study the physical mechanism of flow structures that control leakage flow and obtained some different explanations. Coull et al. [21] argued that the cavity vortex entrains and convects some part of the leakage flow, giving a high sealing effectiveness because of a significant mixing inside the cavity. Virdi et al. [22] emphasized that the friction pulling force of relative moving casing helps the formation and enhancement of the cavity vortex, which wraps leakage flow inside the cavity and intensifies mixing. Yang et al. [23,24] showed that a circumfluence vortex is formed near the moving casing. This vortex makes the leakage flow turn and impinge onto the cavity floor. At present, according to the author's knowledge, a consistent conclusion has not yet been proposed on the physical mechanism of flow structures that control leakage flow, and there seems to be no public paper investigating the dominant flow structure that affects the aerodynamic benefits of the squealer tip.

For further improvement of turbine aerodynamic performance, squealer shape optimization has been investigated in several studies. In order to enhance the blockage effect of the flow separation at the top of pressure side squealer, Prakash et al. [25] suggested a squealer tip with an inclined pressure side squealer rim. The inclined rim can increase the blocked area through a larger flow separation in the pressure side squealer tip gap, enhancing leakage flow control compared to a conventional one. The aerodynamic benefit of this squealer tip has been confirmed by other researchers [26]. Some squealer shape optimizations have also been done to change flow structures inside the cavity for the enhancement of leakage loss control. For example, Mischo et al. [27,28] modified both generation and interaction of vortices inside the cavity by variation of the squealer geometry. By doing so, the recirculation zone in the front part of the cavity is eliminated, resulting that a lower leakage loss generated compared to that with a base line squealer tip. In other researches, the combination of squealers and winglets has also been used for leakage loss reduction [19,24]. Because of a longer distance between the two squealer rims, the flow mixing inside the cavity is enhanced and the size of the separation bubble at the top of the suction side squealer is increased, which effectively reduce leakage loss [29]. From the results of these studies, it is clear that certain aerodynamic benefits have successfully been obtained by the squealer geometry optimizations or improvements. However, because no dominant flow structure has been determined, along with its characteristics' effect on leakage loss control, the investigations of squealer geometry improvement and optimization are so far still in a stage of individual attempts.

To provide a better understanding of the physical mechanism of the flow structures inside the cavity that control leakage loss, this paper relates some aerodynamic parameters at the squealer tip gap outlet to the flow structures evolution. Based on this relationship, the dominant flow structure inside the cavity is captured and validated. The influence of its evolution on leakage flow control is further explained. From point of view of the dominant flow structure, the impacts of blade loading distributions in the tip region and squealer geometric parameters on leakage flow control are investigated. Meanwhile, a squealer tip design method is also discussed, which provides a guide for better design of high performance turbine blades.

2. Computational setup

2.1. Turbine in study

The turbine geometry used in this study is a transonic single stage turbine of the Institute of Thermal Turbomachinery and Machine Dynamics (TTM) [30], as shown in Fig. 1. The main geometric parameters and the operation conditions of the turbine stage are listed in Table 1.

The study of tip leakage flow in this paper is carried out based on two types of tip, a flat tip and a squealer tip with an inclined pressure side squealer rim. Fig. 2 shows the geometry of the squealer tip, with its smooth connection between the pressure side of blade and the squealer rim. The height and the width of the squealers are 1.5τ and 1.0τ, respectively. The inclination of the pressure side squealer rim is 60°, which gradually decreases to 0 near the leading edge and the trailing edge. The definition of the inclination is shown in Fig. 2(b).

2.2. Numerical methods

The numerical simulation in this study uses the software ANSYS CFX to solve steady viscous Reynolds Averaged Navier-Stokes equations with a time pursuing finite volume method. The Shear Stress Transport (SST) turbulence model is employed for turbulence closure. This turbulence model has been verified by inhouse experimental data, which demonstrates that the SST turbulence model can accurately predict the aerodynamic performance and flow details of turbines [32]. The spatial discretization uses a second order upwind scheme. The computational domain consists of one stator channel and one rotor channel as shown in Fig. 3(a). Total temperature, total pressure and inflow angle are used as the inlet boundary conditions, and static pressure as the outlet boundary condition. Both surfaces at the circumferential side of the stator and the rotor channels are set as periodic boundary conditions. All the wall faces are set as adiabatic with no slip.

The domain meshes are generated by the software Numeca Autogrid5. An H-type grid topology is used in the main channel, and an O-type grid topology is used near the blade wall and inside the tip gap. To avoid the numerical discrepancy caused by different meshing methods, the same grid topology is used for the flat tip and the squealer tip endwall zones, as shown in Fig. 3(b) and (c). The wall distance of first mesh cell is set to 0.001 mm, and the calculated y+ is about 1.2.

Grid independency analysis is performed to check the impact of mesh density on numerical results and to determine the appropriate mesh number. The radial mesh number in the gap is analyzed and determined firstly. The five cases of varied gap mesh density calculated in the flat tip environment are shown in Table 2.

A comparison of leakage flow rates for different grid schemes is shown in Fig. 4. The difference in leakage flow rate is becoming smaller following an increasing mesh number. When the mesh number exceeds that of Grid4, the change of leakage flow rate is less than 0.3%. Fig. 5 shows a comparison of flow details at the middle of the gap displayed by Mach number contours. It is indicated that the flow field prediction inside the gap has no obvious change when Grid4 is used.

The spanwise distributions of the circumferentially averaged total temperature, total pressure and velocity flow angle at the rotor outlet are compared in Fig. 6. The rotor outlet is at Plane C3 in Fig. 1(a). The positions of the maximum parameter difference caused by varied mesh number are also displayed in Fig. 6. With