Tip-leakage flow loss reduction in a two-stage turbine using axisymmetric-casing contouring

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Abstract In order to reduce the losses caused by tip-leakage flow, axisymmetric contouring is applied to the casing of a two-stage unshrouded high pressure turbine (HPT) of aero-engine in this paper. This investigation focuses on the effects of contoured axisymmetric-casing on the blade tip-leakage flow. While the size of tip clearance remains the same as the original design, the rotor casing and the blade tip are obtained with the same contoured arc shape. Numerical calculation results show that a promotion of 0.14% to the overall efficiency is achieved. Detailed analysis indicates that it reduces the entropy generation rate caused by the complex vortex structure in the rotor tip region, especially in the tip-leakage vortex. The low velocity region in the leading edge (LE) part of the tip gap is enlarged and the pressure side/tip junction separation bubble extends much further away from the leading edge in the clearance. So the blocking effect of pressure side/tip junction separation bubble on clearance flow prevents more flow on the tip pressure side from leaking to the suction side, which results in weaker leakage vortex and less associated losses.

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1. Introduction

In the design art of modern gas turbine engine, highly loaded airfoils are more and more widely employed as they result in lower cost and engine weight by reducing the number of blades required at each stage. However, increasing the loading could lead to further challenge to aerodynamic performance and one of them is the tip-leakage flow losses in casing region associated with rotor blade, especially unshrouded turbine blade. The tip-leakage flow contributes negatively to the turbine performance. Previous research has pointed out that tip-leakage flow is one of the dominant loss sources and Denton1 indicates that it accounts for approximately 1/3 of the total aerodynamic loss.

Because of its detrimental impact on the efficiency and heat transfer, the nature of a tip-leakage flow/vortex has been investigated by many researchers using both experimental and numerical approaches. In their investigations (Moore and Tilton,2 Bindon,3 You et al.4), much understanding about the detailed flow structure in tip-leakage flow/vortex has been gained. Denton1 has done a review on the loss mechanism of
tip-leakage flow in turbomachinery. And the tip-leakage vortex is known to interact strongly with other vortices present in the tip region, such as passage vortex.\(^5\) The interaction between the leakage vortex and the tip-side passage vortex gets stronger with the increase of flow turning angle and tip clearance.

Based on such knowledge a variety of passive and active flow control approaches have been tested as possible solutions to the tip-leakage problem in the published literature. Key and Arts\(^6\) conducted a comparison of tip-leakage flow for flat tip and squealer tip geometries at high-speed conditions in a linear turbine cascade. Krishnababu et al.\(^1\) also found a cavity tip is advantageous from both the aerodynamics and the heat transfer perspectives by providing a decrease in the amount of leakage, losses and average heat transfer to the tip. Nho et al.\(^8\) investigated the effects of different blade tip shapes on total pressure loss of a linear turbine cascade experimentally. Van Ness II et al.\(^9\) implemented active flow control using a blade-tip-mounted unsteady plasma actuator in a low pressure linear turbine cascade.

Axisymmetric endwall contouring is a type of passive control technique, and has become a promising approach in the reduction of endwall loss.\(^10\)–\(^12\) The axisymmetric endwall contouring reasonably could influence the local flow, and the vortex generation and development are inhibited by the acceleration throughout the cascade. Hence the aerodynamic loss is decreased. Unlike the blade tip shaping, little erodibility to the contoured endwall shape may be caused by the hot gas. So the question arisen here is whether such an idea could still work when implemented to the control of tip-leakage flow in turbine. Bohn et al.\(^13\) did an axisymmetric-casing contouring to a four-stage turbine. It is a relative large off-set arc that stretches into the casing with the same arc shape. Then a numerical investigation is carried out to study the entropy generation rate in the tip region of the contoured and uncontoured casing turbine. And detailed analysis is done to understand the leakage flow in the tip gap and the mechanism by which the contoured casing influences this flow and the associated losses.

### 2. Model and computational method

#### 2.1. Turbine in study

A two-stage unshrouded high pressure turbine of aero-engine is selected as the platform to investigate the influence of axisymmetric endwall contouring on blade tip-leakage flow and the associated loss. This turbine is a new design plan by our group according to the data of the two-stage high pressure turbine in NASA/GE Energy Efficient Engine (GE-E\(^3\)).\(^16\)–\(^17\) In this new design plan, the flow works with higher aerodynamic efficiency but fewer blades in each stage. So the loading in each blade of the turbine is higher. Details about the design are not the topic discussed herein and some of the main design parameters of this two-stage turbine are illustrated in Table 1. In Table 1, \(R_1\) and \(R_2\) are the rotor of first stage and second stage.

Table 1 Design parameters of the two-stage high pressure turbine.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design rotating speed (r/min)</td>
<td>12630</td>
<td>Inlet total pressure (kPa)</td>
<td>1258.4</td>
</tr>
<tr>
<td>Design mass flow rate at the inlet (kg/s)</td>
<td>27.07</td>
<td>Inlet total temperature (K)</td>
<td>1615.15</td>
</tr>
<tr>
<td>Design mass flow rate at the outlet (kg/s)</td>
<td>31.7</td>
<td>Inlet flow angle ((^\circ))</td>
<td>0.0</td>
</tr>
<tr>
<td>Total pressure ratio</td>
<td>4.80</td>
<td>Outlet Mach number</td>
<td>0.429</td>
</tr>
<tr>
<td>Mass flow rate of cooling air (kg/s)</td>
<td>4.63</td>
<td>Blade tip clearance of (R_1) and (R_2) (%)</td>
<td>1, 0.6</td>
</tr>
</tbody>
</table>

Fig. 1 Grid topology of \(S_1\).
2.2. Numerical method and boundary condition

The grid is generated by CFX TurboGrid and the topology of the grid is based on O-grid and H-grid as shown in Fig. 1. In Fig. 1, S₁ is the stator of first stage. And the computational domain and mesh for this two-stage turbine are shown in Fig. 2. In Fig. 2, S₂ is the stator of second stage. The non-dimensional distance $y^+$ of the first node next to the wall surface yields a value of about 5 at the blade and the endwall (both hub and casing included). The elements for each blade passage is about 0.9 million, while 81 layers are set along the spanwise (the nodes in tip clearance of rotor is not included). In order to make sure that the computational result is independent of the grid, the mesh sensitivity is checked with different grid number for each blade row. Fig. 3(a) gives the spanwise distribution of relative total pressure $p_t$ at 50% $c_x$ plane downstream of $R_1$. In Fig. 3(a), $c_x$ is axial chord. Comparison shows little difference between the 81 layers in the span with 46 constant layers (81_46 layers) and the 120 layers. In this study, since the tip-leakage flow is the most concerning phenomenon, enough grid layers should be set in tip clearance so as to ensure that the tip clearance flow can be simulated accurately. Fig. 3(b) and (c) present the spanwise distribution of relative velocity at the middle of tip clearance of first and second stage rotor for different number of grid layers. When 20 layers are set in tip clearance, it captures the same relative velocity distribution as 40 layers at three different axial positions. In the study of Zhao et al.,18 20 layers are set for 2 mm tip clearance height. So it can be concluded that it is enough to give 20 layers in tip clearance for the present study.

The three-dimensional simulation is carried out by commercial computational fluid dynamics (CFD) solver ANSYS CFX. It solves three-dimensional Reynolds average Navier–Stokes (RANS) equations under generalized coordinates by finite volume method for space discretization. High resolution second-order central difference scheme for space is used to discretize the equations.

At the inlet, a total pressure and total temperature boundary condition are specified whose value is presented in Table 1, and the incoming flow is along axial direction. At the exit, average static pressure with value 231.06 kPa is specified. No slip adiabatic wall condition is imposed on all wall surfaces, and the casing of rotor passage is set to be static under the absolute coordinate system. A rotational periodic boundary is applied in pitch-wise direction for each passage. For the rotor/stator interface, stage model is used. It is a mixing-plane method and performs a circumferential averaging of the fluxes through bands on the interface at the upstream side of interface, then these average fluxes are applied at the downstream side of interface. The averaging at the interface incurs a one-time mixing loss, and this loss is equivalent to assuming that the physical mixing supplied by the relative motion between blade rows is sufficiently large to cause any upstream velocity profile to mix out before entering the next row. But it accounts for the spanwise flow distribution and the incoming boundary layer on the endwalls. In the investigation of Krishnababu et al.7 the heat transfer coefficient and pressure coefficient in turbine’s blade tip predicted in CFX5.6 with different turbulence models are compared with their experimental data, and the comparisons indicate that shear stress transport (SST) k-$\omega$ turbulence model can give better agreement. Therefore, the SST k-$\omega$ turbulence model is selected to model the viscous effect of turbulence flow. The flow state of boundary layer is assumed to be fully turbulent due to the high Reynolds number $O(10^6)$. In order to overcome the requirement of highly refined near-wall grid resolution, automatic near-wall treatment is selected in CFX solver. The automatic wall function allows a consistent $y^+$ insensitive mesh refinement from coarse grids, which do not resolve the viscous sublayer, to fine grids placing mesh points inside the viscous sublayer. In the simulation, steady calculation is carried out and the rotating speed of rotor is 12630 r/min.

Fig. 2 Schematic of computational domain and mesh.

Fig. 3 Mesh sensitivity study.
In addition, considering the working fluid entering the turbine is hot gas in real engine, idea gas model is chosen, and the specific heat capacity at constant pressure is specified as:

\[ cp = \left( a_1 T_f + a_2 \right) T_f + \left( a_3 T_f + a_4 \right) T_f + \left( a_5 T_f + a_6 \right) T_f + \left( a_7 T_f + a_8 \right) \]

where \( T_f \) is the temperature in the Fahrenheit scale and the values of coefficient \( a_1 - a_8 \) are as follows:

- \( a_1 = 1.0115540 \times 10^{-25} \)
- \( a_2 = -1.452677 \times 10^{-21} \)
- \( a_3 = 7.6215767 \times 10^{-18} \)
- \( a_4 = -1.5128259 \times 10^{-14} \)
- \( a_5 = -6.7178376 \times 10^{-12} \)
- \( a_6 = -6.5519486 \times 10^{-8} \)
- \( a_7 = 5.1536879 \times 10^{-5} \)
- \( a_8 = 0.25020051 \)

### 3. Axisymmetric-casing contouring

Axisymmetric endwall contouring is a kind of two-dimensional endwall contouring. It is a modification to the endwall surface along the axial direction only. This type of contouring has been widely investigated as a promising technique for the control of endwall flow. To reduce the tip-leakage flow loss in the two-stage high pressure turbine, casing contouring is introduced. By stretching the rotor tip into the axisymmetric trench and redistributing the flow near the casing boundary layer and tip gap, it can influence the vortical structure in the tip region, and proper contouring may gain benefit in the reduction of tip-leakage flow loss.

In the present study, the contoured axisymmetric casing is realized by introducing an axisymmetric arc along the axial direction instead of the original casing for both rotors’ blade tip while the relative tip clearance with respect to the rotor blade height, \( \delta_t/h \) (where \( \delta_t \) represents the size of tip clearance and \( h \) is the rotor blade height), remains the same as that of the original turbine. Hence the shape of the blade tip is a curve parallel to the contoured casing along the axial direction when viewed from the meridional direction. And the tip clearance is the same off-set arc shape. Fig. 4 presents a sketch of the casing contouring and lists the key design parameters. Here \( x_L \) and \( x_T \) respectively mean the start position and the end position of the casing arc in axial direction. The values of both are relative to the leading edge (LE) and the trailing edge (TE) of blade respectively, and it is set that the value of the position on the right-hand side of leading edge or trailing edge is positive. \( \Delta_p \) is the parameter to control the depth of the casing trenching. Table 2 gives the value of design parameters. And here circle arc is selected as the contouring curve. Since there is little change to the blade height, the tip clearance is almost the same as the original case. It should be kept in mind that in the present study, the casing contouring has a small axial length which just begins before the rotor’s leading edge and ends after this bladerow. However, the off-set arc endwall contour by Bohn et al.\(^{13}\) is large enough to contain a whole stage of turbine. So the method in the present study is much more like the casing treatment that is commonly applied to compressor’s rotors.

![Fig. 4 Sketch of the casing contouring and key design parameters.](image)

![Fig. 5 Passage geometry of two-stage high pressure turbine.](image)
Table 3  Comparison of aerodynamic performance with/without casing contouring.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Original</th>
<th>Contoured</th>
<th>Δ</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency (%)</td>
<td>92.56</td>
<td>92.70</td>
<td>+0.14</td>
</tr>
<tr>
<td>The exit mass rate (kg/s)</td>
<td>31.73</td>
<td>31.80</td>
<td>+0.07 (0.22%)</td>
</tr>
<tr>
<td>Mass-averaged exit absolute flow angle of R₁ (°)</td>
<td>5.5</td>
<td>2.7</td>
<td>−2.8</td>
</tr>
<tr>
<td>The kinetic energy across blade tip of first stage (W)</td>
<td>1546.83</td>
<td>1397.61</td>
<td>−149.22 (9.65%)</td>
</tr>
<tr>
<td>The kinetic energy across blade tip of second stage (W)</td>
<td>668.59</td>
<td>619.92</td>
<td>−48.67 (7.28%)</td>
</tr>
</tbody>
</table>

Fig. 5 presents the passage geometry of the original case (the black solid line) and the contoured case (the red dot line) of the two-stage high pressure turbine and two segments of arc are obvious on the contoured casing of rotors’ blade tip. The tip region of first rotor is illustrated in the scheme of enlarged drawing beside it. Besides, in the sketch of Fig. 5 the tip gap and the tip shape of rotor blade are not plotted.

4. Result and analysis

4.1. Performance of the turbine with/without axisymmetric-casing contouring

The performance of the high pressure turbine with axisymmetric casing contouring is presented in Table 3, in contrast with the original performance. As Table 3 shows, with the casing contouring, the mass flow rate remains almost the same as the original design, which indicates that the axisymmetric-casing contouring in the present study has little influence on the design working point of this turbine. Meanwhile, the overall isotropic efficiency of this two-stage cooled turbine achieves a promotion of 0.14%. Here the definition of aerodynamic efficiency of the cooled turbine can be referred to the second approach, stage-to-stage model, by Kurzke.19 Moreover, the mass-averaged absolute exit flow angle from R₂ decreases to 2.7° from 5.5°. It means that the flow from this high pressure turbine is closer to the axial direction, which is one of the important design requirements and provides better inflow for the downstream low pressure turbine.

The energy taken away by the leakage flow through the tip clearance would not give work to the blade as it does not turn with the passage flow. And the most important point is that higher kinetic energy would lead to stronger tip-leakage vortex which would lead to extra energy losing by viscous mixing. In this study, the total kinetic energy through the tip clearance of R₁ and R₂ is taken at the middle surface of clearance based on the relative velocity. The kinetic energy through the gap for R₁ and R₂ reduces by 9.65% and 7.28% respectively, shown in Table 3. In Table 3, Δ is the relative change aspect to the original case.

It should be noted that the contoured casing shape used in the study is not the optimized design, so it is reasonable to believe that more performance promotion can be achieved by the axisymmetric-casing contouring.

Fig. 6 presents the diagram of total pressure recovery coefficient for each blade row. Fig. 7 presents the diagram of total pressure loss coefficient.
\[ C_{pt} = \frac{p_{t,2}}{p_{t,1}} \]  

where \( p_t \) is the relative total pressure relative to the local coordinate; subscript 1 represents the plane at the inlet of each blade row and subscript 2 at the outlet.

In Fig. 6, it can be observed that the total pressure loss in the passage of \( R_1 \) is the highest and the others’ are at a close level. With the casing contouring the total pressure recovery coefficient for the stators remains almost the same as the original design, but this coefficient in the two rotors is improved. It indicates that the contouring has very little influence on the aerodynamic flow produced in the stators, but it improves the flow in rotor. Of course, it may be for the reason that the mixing-plane method is set for rotor/stator interface.

In order to assess the effect of casing contouring on the rotors’ flow behavior, the spanwise distributions of total pressure loss coefficient and the rotors’ relative exit flow angle are presented in Figs. 7 and 8. Here the total pressure loss coefficient is defined as \( Y_p = \frac{(\bar{p}_{t,1} - p_t)\bar{p}_{t,1}}{\bar{p}_{t,1}} \), where \( \bar{p}_{t,1} \) is the mass-averaged relative total pressure at rotor’s inlet. At a certain span the data in circumferential direction is mass-averaged. In Fig. 7 the total pressure loss in blade tip region is much higher than that at other span for both rotors of the turbine. There are two main loss peaks near the blade tip and the highest one is mainly induced by tip-leakage flow. In Fig. 7(a) nearly 1/3 of the rotor’s height is taken by losses caused by the tip-leakage vortex and passage vortex. It is also shown in Fig. 7 that the highest loss peaks of both rotors are reduced by nearly 0.06 when using axisymmetric-casing contouring. But the change of the second peaks for these two rotors is totally different. It is for the reason that part of the endwall incoming boundary layer is pumped away by the suction effect of leakage flow at the leading edge, but whether this effect is positive to the development of passage vortex system largely depends on the size of tip clearance and the incoming boundary layer.

In Fig. 8, the tip-leakage vortex and passage vortex in both rotors cause great under- and over-turning to the exit flow angle. Meanwhile, the influence of the contoured casing on the exit flow angle of both rotors can be clearly observed. The contoured casing reduces the radial distortion of exit flow angle associated with tip-leakage vortex and casing passage vortex and the exit flow tends to be more uniform. But the great deviation of the flow turning angle from design at 90% span is not eliminated completely though a reduction of about 15° on the deviation peak value is obtained.

## 4.2. Flow structures in tip region

The flow in tip region is three-dimensional and complex due to the existence of various vortical structures which would lead to...
significant aerodynamic loss. Fig. 9 presents the contours of axial vorticity at six different axial locations in the passage of $R_1$. In the tip region, scraping vortex, passage vortex, tip-leakage vortex, and wall vortex can be clearly observed. But it should be kept in mind that the axial vorticity is only a component of streamwise vorticity, and the difference in magnitude between two different axial positions does not simply mean the vortex becomes stronger or weaker.

Because of the scraping effect of the rotor blade on the casing boundary layer when the rotor blade moves relatively to the static casing endwall, the endwall boundary layer on the suction side accumulates and rolls up with the suction surface boundary layer into the scraping vortex. But after the approaching of horse-shoe vortex from the adjacent blade’s pressure side, most of the low momentum fluid is swept away by the passage vortex and tip-leakage vortex, and the scale of scraping vortex becomes smaller. At $x/c_s = 0.6$ the scale of tip-leakage vortex is still relatively small and attached to blade suction/endwall corner. However, beyond $x/c_s = 0.8$ its size expands rapidly until it reaches the trailing edge. Meanwhile, the passage vortex climbs up the suction surface after it reaches the suction side. Because of the skew effect of passage vortex on the boundary layer of the suction surface, the fluid in boundary layer is blown up and forms a vortex rotating in the opposite sense, termed as wall vortex. After the passage vortex lifts up from the suction surface, most of the feeding of boundary layer fluid from the endwall and the suction surface is cut off by the tip-leakage vortex and wall vortex. It can also be observed that the tip-leakage vortex at the exit plane of passage has the largest size and the strongest vorticity.

From Fig. 9(b), with casing contouring, the size tip-leakage vortex (C) becomes smaller and the intensity is weaker. Meanwhile, the extension of the scraping vortex (A) is decreased, which indicates the reduction of tip-leakage flow. However, it can also be observed that the passage vortex (B) is larger, so the corresponding loss is increased. Analysis shows that it is the trench on the casing that forms a locally divergent endwall which decorates the endwall flow, yielding stronger cross flow.
flow near the passage casing and resulting in stronger second flow.

In Fig. 10 the total pressure loss coefficient at the exit plane of the \( R_1 \) is shown. It can be seen that the radial concentrated region of high loss is the wake downstream of the blade trailing edge. In the main flow both geometries show little difference. But in the casing region, the core region of high losses is clearly smaller, which means the reduction of tip-leakage vortex intensity. Such improvement can also be observed at the second stage exit plane.

### 4.3. Entropy generation in rotors

The complex vortices in tip region are dissipated and responsible for losing energy through viscous losses in the vortex core as well as in mixing with the mainstream. Entropy is a particularly convenient measure for aerodynamic loss.\(^{1}\) So the entropy generation rate can offer a measure to detect the regions where the aerodynamic loss is originating.

Fig. 11 shows the contour of entropy generation rate per unit volume at several axial planes through \( R_1 \), from which further insight into the spatial distribution of loss sources can be gained. Here the entropy generation rate per unit volume of fluid contains two components-viscous and thermal dissipation.\(^{20,21}\) and is written as

\[
S_{\text{vol}} = \frac{1}{T} \tau_{ij} \frac{\partial u_i}{\partial x_j} + \frac{\lambda}{T} \left( \frac{\partial T}{\partial x_j} \right)^2
\]

where \( \lambda \) is the thermal conductivity, \( \tau_{ij} \) the shear stress, \( u_i \) the velocity component in \( x_j \) direction, \( T \) the static temperature. And the valued is normalized by the value of 0.5(\( \rho u_i/T \))\( /c_x \) at the exit plane of bladerow, where \( \rho \) is the density, \( u_i \) the axial velocity component.

In Fig. 11, it can be observed that most of the entropy-generating region is located in the endwall region and the blade boundary layer. Upstream of the plane \( x/c_x = 0.6 \), the generation rate in boundary layer on the suction surface is relatively high, which is the reason why the state of boundary layer is fully turbulent. Although the loss in boundary layer is still very high the loss in the tip region becomes dominant and its extent is very large beyond the axial position \( x/c_x = 0.6 \). And the passage vortex system generates only relatively mild losses.

Fig. 12 presents the entropy generation rate along the axial position through the passage of \( R_1 \) (the solid line). The blade leading edge is located at \( x/c_x = 0 \). The rate near the leading edge and the trailing edge gets a sudden jump. The loss peak is located between \( x/c_x = 0.6 \) and 0.8. Combined with Figs. 9 and 11, it can be seen that most of the leakage fluid from the first half of the rotor is jetted into the main-flow within this region and the mixing process causes the highest entropy generation rate there. Of course, in the latter half part of blade the leakage flow will continually flow across the latter half blade tip clearance and join in the leakage vortex. In Fig. 12, the distribution of entropy generation rate along axial position of \( R_1 \) in the contoured case is also presented (the dashed line). Compared with the original design, the rate in the rotor passage of the axisymmetric contouring is reduced obviously.

And Fig. 13 shows the entropy generation rate along circumferential direction at 90% span in \( R_1 \) exit plane where the center of tip-leakage vortex locates. The aerodynamic loss creation in vortex region is obviously much higher than that in main flow. And the axisymmetric-casing contouring decreases the entropy generating rate in the tip-leakage vortex and the size of tip-leakage vortex becomes smaller. Therefore, they are the direct evidences for the reduction of the scale and intensity of tip-leakage vortex.

### 4.4. Detailed analysis of tip-leakage flow

In order to gain further understanding of the mechanism of tip-leakage flow loss reduction using axisymmetric-casing contouring, more detailed insight into the flow field is achieved.

As is well known, the leakage flow is driven by the blade loading and the flow on the tip pressure side leaks to the suction side through the tip clearance. When the leakage flow enters the suction side, it mixes up with the mainstream, and increases the total pressure loss. So in Fig. 14 the blade loading distributions at 95% span of both rotors are presented by the isentropic Mach number on surface. The isentropic Mach number is based on the blade inlet relative total pressure. It can be observed that the contoured axisymmetric casing changes the rotor blade loading distribution at the blade tip. For both rotors, the peak isentropic Mach number near 70% \( c_x \) is reduced while the loading at before 35% \( c_x \) rises. The
change of pressure difference between the pressure and suction side decreases the intensity and position of tip-leakage vortex and scraping vortex though the more forward loading may increase the passage vortex. As a result, the power driven the pressure side fluid across the tip clearance is weakened.

Fig. 15 shows the streamline of leakage flow in the tip clearance, colored by the local relative velocity. And in each picture, additional figure concerning the leading edge flow structure is presented. It can be obviously seen that the leakage flow originates from two parts, the incoming near-tip flow and the upstream casing boundary layer. The skewed flow in casing boundary layer with negative angle of attack, in some way, prevents the flow from pressure side from leaking to the suction side at the leading edge region, and then it flows downstream together with the leakage flow from mainstream. In the front part of the clearance, there is a separation vortex formed and developed downstream along the blade/tip junction edge. After its formation, the suction side leg flows into the passage and soon is blown away by the high speed main flow. But the pressure side leg still rotates along the pressure side/tip junction edge, and grows by rolling in more low-momentum fluid. Behind the arc-shape separation vortex at leading edge, there is a low velocity zone (C). And it can be clearly seen that the blue area in the leading edge part becomes darker in the contoured rotor, which means the velocity there is lower than that in the original case.

Along the pressure side edge a separation bubble can be observed in the tip clearance in Fig. 15, whose schematic is shown in Fig. 16. The separation vortex is formed in this bubble by fluid from boundary layer, and it stretches downstream...
along the edge of pressure side and blade tip. In the original rotor, as the bubble flow grows in size and the gradient falls away, this bubble is blown away by the high speed leakage flow from pressure side near position (A) while it develops to the trailing edge of rotor in the gap of the contoured case. In Fig. 17, the pressure side/tip junction separation bubble structure in the gap at 76%\(c_t\), plane of \(R_1\), is presented, whose forming mechanism can be referred to the publication of Bindon.\(^3\) As the above analysis shows, the size of the bubble in the original case is no more than half of the tip clearance height while in the contoured case it takes most space of the gap. For the reason of the blocking effect by the pressure side/tip junction separation bubble\(^2\), the flow velocity in zone (B) becomes lower. So the kinetic energy of tip-leakage flow is reduced and the blocking area (D) by the leakage flow in the mainstream passage is smaller for the casing-contoured rotor. And this gives the explanation to the reduction of kinetic energy across the tip clearance in Table 3.

In Fig. 18 the contours of wall shear stress on the blade tip of \(R_1\) are presented. With casing contouring, the zone of high wall shear stress contracts but the zone of low shear stress at the front of blade profile expands obviously. The zone of low wall shear stress indicates the gas velocity near the wall is low. So the momentum in this tip clearance region is lower than that of the original design. This can also be concluded from Fig. 19. In Fig. 19, the relative velocity profile at the middle of tip clearance of \(R_1\) for this low wall shear stress zone is presented. It is clearly shown that the flow at the leading part of blade tip clearance is decelerated by the contoured arc.

Therefore, it can be inferred that it is the arc-shape contouring that achieves some influences on both the flow in the leading part of clearance and the pressure side/tip junction separation bubble and delays its ejection by the leakage jet from the pressure side of gap. By this way the contouring strengthens the blocking effect on the tip clearance and reduces the leakage flow’s intensity.

5. Conclusions

(1) According to the computational results of a two-stage high pressure turbine, the arc-shape contouring of the casing and the rotor tip gains a promotion of 0.14% to the overall efficiency. And the absolute exit flow angle from this two-stage turbine becomes closer to the axial direction, which is required by the design. Besides, the kinetic energy taken away by leakage flow becomes less. So the total pressure recovery coefficient through both rotors is improved. It can also be inferred that more promotion of efficiency can be achieved by an optimization.

(2) Compared with the original design, the region of high total pressure loss at the exit plane of the casing-contoured rotors is clearly smaller. And the associated local entropy generation rate through both rotors is reduced, especially in the tip-leakage vortex.

(3) In the contoured rotors, the loading distribution in the tip region becomes smoother and the peak value is reduced. The low velocity region in the leading edge part of the tip clearance becomes larger. For the ejection of the separation bubble at the pressure side/tip junction of the gap by the leakage jet is delayed, the separation bubble extends to the rotors trailing edge. As a result of its blocking effect, the flow velocity there becomes lower than the original design. So it is inferred that it is the contoured arc shapes of the casing and the blade tip that cause some influences on both the flow in the leading part of the tip gap and the junction separation bubble, and enlarge the area of low velocity zone and the separation bubble size. Through the blocking effect on the tip clearance flow, the axisymmetric-casing contouring prevents more flow on the tip pressure side from leaking to the suction side. As a result, the intensity of leakage vortex and the associated losses are reduced.

References


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