Effects of Tip Cavity Depth, Width, and Location on the Leakage Flow in an Axial Turbine Cascade*

Mohamed EL-GHANDOUR**, Mohammed K. IBRAHIM***, Katsunori DOI*** and Yoshiaki NAKAMURA***

** Graduate School of Engineering, Department of Aerospace Engineering, Nagoya University, Furo-cho, Chikusa-ku, Nagoya 464-8603, Japan
Email: mghandour@fluid.nuae.nagoya-u.ac.jp

*** Department of Aerospace Engineering, Nagoya University, Furo-cho, Chikusa-ku, Nagoya 464-8603, Japan

Abstract
In this study, effects of the tip cavity with various depths, widths, and locations on the leakage flow and performance of an axial turbine cascade have been investigated numerically. The blade was a linear model of the tip section of the GE-E3 high-pressure turbine first-stage rotor blade. The Delayed Detached Eddy Simulation (DDES) model was used in the simulations. The computational results showed that the leakage mass flow rate and mass-averaged total pressure loss decreased as the depth and width of the tip cavity increased. And, it was shown that the cavity near the pressure side is more effective than that near the suction side. These effects depend on a vortex generated behind the pressure side in the cavity, which is changed with the depth, width, and location. The vortex entrains the leakage flow through the clearance toward the bottom of the cavity and in the chord wise direction, which reduces the flow leaking out.

Key words: Double Squealer, Tip Clearance, Tip Cavity, Leakage Flow, Axial Flow Turbine, DDES, CFD

1. Introduction

In axial flow turbomachines, there must be a finite clearance between the rotor tip and the casing. This clearance is usually called “tip clearance.” It allows the relative motion between the blade tip and casing, prevents mechanical friction between them, and provides a suitable space for centrifugal and thermal expansions. Although the tip clearance is small, typically 1.5% of the blade span, it has a large effect on the turbine performance.

Due to pressure difference between the pressure and suction sides, the tip clearance provides a path for the leakage flow from the former to the latter. Incidentally, the tip clearance accounts for up to one-third of the total losses in a blade row.1) The leakage flow contributes negatively to the turbine performance in several ways. It does not give work to the blade as it does not turn with the passage flow. After it exits from the suction side, it interacts with the main stream. This interaction leads to additional entropy generation. Furthermore, due to the radial distribution of temperature,2,3) the thermal load on the tip is very high. This requires further cooling which imposes an additional penalty on the engine efficiency; otherwise the blade tip may burn-out or wear-out. Blade unloading, mainstream blockage and unsteadiness are also typical side
effects of the leakage flow.

Many researchers have previously studied the problem of the tip leakage flow experimentally and numerically. Sjolander\textsuperscript{4} reviewed physics of the tip-clearance flow in axial turbines. Bunker\textsuperscript{5} made a concise and informative review of turbine blade tip functional, design, and durability issues.

Control of the leakage flow by modifying the blade tip shape has been the main subject of much research. Ameri\textsuperscript{6} investigated the effect of single squealer (strip) along the camber line on the flow and heat transfer on the blade tip of a gas turbine. Heyes et al.\textsuperscript{7} studied about the single squealer along the pressure or suction sides, and they found that the squealer can provide a benefit over the flat tip. Azad et al.\textsuperscript{8} studied experimentally about single squealer as well as combinations of two squealers, and Kwak et al.\textsuperscript{9} extended their work to cover the neighboring regions. The two squealers: one along the suction side and the other one along the pressure side are commonly called “double squealer (recess tip or tip cavity).” Krishnababu et al.\textsuperscript{10} compared the performance of the double squealer to that of suction side squealer, and they showed that the double squealer reduced the leakage mass flow rate whereas the suction side squealer increased it. Prakash et al.\textsuperscript{11} proposed two blade tip shapes: a pressure side tip shelf with a vertical squealer tip wall and a pressure side tip shelf with an inclined squealer tip wall, which reduces the leakage mass flow and improves efficiency. Mischo et al.\textsuperscript{12} also proposed an improved design of the double squealer tip for a highly loaded axial turbine rotor blade. The increment in the overall efficiency was 0.2\% in experiment and 0.38\% in CFD. El-Ghandour et al.\textsuperscript{13,14} modified the double squealer shape by adding a third middle squealer along the tip camber line, which was called “triple squealer”. They showed that the triple squealer shape reduced the leakage mass flow rate more than the flat tip and the double squealer however the total pressure loss was increased. Although the tip cavity was proven to be efficient in reducing the leakage flow, the mechanism by which the tip cavity affects the leakage flow was not fully revealed.

In this paper, we study about the simple double squealer by numerical simulations to clarify the mechanisms of its effects on the leakage flow. The tip shapes is composed of pressure side squealer (PSS), suction side squealer (SSS) and a cavity between them as shown in Fig. 1. In particular, we investigate effects of the tip cavity with various depths, widths, and locations on the performance by three comparisons. Firstly, we investigate the double squealers shown in Fig. 1(b) with various depths of the cavity to study effects of the depth. The cavity depths were expressed using Depth to Span Ratio (DSR). The cavity DSR took the following values: 0.75\%, 1.5\%, 2.25\%, 3.0\% and 4.5\%. These values were 0.5, 1.0, 1.5, 2.0 and 3.0 times the tip clearance height, respectively. Secondly, we compare the performance of wide tip cavity shown in Fig. 1(b) with a thinner one (1st cavity shown in Fig. 1(c)) to study the effect of cavity width. Thirdly, we compare the performance of the 1st cavity located near the pressure side as shown in Fig. 1(c) with that of the 2nd cavity located near the suction side as shown in Fig. 1(d) to study effects of the location.

2. Nomenclature

\begin{itemize}
  \item \textbf{A}: area
  \item \textbf{ARC}: distance along airfoil surface from stagnation point to trailing edge
  \item \textbf{b}: squealer thickness
  \item \textbf{C}: blade chord
  \item \textbf{Cx}: axial chord
  \item \textbf{d1}: cavity depth
  \item \textbf{DSR}: ratio of cavity depth to span
  \item \textbf{h}: blade height
  \item \textbf{P}: Pressure
  \item \textbf{PR}: pressure ratio
\end{itemize}
Q: dynamic pressure at inlet
S: pitch
\( t \): tip clearance height
\( u^* \): friction velocity
\( U \): velocity at inlet
\( x \): axial coordinate
\( y \): distance from the wall
\( Y^+ \): dimensionless wall distance = \( u^* y / \nu \)

Greek letters
\( \bar{\Omega} \): leakage mass flow ratio
\( \bar{\xi} \): mass-averaged total pressure loss coefficient
\( \phi \): leakage mass flow rate
\( v \): laminar kinematic viscosity
\( \rho \): density
\( \xi \): total pressure loss coefficient

Subscripts
\( o \): inlet
\( 1 \): exit
\( t \): total
\( s \): static
\( x \): axial
\( l \): Local

3. Model in Study

The blade profile is the same as the tip section of NASA/GE Energy Efficient Engine (GE-E³) first-stage rotor blade because results of several studies have been reported regarding it and were used to validate the numerical code.\(^{8,9,13-17}\)

Calculations were performed for a three-times scaled-up model in the same way as El-Ghandour et al.\(^ {13,14}\) and Yang et al.\(^ {16}\). This scaled-up model has an axial chord of 86.1 mm (\( C_x = 86.1 \)), a span of 122 mm (\( h = 122 \)), and an aspect ratio (span/chord) of 1.4. The blade model is two-dimensional with the same profile in the span direction. The tip

![Fig. 1. Schematic diagram of different tip shapes used in this study.](image-url)
clearance \( t \) is constant for all computations, which is 1.5\% of the blade span \( t = 0.015h \), and the squealer thickness, \( b \), is 2.3mm.

4. Computational Method

The numerical solver used here is an in-house code. It is an unstructured, finite volume, multiblock, 3D compressible Reynolds-Averaged Navier-Stokes (RANS) solver. The inviscid numerical fluxes at the cell interface were calculated using Roe's approximate Riemann solver. To improve the spatial accuracy, the 2nd-order Monotone Upstream-centered Schemes for Conservation Laws (MUSCL) scheme with Van Albada limiter were used. The 2nd-order central differencing was applied for viscous numerical fluxes. For time integration, the Lower-upper symmetric Gauss-Seidel (LUSGS) method was employed together with the dual-time stepping which considered second order accurate\(^1\) which is sufficient for Detached Eddy Simulation (DES) as reported by Spalart.\(^1\) The code was parallelized using Message Passing Interface (MPI), and the calculations were done using Nagoya University supercomputer Fujitsu HX600.

The turbulent viscosity was calculated by the Delayed Detached Eddy Simulation (DDES)\(^2\) model which is based on the Spalart-Allmaras one equation turbulence model.\(^2\) The DDES model is a recent modification of the DES model which was proposed as a remedy for Modeled stress depletion (MSD)\(^2\). This model was selected because of its ability to simulate massive separated flow, which is the case in the leakage flow, with a reasonable computational cost.

The boundary conditions are as follows: at the inlet plane total pressure of 129.96 kPa \( (p_{in} = 129.96 \times 10^3) \), total temperature of 300 K \( (T_{in} = 300) \), and flow angle of 32 deg. were given. While at the exit plane, a static pressure of 108.3 kPa \( (p_{out} = 108.3 \times 10^3) \) was imposed and the other flow variables were extrapolated from inside domain. The no-slip and adiabatic boundary conditions were applied at the walls. The periodic boundary condition was applied in the pitch direction. The casing wall was considered stationary.

The grid was designed in such a way that all the test cases have the same base grid and obtaining certain shape is done by switching on/off some additional blocks. This minimized the grid effects on the numerical results and facilitated adopting new shapes. Figure 2(a) shows a 3-D view of the grid used in this study. The computational domain, for a single pitch of the GE-E3, is shown in Fig. 2(b). The number of blocks and grid cells vary with the cavity depth. For example, for the double squealer case with a Depth to Span Ratio (DSR) of 1.5\%, there are 1,528,374 grid points distributed in 86 blocks, which includes 165,525 nodes in the tip clearance and 95,073 nodes in the cavity. The grid numbers in the present study is, for example, more than double the grid points in our previous study\(^3\), and more than ten times the grid used by Harvey et al.\(^4\). The grid was clustered to the blade and endwall surfaces. The first cell exists at 5 \times 10^{-6} m from the wall which corresponds to dimensionless wall distance, \( Y^+ \), of 1.47.

The grid dependency study was carried out using three different grids, namely, coarse, intermediate, and fine. The number of grid points were 0.5, 1.0, and 1.5 M, respectively. Simulations were carried out for flow past a flat tip blade with a clearance of 1.5\%h. Because of its importance to the preset study, the leakage mass flow rate was estimated for the three grids and depicted in Fig. 3. It shows that the leakage mass flow rate is converged to a certain value with increasing the grid points. Therefore, we selected the fine grid. This would be enough to capture details of the flow field.

The MarkII guide vane cascade was studied experimentally by Hylton et al.\(^5\) It was chosen for validating the solver. Figure 4 shows the calculated surface pressure distribution against the experimental data of Hylton et al.\(^5\) The vertical axis is the ratio of the local static pressure to the inlet total pressure. The present computational result shows agreement
with the experimental results.

![3-D grid view](image1)

![Computational domain](image2)

**Fig. 2.** Grid and computational domain

![Grid dependency study](image3)

**Fig. 3.** Grid dependency study: leakage mass flow rate for coarse, intermediate and fine grids.

![Surface pressure distribution](image4)

**Fig. 4.** Surface pressure distribution.
5. Results

5.1 Effect of cavity depth

Figure 5 shows streamlines of the leakage flow (colored by Mach number) for double squealer with different depths of the cavity. The streamline shown in this figure passes through each referential point which is very close to the edge of the pressure side. It shows that the flow pattern is marked by two horseshoe vortices on the tip, HS1 and HS2. HS1 is formed on the top of the squealer while HS2 is inside the cavity. The right-side arm of HS1, HS1-RA, moves along the pressure side squealer from the leading edge to the trailing edge. The left-side arm of HS1, HS1-LA, proceeds along the suction side squealer till it leaves the tip around point A where it sheared with the mainstream forming the leakage vortex. On the other hand, the right-side arm of HS2, HS2-RA, moves nearly in the axial direction and hits the suction side squealer around point B, and then a part of it exits the tip while the remaining part moves close to the suction side squealer toward the trailing edge. The left-side arm of HS2, HS2-LA, moves adjacent to the inner wall of the suction side squealer before crossing it around point C and finally got entrained around the leakage vortex. In order to compare the flow patterns, the number of the streamlines and their referential points were kept fixed in all cases. Figure 5 indicates that the dimension of HS2-RA increases with the depth of the cavity.

In the following, the total pressure loss coefficient, $\xi$, is used, which is defined by

$$\xi = \frac{P_{to} - P_{tl}}{Q}, \quad Q = \frac{1}{2} \rho U^2$$

where $P_{to}$ is the inlet total pressure, $P_{tl}$ the local total pressure, $\rho$ the density and $U$ the velocity at inlet.

Figure 6 shows the projection of velocity vectors on the plane p1 shown in Fig. 2(b), superimposed on the contours of $\xi$, for double squealer with different DSR. It shows that the flow entering the tip clearance, adjacent to the tip surface, separates at the edge of the pressure side forming a recirculation zone and vena contracta. This recirculation zone corresponds to HS1-RA in Fig. 5. Then it generates a vortex behind the pressure side in the cavity. This vortex corresponds to the HS2-RA in Fig. 5. The leakage jet is formed over the separated flow and is marked by low loss as shown in Fig. 6. The generated vortex in the cavity tends to entrain the neighboring flow around its core. Therefore, the leakage jet is diverted toward the bottom of the cavity and in the chord wise direction, and then the entrained flow is partially blocked by the suction side squealer. As a result, the vortex in the cavity tends to reduce the flow leaking out. In the comparison of various depths, it is shown that the dimension of the vortex in the cavity increases with the depth. Therefore, the leakage flow decreases as the depth of the cavity increases as shown in Fig. 9.

Figure 7 shows the static pressure distributions on the plane p1 shown in Fig. 2(b) for double squealer with different DSR. The flow structure shown in Figs. 5 and 6 are manifested in the static pressure distributions. HS1-RA is shown as low pressure zone over the pressure side squealer and the leakage vortex is shown as a low pressure zone beside the outer wall of suction side squealer. There is also a high pressure zone at the inner side of the suction squealer which corresponds to the impingement of the leakage jet on the inner side of the suction side squealer and on the cavity floor. The pressure at that zone in the cavity with DSR of 1.5% is higher than that of 4.5% because the leakage jet is mainly blocked by the suction side squealer with DSR of 1.5% whereas it is diverted smoothly by the larger vortex with DSR of 4.5%.
Fig. 5. Streamlines of the leakage flow (colored by Mach number) for double squealer with different DSR.
Figure 8 shows the projection of velocity vectors on plane p2 shown in Fig. 2(b), superimposed on the contours of $\xi$, for double squealer with different DSR. It shows that the leakage flow, emerging from the tip, generates a vortex behind the suction side squealer. This vortex is called the leakage vortex. And then, this vortex is usually accompanied with an increase in the total pressure loss. In the comparison of various depths, it indicates that the dimension of the leakage vortex decreases as the depth increases which is a result of the reduction in the leakage flow. Therefore, the total pressure loss decreases as the depth of the cavity increases as shown in Fig. 10.

The performances of the various depths are evaluated by two indicators, the leakage mass flow ratio, $\bar{\alpha}$, and the mass-averaged total pressure loss coefficient, $\bar{\omega}$.

The leakage mass flow ratio is the ratio of the leakage mass flow rate of the current result to that in the case of the flat tip, and the leakage mass flow is estimated by integrating the mass flux over the area between the tip and casing along the suction side. In Fig. 9, the leakage mass flow ratio is plotted for double squealer with different DSR. It is shown that the leakage mass flow ratio decreases as the depth of the cavity increases with a non-linear reduction rate. For example, 4.5% of DSR decrease about 18% of the leakage mass flow.
The mass-averaged total pressure loss coefficient, $\bar{\xi}$, is defined as:

$$\bar{\xi} = \frac{\int \xi \rho V \delta A}{\int \rho V \delta A}$$

(2)

where $A$ is the observation area, and $V$ the local velocity component normal to the area. The mass-averaged total pressure loss coefficient on the plane $p3$ shown in Fig. 2(b) is depicted for double squealer with different DSR in Fig. 10. The $p3$ is 20% chord length of distant from the trailing edge. It is shown that the mass-averaged total pressure loss coefficient decreases as the depth of the cavity increases. For example, 4.5% of DSR decreases about
10% of the total pressure loss. Losses due to leakage flow could be classified, based on the location relative to the tip clearance, into two types: internal and external. The internal losses (losses take place inside the tip clearance), which are increased with DSR due to the increase in HS2-RA size as shown in Fig. 5; whereas the external losses (losses take place outside the tip clearance) are decreased due to the decrease in the leakage vortex size as shown in Fig. 7. The very small reduction rate between DSR 1.5 and 2.25% is referred to the balance between the increase in the internal losses and the decrease in the external losses.

![Fig. 8. The projection velocity vectors on p2, superimposed on the contours of ξ, for double squealer with different DSR.](image-url)
5.2 Effects of cavity width and location

Figure 11 shows the streamlines of the leakage flow (colored by Mach number) for the 1st and 2nd cavities as shown in Fig. 1(c) and Fig. 1(d). The DSRs of the cavities are 1.5%. In the 1st cavity, there are one horseshoe vortex, HS1, and another vortex instead of HS2. The new vortex is labeled “cavity vortex” in Fig. 11(a). This vortex has the same direction as HS2-RA but it is weaker than it. On the other hand, in the 2nd cavity, the horseshoe vortices, HS1 and HS2, are found. HS1 is similar to that in the double squealer and 1st cavity, while HS2 is weaker than that in the double squealer.

Figure 12 shows the projection of velocity vectors on the plane p1 shown in Fig. 2(b), superimposed on the contours of $\xi$, for the 1st and 2nd cavities.

For the 1st cavity, it shows that the flow entering the tip clearance, adjacent to the tip surface, separates at the edge of the pressure side forming a recirculation zone and vena contracta. Then it generates a vortex behind the pressure side in the cavity. This is similar to that of the double squealer. However, the dimension of vortex is smaller than that of the double squealer due to the limited cavity width and the cavity is filled with the vortex. As a result, the entrainment of the flow through the clearance by the vortex in the cavity is weaker than that of the double squealer. And, the leakage flow and the pressure loss are higher than those of the double squealer as shown in Fig. 14.
For the 2nd cavity, it shows that the leakage flow separates at the edge of the pressure side, but it reattaches on the pressure side squealer because the distance between the tip entrance and the cavity entrance is relatively long. It also shows that the vortex generated in the cavity is very weak, and then the entrainment of the flow through the clearance by the vortex in the cavity is also very weak. Therefore, the leakage flow and the pressure loss are higher than those of the double squealer and the 1st cavity as shown in Fig. 14.

Figure 13 shows the static pressure distributions on p1 for the 1st and 2nd cavities. It shows that the cavity vortex formed in the 1st cavity is larger and stronger than HS2-RA in the 2nd cavity. Therefore, the pressure at the inner side of the suction squealer of the 1st cavity is lower than that of 2nd cavity because it is diverted smoothly by the larger vortex in the 1st cavity whereas the leakage jet is mainly blocked by the suction side squealer in the 2nd cavity.

![Fig. 11. Streamlines of the leakage flow (colored by Mach number) for the 1st and 2nd cavities.](image-url)
Fig. 12. The projection of velocity vectors on p1, superimposed on the contours of $\xi$, for the 1st and 2nd cavities.

Fig. 13. The static pressure distribution on p1 for the 1st and 2nd cavities.
Figure 14 depicts the leakage mass flow ratio and the mass-averaged total pressure loss coefficient at the plane p3 for the flat tip, double squealer with 1.5% of DSR, and 1st and 2nd cavity. It shows that the double squealer is the best shape among all shapes tested in this paper. This effect includes the effect that the cavity is wider and the effect that the pressure side squealer is thinner.

![Leakage mass flow ratio and Mass-averaged total pressure loss coefficient](image)

Fig. 14. $\phi$ and $\xi$ (at plane p3) for the 1st and 2nd cavities and double squealer.

6. Conclusion

The present study investigated numerically the effects of the tip cavity with various depths, widths, and locations on the performance of an axial turbine blade and clarified their mechanisms. Results obtained in this study are summarized as follows:

1. The leakage flow rate decreased as the depth of the tip cavity increased because the vortex behind the pressure side in the cavity grows with the depth and entrains the flow through the clearance toward the bottom of the cavity and in the chord wise direction.

2. The leakage flow rate in the case of the wide cavity was smaller than that of the narrow cavity because the growth of the vortex in the cavity was limited by the side wall of the narrow cavity.

3. The leakage flow rate in the case of the cavity near the pressure side was smaller than that near the suction side because the wide squealer along the pressure side suppresses the vortex formation in the cavity due to the viscous effect.

4. The total pressure loss decreased with the leakage flow rate because the leakage vortex grows with the leakage flow.

Consequently, the vortex in the cavity plays the vital role for these effects. In other words, the effects of various tip cavities can be explained based on the vortex in the cavity.

These results can provide useful information on how to design efficient tip shape that is able to reduce the loss which has environmental and economical impacts by reducing the specific fuel consumption.
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References

(14) El-Ghandour, M., Ibrahim, M. K., Mori, K. and Nakamura, Y.: A Triple Squealer


