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Effects of Double-Side Labyrinth Seals on Aerodynamic Performance in a Transonic Shrouded Turbine Stage

Labyrinth seals on both rotor casing and blade tip as an effective method to control the leakage flowrate of the shroud and improve aerodynamic performances in a transonic turbine stage are investigated in this study. Compared to the case without the labyrinth seal structure, the cases with three different types of sealing teeth have been shown to reduce significantly the tip leakage flow by computational simulations. The double-side sealing teeth case reduces the leakage flowrate $m_{leakage}/m_{passage}$ from 3.4% to 1.3% and increases the efficiency by 1.4%, which is the maximum efficiency improvement of all cases. The sealing structures increase the loss inside the shroud while reducing the momentum mixing between shroud leakage flow and mainstream. Therefore, the circumferential distribution of leakage velocity is changed, as well as the distribution of high-loss zones at turbine outlet. Furthermore, the leakage-vortex loss, which is associated with the blockage effect of sealing structure to the tip leakage flow, gains more improvement than the passage-vortex at the rotor outlet section in double-side seal case. In addition, it has also been found that with a larger gap at tip, the double-side seal has better effects of reducing the leakage flow and improving the aerodynamic performance in the transonic turbine stage. [DOI: 10.1115/1.4045182]

Keywords: labyrinth seal, tip leakage flow, shrouded turbine, aerodynamic performance, transonic turbine stage

1 Introduction

In an axial flow turbine, the aerodynamic losses generated by the tip leakage flow account for a significant proportion of the overall turbine stage losses. Encasing the tips of the turbine blade by shrouds has been studied extensively and has boosted the proposal of many innovative sealing structures. Further researches on the flow in the tip clearance are of great value in real industrial applications on the purpose of improving the aerodynamic performance of the shrouded turbine stage.

The shroud leakage flow was investigated through theoretical analyzing in early researches. Denton [1] did some researches and proposed preliminary leakage flow loss mechanisms. In recent years, with the increasing development of related technology, a wide variety of researches on the loss mechanisms of shroud leakage flow have been carried out. Stetter et al. [2] performed a 1.5 stage axial low-speed air turbine test at the Ruhr University. Anker and Mayer [3], afterward, performed the corresponding numerical simulations of the 1.5 stage turbine and pointed out that the leakage flow not only introduced the mixing losses but also dominated the secondary flow, which could lead to a major increase of loss. In addition, the computational results also showed that at practical tip clearance heights, the leakage flow resulted in negative incidence over a significant portion of the downstream stator, and thus the flow separation was usually apt to occur there.

The shroud leakage physical mechanisms involved in the turbine rim seal structure and leakage flow injection process. For low-pressure turbines with and without flowpath endwall gaps, secondary flow details of endwall cavities, seal leakage, and loss features of each blade row were analyzed by Cherry et al. [4]. Gao et al. [5] found that the special hexagon structure of honeycomb seals was able to reduce the velocity difference between the

leakage flow and main flow. Bohn et al. [6] and Pei et al. [7] studied the effects of the axial gap between the shroud and endwall on the aerodynamics and heat transfer performances separately. Palmer et al. [8] performed computational studies to quantify the loss mechanisms in the shroud cavity flows of turbine tip. It was concluded that the tip seal leakage jet loss appeared to scale approximately linearly with the corrected mass flowrate per unit area over a variety of tip gaps.

When the leakage flow spilling out of the cavity, the mixing will occur between the leakage flow and the main flow in the passage, and the interaction will have a certain influence on losses. Gier et al. [9] and Mahle [10] investigated the interaction between the shroud leakage flow and main flow in a low pressure turbine. Both research groups found that the mixing loss between the shroud leakage flow and main flow account for the majority of the total loss, and the other losses included the clearance loss of the shroud, the loss of the attack angle, and the loss of the secondary flow [11].

Considering the complexity of the shroud geometry and the strong blending between the shroud leakage flow and main flow, it is not easy to perform computational studies with satisfactory accuracy and time-costing in shrouded turbines. Thus, multidimensional coupling computational fluid dynamics (CFD) method, as an effective approach for simulating complex flow systems, was widely applied [12–15]. El-Dosoky et al. [16] presented an analytical model that uses directly measurable flow quantities to predict the effects of leakage flow on the shrouded turbine stage performance. They concluded that an injection angle of 90 deg could increase the mixing loss by about 28% compared to that injecting parallel to the main flow. Zou et al. [17] used a shroud leakage flow model and a multidimensional coupled simulation method to predict the shroud flow of the shrouded turbine blade stage.

When taking a look at the flow in shroud cavity in depth, the flow inside the shroud cavity shows strong three-dimensional characteristics and the physical mechanism is too complicated to

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understand. [18] Unsteady flow interactions of a low pressure turbine were investigated in the numerical studies by Biester et al. [19] and Pfau et al. [20]. It was found that there was a strong unsteady interference between the annular vortex and rotor blade pressure field in the inlet cavity. Consequently, the strong unsteady interactions would cause the airflow to enter the top of the rotor blade as a negative angle of attack. In addition, the interaction effects of rotor and stator would also become more complex due to the unsteadiness of wake. The impacts of the straight and bowed vanes [21] to the subsequent rotor were investigated, and the results showed that the upstream bowed wake has a beneficial effect on casing endwall flow behavior. The clearance heights of shrouded rotor [22] on the developments of the flow field and losses in the subsequent stator were investigated, and the results indicated that the interaction between the tip leakage flow and the secondary losses of the subsequent stator have an important influence on aerodynamic performance of turbines.

In addition, the variations of rotor entry cavity geometry had a great influence on the shrouded turbine stage performance, a significant efficiency gain could be achieved by geometry optimization [23,24]. For the purpose of decreasing the aerodynamic losses related with its re-injection into the main flow, Wallis et al. [25] attempted to turn the fluid leaking over shrouded turbine rotor blade rows. They found that a large part of the increased losses from leakage flow could be attributed to the presence of axial gaps upstream and downstream of the tip shroud cavity, and the axial gaps enabled the mainstream fluid periodic ingress and egress in the shroud cavity under the influence of the rotor potential field.

In a three-stage model air turbine with low aspect ratio, the circumferential mixing was considered to be crucial for the aerodynamics due to the dominant influence of the leakage flow [26–28]. In order to control the shroud leakage loss by reducing circumferential mixing, an experimental study was undertaken to test a set of turning vanes. These vanes were placed in the shroud exit cavity and were designed to turn the rotor shroud leakage flow in the direction of the main blade passage flow. It was shown that the configuration with eight turning vanes per rotor blade pitch significantly improved the loss and flow angle distribution downstream of the stator blade rows, together with an increase in the turbine efficiency of 0.4%. In addition, for high-pressure turbine rotor blades, some other complex tip structures such as chord seal structure [29], shroud honeycomb seal [30], and shroud cavity injection [31] have been investigated to control the blade tip clearance leakage flow. The sealing structure also has an important influence on the flow of the shroud and blade tip. A test rig for double-sided labyrinth seals, which simulated the rim seal of a gas turbine rotor disk, was setup and aerodynamic performances of two different configurations (straight and stepped) were tested by Kim et al. [32]. The study of the leakage performance of various stepped labyrinth seals by Kang et al. [33].

In this study, to provide a better understanding of the physical mechanism of the flow structure and leakage injection process inside the shroud cavity, four configurations with or without labyrinth seal arrangements were applied and the effects of the labyrinth seal structure on the shrouded turbine stage performance were investigated. The studied turbine stage in this study is a transonic single-stage turbine from the Institute of Thermal Turbomachinery and Machine Dynamics (TTM) [34]. The tip clearance height, as an important parameter that influences the shroud leakage flowrate, was discussed carefully. The investigations on the inner three-dimensional flow field structures and the loss mechanism of the shrouded turbine tip leakage flow could be meaningful to improve the aerodynamic performance of turbine stage.

2 Numerical Method and Validation

The studied transonic single-stage turbine model has been released by TTM, and the geometry and configurations of the model are shown in Fig. 1. The stack profiles of the rotor and the

Fig. 1 TTM transonic turbine stage [34]: (a) Meridional flow path and (b) blade profiles

stator are given in Fig. 1(b). The total-to-static expansion ratio of the turbine stage is 3.12, and the other key geometrical parameters of the TTM stage are listed in Table 1.

The CFD methods have been proven to be able to effectively simulate the flow characteristics of realistic applications when analyzing the aerodynamic performance of the tip leakage flow in a turbine stage [3,9,26–28]. The 3D steady Reynolds-averaged Navier–Stokes equations with finite volume elements are applied in this code. Moreover, the turbulence kinetic energy and turbulence kinetic energy dissipation rate equations are set to the high-resolution scheme. The baseline turbulence model is employed for equation closure and the transitional turbulence is gamma theta model. All the surfaces at the circumferential sides of the stator and rotor channels are set as rotational periodicity boundary conditions. All the walls boundary conditions are set to adiabatic

Table 1 Geometrical parameters and flow conditions of TTM stage

Parameter	Symbol	Value	Unit
Stator number	N_s	24	
Rotor number	N_r	36	
Stator aspect ratio	A_s	0.78	
Rotor aspect ratio	A_r	1.35	
Tip clearance height	τ	1	mm
Expansion ratio (total to static)	π	3.12	
Inlet total temperature	T_0^*	454.4	K
Exit Re (based on rotor exit)	Re	1.6×10^6

Fig. 2 Geometric schematic of the labyrinth seals: (a) sealing cases and (b) geometric parameters of case 2

with no slip. The type of fluid is gas with a molar mass of 28.96 kg/kmol., a specific heat capacity of 1004.4 J/(kg K), and the dynamic viscosity based on the Sutherlands formula.

In the present study, four tip configurations with or without labyrinth seal arrangements are considered to analyze the flow structure and physical mechanism. Figure 2 shows the geometry structures of the four labyrinth seals and gives the definitions of some key parameters. The axial and radial clearances between the rotor blade and the casing endwall are 4τ and τ , respectively. In all of these four labyrinth seals, case 1 is chosen as the baseline, and the case is just a conventional shrouded turbine without any labyrinth seal structure. As shown in Fig. 2, case 2, case 3, and case 4 represent separately the shrouded turbines with double-side labyrinth seal structure, single-side (casing side) labyrinth seal structure, and single-side (blade tip side) labyrinth seal structure.

The whole grid domain is mainly built with H-type structured mesh, and the blade is surrounded by O-type grids to improve the solution accuracy near the blade. Besides, the grids between the labyrinth seal and the main channel are matched. And the axial grids of the main channel and the labyrinth are refined at the entry and exit of the cavity, as shown in Fig. 3. The distance of the first grid layer to the solid wall is fixed at 0.001 mm. The averaged y^+ on all the walls is about 0.77 and the maximum y^+ on the walls is about 3.01.

In order to save the cost of calculation while guaranteeing the precision of solution, a grid independency analysis is conducted to eliminate the impact of the number of grids. When keeping the same stator grid settings, six different rotor grid configurations without labyrinth seals are shown in Table 2, and a comparison of stage powers for different grid schemes is shown in Fig. 4. The stage grid numbers increase from 2.86×10^6 to 9.17×10^6 , and the variation of stage power decreases with the increase of grid numbers. When the grid number exceeds 6.97×10^6 , the change of the stage power with the grid number is less than 0.02%.

Furthermore, Fig. 5 shows the comparison of Mach number contours in the middle part of the rotor blade tip clearance. With increasing mesh number, the change of Mach numbers at the local position of rotor blade tip clearance for different grids decreases

Fig. 3 Computational domain and mesh of case 2

Table 2 Settings of different grids

	Axial $\times$ tangential $\times$ radial		
	Stator	Rotor	Total number/ $\times 10^6$
Grid 1	$167 \times 68 \times 93$	$155 \times 85 \times 117$	2.86
Grid 2	$167 \times 68 \times 93$	$163 \times 97 \times 129$	3.43
Grid 3	$167 \times 68 \times 93$	$187 \times 125 \times 153$	5.15
Grid 4	$167 \times 68 \times 93$	$199 \times 129 \times 165$	6.01
Grid 5	$167 \times 68 \times 93$	$211 \times 137 \times 177$	6.97
Grid 6	$167 \times 68 \times 93$	$227 \times 153 \times 201$	9.17

gradually. Specifically, the difference of Mach numbers for grids 5 and 6 is seen to be small, and therefore the grid setting of grid 5 is chosen to perform the numerical investigation. In addition, the labyrinth seal has the same circumferential grid layout as the main channel, and both the radial and circumferential grids are refined. Overall, the total grid number of the labyrinth seal and the shrouded turbine stage are about 8×10^6 and 15.2×10^6 , separately.

Fig. 4 Comparison of stage powers for different grid settings

The inlet total pressure and the total temperature are set to 400,000 Pa and 454.4 K, respectively. The outlet static pressure is 100,000 Pa and the inlet turbulence intensity is set to 5%. The advection scheme and turbulence numerics option are selected to “high-resolution.” For validating the numerical model, the numerical results are compared to the test data in Erhard’s paper [34]. The comparison results of stage power, inlet mass flowrate, and outlet mass-flow-averaged temperature under different inlet and outlet expansion ratios are presented as shown in Fig. 6. In the condition of lower expansion ratio, the numerical results agree well with the experimental data. In the condition of the largest expansion ratio, however, the computational turbine stage powers are slightly different from the experimental data. It can be seen that the numerical results could reflect the aerodynamic features with the same of the experimental results. Therefore, the numerical method is appropriate and feasible in the present study.

3 Results and Discussion

3.1 Comparison Between the Cases With and Without Labyrinth Seal. For a turbine stage blade, in general, the blade tip leakage flow loss is one of the most important loss sources. Different tip seal forms could lead to different tip leakage flows. In this paper, four configurations with or without labyrinth seal arrangements were investigated for analyzing the effects of labyrinth seal on the shrouded turbine stage performance. Figure 7 shows the variations of stage efficiency η (defined by Eq. (1)) and the leakage flowrate distributions under different labyrinth seal cases

Fig. 6 Comparison between experimental and numerical results: (a) stage power, (b) inlet mass flowrate, and (c) mass-flow-averaged temperature at stage outlet

$$\eta = \frac{P}{m \cdot C_p \cdot T_0^* \cdot \left[1 - \left(\frac{p_2}{p_0^*} \right)^{\frac{\gamma}{\gamma-1}} \right]} \quad (1)$$

Fig. 5 Contours of Mach number in the middle of tip gap under different grids

Fig. 7 Variations of stage efficiency η and leakage mass flowrate distribution $m_{\text{leakage}}/m_{\text{passage}}$ for different labyrinth seal cases: (a) stage efficiency distribution and (b) relative leakage flowrate distribution

As shown in Eq. (1), where P is the power of the turbine stage, m is the mass flowrate at the stator inlet, C_p is the constant pressure specific heat capacity, C_v is the constant volume specific heat capacity, T_0^* is the stator inlet total temperature, p_2 is the mass flow average static pressure at rotor outlet, p_0^* is the stator inlet total pressure, and γ is ratio of C_p to C_v .

As shown in Fig. 7(a), the η of case 2 and case 3 are improved observably compared to that of case 1 without labyrinth seal, and the η of case 4 is almost the same as that of case 1. Figure 7(b) shows the relative leakage flowrates $m_{\text{leakage}}/m_{\text{passage}}$ of the four schemes. It is found that the leakage flow of case 2 is significantly inhibited, and the leakage mass flowrate is reduced by about 60% compared with case 1. Case 2 has the highest efficiency and the least leakage flowrate in all four cases. Although the leakage flowrate of case 4 has a slight decrease, the stage efficiency has not been improved with the reduction of the leakage flowrate. Therefore, the labyrinth seal structure is considered to have complex effects on tip leakage flow, and thus influence the turbine stage efficiency.

In order to analyze in detail the change of the leakage mass flowrate along the pitch direction, Fig. 8 provides the circumferential distributions of axial average leakage flowrates at the cavity inlet for different labyrinth seal cases. The leakage flowrate during the period of one pitch is defined by the following equation:

$$m_F = \frac{\int \rho V_n dA}{\int dA} \quad (2)$$

where ρ is the density of the cavity flow, V_n is the normal velocity of cavity inlet, and A is the area of cavity inlet. The circumferential location of the rotor blade is shown in Fig. 8, and the leading edge of the rotor is at the circumferential position of 0.45 from the marked "0" point. For the baseline without the labyrinth seal structure, the inlet leakage flowrates at all circumference position are positive, and the maximum m_F appears at the circumferential position of 0.55. For the cases with the labyrinth seal structure, however, there are some locations that the inlet leakage flowrates

Fig. 8 Circumferential distribution of leakage mass flowrate at the cavity inlet for different labyrinth seal cases (one period)

are always negative. In the region of the circumferential position from 0.2 to 0.55 as shown in Fig. 8, for case 3 and case 4, the changing trends of m_F have a notable difference compared with case 1 and case 2, and present remarkable fluctuations in opposite directions. When using the double-side labyrinth seal structure as case 2, the changing trend of m_F is similar with that for the baseline case 1, and both the cases have not the large fluctuations of leakage flowrate. The single-side labyrinth seal structure, such as case 3 and case 4, can reduce the leakage flowrate, but increase the additional perturbations inside the tip shroud cavities. It could be related to the insufficient control efficiency for the single-side labyrinth seal structure. As the best control solution in all the labyrinth seal cases, the double-side labyrinth seal structure of case 2 makes the leakage flowrate has more reduction than the other single-side labyrinth seal structure at nearly all the circumferential positions. Specifically, the reduction of the leakage flowrate near the leading edge of the rotor is significantly less than those far away from the leading edge of the rotor.

The leakage flow characteristics inside the tip gap the labyrinth seal will significantly influence the second flow loss at the rotor tip, even the stage performance. The inlet condition of the tip shroud cavity is one of the most critical influences on the tip leakage flow, and the different inlet conditions resulted from the four labyrinth seal configurations will present various flow fields in the inner tip shroud cavity. For a more detailed description of the flow field, the contours of mass flowrates at different tip axial positions and the surface streamlines on two axial cross sections (plane 1 and plane 2) with the first half of blade tip gap are shown in Fig. 9. The circumference location of plane 1 and plane 2 is shown in Fig. 8. As found in the figure, for the four labyrinth seal configurations, there are similar large vortex structures and high mass flowrates near plane 2 at the entrance of the shroud cavity. The higher flowrates lead to more tip clearance flow and more complicated secondary flow at the entrance of the shroud cavity. However, there are obvious differences for the flow fields at the entrance of the shroud cavity near plane 2, and the lower flowrates and relatively simple flow field are presented. It indicates that there are pronounced circumferential nonuniformity of flow fields and aerodynamic parameters at the tip gap inlet.

For case 2, the circumferential nonuniformity still exists from the tip gap inlet to the first sealing tooth after the leakage flow passing through the cavity inlet, and is the most obvious among the four cases. Besides, at the top of the first sealing tooth, a significant backflow is formed near plane 2, and the emerging of the recirculation zone can effectively reduce the flow-through area of clearance flow and then hinder the leakage flow from passing through the clearance. However, in the areas near plane 1, there are only small backflow regions and thus low local leakage flow. For case 1, case 3, and case 4, specially, there are no such

Fig. 9 The mass flowrate contours at different tip axial positions and the surface streamlines on two axial cross sections at the first half of blade tip gap

backflow regions as presented for case 2. The lower local leakage flowrate can be seen as an indicator of the local clearance flow. In addition, inside the following sealing teeth, the distribution of tip leakage flowrate for case 2 becomes gradually more uniform than that inside the first sealing tooth. Small backflow regions can be found at the tip of each sealing tooth and will also limit the local leakage flow as mentioned before. Because the number of sealing

teeth for case 2 is twice than those for case 3 and case 4, the staggered arrangement of the sealing teeth can further increase the resistance of the local clearance flow and can effectively reduce the leakage flowrate.

Similar with the nonuniform flow characteristics as shown in Fig. 9, the leakage flow inside the rear sealing tooth still remain quite an extent of flow difference. The resulting complicated

Fig. 10 The mass flowrate contours at different tip axial positions and the surface streamlines on two axial cross sections at the second half of blade tip gap

interaction between the leakage flow and the mainstream flow will influence the aerodynamic performance at the outlet of the shroud cavity. Figure 10 shows the mass flowrate contours at different tip axial positions and the surface streamlines on two axial cross sections at the second half of blade tip gap. The areas far from the

blade, as shown in Fig. 10, represent the outflow regions of the shroud cavity, and the areas near the blade represent the inflow regions of the shroud cavity. Without the labyrinth seal structure for case 1, large amounts of fluid flows into the shroud cavity and mixes with the fluid inside the shroud cavity. Subsequently, the

Fig. 10 (Continued)

part of fluid spills out of the shroud cavity and further mixes with the mainstream flow. The maximum inflow mass flowrate reaches to about $60 \text{ kg}/(\text{s} \cdot \text{m}^2)$, and for the fluid spilled out from the shroud cavity, the maximum inflow mass flowrate is about $230 \text{ kg}/(\text{s} \cdot \text{m}^2)$. With the single-side labyrinth seal structures for case 3 and case 4, the distribution forms of the mass flowrates at the cavity outlet are similar with that for the baseline case 1. However, the local mass flowrates for case 3 and case 4 are obviously lower than that for case 1. It indicates that the mixture and interaction between the leakage flow and main flow at the tip cavity outlet have a significant reduction by applying the single-side labyrinth seal in the shrouded turbine. Moreover, the lowest mass flowrate at the cavity outlet in all the cases is found by using the double-side labyrinth seal structure, and the local lower peak negative and peak positive mass flowrates for case 2 implies smaller flow mixture losses at the cavity outlet than the other cases.

The flow losses resulted from the labyrinth seal may account for a significant portion of the overall losses of the shrouded turbine. Figure 11 gives the variation of the mass-averaged total pressure loss coefficient at the plane 3 section under different cases. The total pressure loss coefficient ($\bar{C}_{pt}$) is given by the following equation:

$$\bar{C}_{pt} = \frac{P_1^* - P_{\text{plane3}}^*}{P_1^*} \quad (3)$$

where P_1^* is the mass-averaged total pressure in the relative frame at the rotor inlet and P_{plane3}^* is the mass-averaged total pressure in the relative frame at the plane 3 section, as shown in Fig. 10. In the absence of sealing teeth for case 1, the $\bar{C}_{pt}$ is only about 0.29.

Fig. 11 The variation of the mass flow average total pressure loss coefficient ($\bar{C}_{pt}$) of the plane 3 section under different labyrinth seal cases

(a)

(b)

Fig. 12 The distribution of the normal and tangential momentum differences during one pitch along the circumferential direction at the cavity outlets of different labyrinth seals: (a) normal momentum difference and (b) tangential momentum difference

(a)

(b)

(c)

Fig. 13 The contours and spanwise pitch-averaged distributions of total pressure loss coefficient at the rotor outlet of different labyrinth seals (PV, passage-vortex high-loss region; LV, leakage-vortex high-loss region)

After adding the sealing teeth on the casing (case 3) and at the top of the blade (case 4), the $\overline{C_{pt}}$ increases to about 0.37 and 0.44, respectively, and the $\overline{C_{pt}}$ of case 2 is between 0.37 and 0.44. Although case 2 uses the double-side labyrinth sealing teeth, the more complicated tip structure for case 2 than case 4 does not lead to the higher $\overline{C_{pt}}$.

The leakage flow from the outlet of the cavity and its mixing with the mainstream flow has an important influence on the downstream flow field. In order to analyze the momentum difference between the leakage flow during one pitch and the mainstream flow as the estimation of the mixing loss, the momentum difference during one pitch is defined by the following equation:

$$\Delta(m_F V) = \frac{\int \rho V_n V_{dif} dA}{\int dA} \quad (4)$$

where V_{dif} is the velocity difference between the outlet section of the cavity and the mainstream flow field. Figure 12 shows the distribution of the normal and tangential momentum difference during one pitch along the circumferential direction at the cavity outlets of different labyrinth seal cases. The normal and tangential directions are the radial and circumferential direction,

Fig. 14 Variation of turbine stage efficiency and leakage flowrate with tip clearance size under different labyrinth seal cases: (a) stage efficiency distribution and (b) leakage flowrate distribution

Fig. 15 Mach number contours in the shroud cavity at plane 2 under different clearance sizes for cases 2 and 4

respectively. It can be observed from Fig. 12 that the normal and tangential momentum differences between the leakage flow from the cavity outlet and the mainstream flow for any labyrinth seal can be reduced and are all positive values. For the labyrinth seals of case 3 and case 4, the reductions of the normal and tangential momentum differences are lower than that for case 2. Specially, case 2 has an advantage in reducing the tangential momentum difference compared to the other two cases with labyrinth seals. The weaker mixture and interaction of the leakage flow and the mainstream flow will benefit the decrease of the tip clearance flow losses. In addition, it is worth noting that the tangential momentum difference for case 4 is even higher than that for the baseline at the 0.2 circumferential position. It could lead to the stronger interaction between the leakage flow and the mainstream flow and thus results into the increase of overall flow losses.

The leakage flow through the labyrinth seal will cause the flow loss due to the sealing teeth and mixing loss from the interaction with the mainstream fluid at the outlet of the cavity. The overall losses of the rotor are closely related with the two critical losses mentioned above. For further discussion, the contours of the total pressure loss coefficient contours at rotor outlet of different labyrinth seal cases are presented in Fig. 13(a). As shown in the figure, there are two remarkable high-loss regions at the outlet cross section corresponding to the passage-vortex (PV) and leakage-vortex (LV) high-loss regions, respectively. As the name indicates, the two high-loss regions are resulted from the passage-vortex and leakage-vortex, respectively, and the distribution of the pitch-averaged total pressure loss coefficient along the span is shown in

Fig. 13(b). The flow losses at the lower span for different cases seem like a tiny difference; however there existed large differences of loss at the upper span. By using the labyrinth seals, the positions of the PV high-loss regions move toward the midspan, and the positions of the LV high-loss regions move from the suction side to the pressure side. For case 2, the peak value and range of the LV high-loss region are the lowest for all the studied schemes. As shown in Fig. 13(b), the loss reduction in the spanwise region from 0.8 to 1.0 is probably the most important factor of decreasing the overall loss. Figure 13(c) is the C_{pt} contours with spanwise region from 0.8 to 1.0. For case 4, the total pressure loss coefficient of spanwise region from 0.8 to 1.0 is the largest, reaching 16.62%. For case 2, the total pressure loss coefficient of spanwise region from 0.8 to 1.0 is the smallest, reaching 13.34%.

3.2 Effect of the Clearance Size on the Tip Leakage Flow.

It is considered that the labyrinth seal structure has a different effect on the aerodynamic performance under different tip clearance sizes. Based on the study of the gap height of 1 mm, the additional two clearance sizes (0.6 mm and 1.4 mm) are also researched for analyzing the effect of labyrinth seal structure in the shrouded turbine with different tip clearance sizes. Figure 14 shows the variations of turbine stage efficiency and leakage flowrate with tip clearance size for different labyrinth seal cases. As shown in Fig. 14(a), case 2 with the double-side labyrinth seal has the highest turbine stage efficiency in all the cases, and the greater improvement of the stage efficiency is achieved with the larger

clearance size. For the clearance size of 1.4 mm, the maximum increase of the stage efficiency for case 2 is about 1.5% compared to case 1. For the clearance size of 0.6 mm, the influence of the labyrinth seal on the stage efficiency becomes small, and only the stage efficiency of case 2 is better than that of case 1. The studying results of the stage efficiency have to be related with the leakage flowrate; however, it is important to note that the lower leakage flowrate does not necessarily lead to the improvement of the stage efficiency in the shrouded turbine. For instance, as shown in Fig. 14(b), the leakage flowrates for all cases are reduced under the three clearance sizes, but not all of the stage efficiencies can be improved. In addition, the reductions of the leakage flowrates under different tip clearance sizes for case 2 are most obvious in the shrouded turbine with the double-side labyrinth seal. For the clearance size of 1.4 mm, the maximum reduction of the leakage flowrate for case 2 is about 64% compared to case 1.

As discussed previously, the improvement effects of the double-side and single-side labyrinth seal on the turbine stage performance at different clearance conditions, especially case 2 and case 4. For further analyzing the cause of the performance improvement under different tip gap sizes, Fig. 15 shows the Mach number contours in the shroud cavity at plane 2 under different clearance sizes for case 2 and case 4. As the clearance size increases, for case 2, the flow velocity in the shroud cavity has a slow increase due to the serpentine flow channel from the double-side labyrinth seal. However, for case 4, the flow velocity in the shroud cavity has a large increase and the effect of the single-side labyrinth seal on controlling the clearance flow is degraded significantly. A higher velocity of tip clearance flow may indicate a larger flowrate of clearance flow, which could lead to a decrease in turbine stage performance.

4 Conclusions

In this paper, a steady numerical study was carried out to evaluate the effect of the labyrinth seal on improving the aerodynamic performance of a transonic turbine stage. Some key factors affecting the performance including the labyrinth seal structures and tip clearance sizes were performed for obtaining the further understanding of the mechanism of labyrinth seal for controlling tip leakage flow. The reliability of a commercial CFD code was validated and the computed results showed a good agreement with experimental data from the literature. The following conclusions can be drawn on the basis of results in the present study.

First, the labyrinth seal turned out to be an effective method to control the leakage flowrate of the shroud and to improve the aerodynamic performances in the transonic turbine stage. The cases under three different types of sealing teeth all reduced significantly the tip leakage flow compared with the case without the labyrinth seal structure. The double-sided sealing teeth could reduce the tip leakage flowrate and increase the efficiency of the turbine stage more effectively than these cases adding sealing teeth on the casing or on the top of blade. With the gap size to be 1 mm, the turbine stage efficiency was increased by 0.72% and the $m_{\text{leakage}}/m_{\text{passage}}$ was reduced from 3.02% to 1.07%.

Second, although the sealing structures increased the flow loss inside the shroud, they reduced the momentum mixing of the shroud leakage flow with the mainstream at the exit section of the cavity. Therefore, the circumferential distribution of leakage velocity in the cavity was changed, and then the distribution of high-loss zones at the turbine outlet was also altered. With the labyrinth seal, the positions of the PV and LV high-loss regions had an obvious change compared with the baseline without the flap tip. Furthermore, the PV losses for the cases with the labyrinth seal were increased, and the corresponding LV losses were decreased. The double-side seal improved the leakage-vortex loss more effectively than the passage-vortex loss at the rotor outlet section, which was associated with the blockage effect of tip leakage flow caused by the sealing structure.

Finally, the double-side labyrinth seal achieved the highest turbine stage efficiency in all the cases, and the larger clearance size, the greater improvement of the stage efficiency. When the gap was 1.4 mm, the application of the double-side sealing teeth obtained the maximum turbine efficiency improvement of 1.4%, and the relative leakage flowrate $m_{\text{leakage}}/m_{\text{passage}}$ was reduced from 3.4% to 1.3%. In addition, as the clearance size increased, the lower leakage flowrate in the shroud cavity due to the serpentine flow channel from the double-side labyrinth seal might be an important factor of reducing the tip clearance loss.

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Nomenclature

A	= aspect ratio
LV	= leakage-vortex
m	= mass flow
N	= blade number
p	= static pressure
P^*	= total pressure
PV	= passage-vortex
R	= region
T	= temperature
V	= velocity
y^+	= dimensionless wall distance
η	= stage efficiency
π	= expansion ratio
ρ	= density
τ	= tip clearance height

Subscripts

dif	= difference
F	= per unit area
n	= normal
pt	= total pressure loss
r	= rotor
s	= stator
t	= tangential
0	= stator inlet
1	= rotor inlet
2	= rotor outlet

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