NUMERICAL STUDY OF THE TIP CLEARANCE FLOW DEVELOPMENT
IN A PROPULSION PUMP STAGE

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ABSTRACT
This paper summarizes a numerical investigation of the fundamental structure of the rotor tip-clearance vortex and its interaction with a passage trailing-edge vortex in a single-stage stator-rotor pump. The flow field of a highly-loaded rotor measured in a high Reynolds number pump facility (HIREP) is used for comparison. The numerical solution was obtained by solving the three-dimensional Reynolds averaged Navier-Stokes equations. The calculated results are visualized in order to understand the details of the tip-vortex structure. The study shows that the tip geometry should be accurately represented to predict the tip-vortex structure correctly.

NOMENCLATURE

Cn amplitude of Fourier decomposition
Cp pressure coefficient normalized by dynamic pressure based on rotor tip speed
IGV inlet guide vane
k turbulent kinetic energy
PS pressure side
R radius
SS suction side
Vx axial velocity
VT tangential velocity
V velocity
x axial distance
ε turbulent dissipation

Subscripts
tip rotor tip

INTRODUCTION
High performance rotors and narrow design margins for future turbomachinery demand accurate flow analysis methods and design products. With the advance of computational power, numerical methods which solve the Reynolds averaged Navier Stokes equations not only provide detailed flow analyses, but become an important element of turbomachinery design. In this paper, we use a Navier-Stokes solver to analyze a rotor clearance flow for a single-stage stator-rotor pump.

Rotor tip-clearance flows have been experimentally investigated by Betz [1], Meldahl [2], Rains [3], Hunter and Cumpsty [4], Yalas and Sjolander [5], Schuab et al. [6], Zierke et al. [7]. On the other hand, computational methods (Hah [8], Dawes [9], Crook [10], Adamczyk et al. [11], Copenhaver et al. [12], Yang [13], Lee et al. [14], Chen et al. [15]) have also been used successfully to interpret many tip-clearance flows. Because of these continuous efforts, many tip-clearance flow features have been revealed. Most are primarily related to either the formation of the tip-clearance vortex or its transport downstream and interaction with other components. Many questions, however, still remain to be answered.

The present study investigates the physical process of tip-clearance vortex generation and the transport of the vortex in a highly loaded, high Reynolds number axial rotor. Three-dimensional Navier Stokes analyses were performed using two gridding approaches for the benchmark rotor flow field. The numerical analyses focus on studying effects of gridding the tip-clearance region on the generation and the transport of the vortical structure.
NUMERICAL METHOD

The steady Reynolds averaged Navier Stokes equations were solved to analyze the flow features. The numerical method [16] used is a finite-volume approach for incompressible flows. The governing equations are solved with a pressure-based implicit relaxation method using a fully conservative control volume approach for each blade row. A third-order accurate interpolation scheme is used for the discretization of the convection terms and central differencing is used for the diffusion terms. The turbulence closure is a two-equation k-ε model with a modification to integrate the governing equations up to the wall, following the studies of Chien [17]. The rotor inflow boundary conditions are obtained from either an upstream blade-row calculation or a uniform flow. The downstream outflow boundary conditions are extrapolated from the interior grid points. Mass conservation is enforced between the inlet and the exit planes. Periodicity conditions are enforced at a midplane between the blade suction and pressure sides.

A simple mixing plane is used to treat stage flows in a steady manner. Details of the numerical procedure for steady multiple blade rows with the mixing plane are explained by Lee et al. [14].

EXPERIMENTAL ROTOR CONFIGURATION AND TEST DATA

A comprehensive data set is required to assess the validity of rotor tip-clearance flow predictions. The pump rotor geometry chosen for this study, shown in Fig. 1, was tested in the HIREP water tunnel [7, 18]. The pump consists of a 13-blade IGV (inlet guide vane) and a 7-blade backward swept rotor. The separation at the tip between the blade rows of the IGV and the rotor is approximately 1.3 IGV chord lengths. The origin of the axial coordinate used is located at the center of the rotor chord. The axial-flow pump has a constant hub diameter of 0.5334 m and a constant casing diameter of 1.067 m. The IGV blades have a constant spanwise chord of 0.1753 m. The rotor blades have a chord length of 0.2850 m and a solidity of 1.19 at the hub, and a chord length of 0.2664 m and a solidity of 0.56 at the tip. The tip section has a thickness-to-chord ratio of 10.14%. The nominal rotor tip clearance is 0.3302 mm, which is 1.24% of the chord of the rotor tip section.

The pump operates at an inlet velocity of 10.67 m/s and an inlet pressure of 2.978 x 10^6 dynes/cm^2. The rotor rotates at 260 rpm with a flow coefficient of 1.36 and a tip speed of 14.51 m/s. The tested Reynolds number based on the chord of the IGV blade is 2.3 x 10^6. The details of the measurement hardware and results are given in Zierke et al. [7, 18].

NUMERICAL RESULTS

Numerical Representation of Tip Region

The numerical results presented in [14] were obtained using the same numerical method as above and show a correct global trend for the rotor passage flow. The trailing-edge and tip-clearance flows, however, were not predicted accurately. Figure 2 shows a comparison of fluid particle traces between the visualized and the predicted rotor suction-surface flows. The sketch, shown in Fig. 2a, from the experimental paint visualization indicates that the trailing-edge separated flow is convected radially to the rotor tip by centrifugal force. This radial trailing-edge flow was not completely predicted by the earlier calculation [14] which is shown here in Fig. 2b. Although the hub separation was predicted, the radial trailing-edge flow terminates at one-third of the rotor span. This result seems to be due to the turbulence model used for predicting the flow separation. In [14] a modified turbulence length scale, which reduces the turbulent viscosity near the wall, was investigated with limited success. The hub-to-tip radial flow was predicted reasonably. The trailing-edge flow pattern, however, contained more three-dimensional features than observed in the experiment.

In this paper, a precise gridding treatment of the tip-clearance region is employed. Figure 3 shows the different grid topologies used to obtain the previous and present solutions. The grid used for the previous prediction (called the regular grid) is shown in Fig. 3a with the same blade cross sections used in the tip clearance region as in the blade tip section. The top of the blade tip was not gridded. Instead of applying a periodic boundary condition from the tip to the casing for a zero-thickness blade profile, an approximate condition was applied at each surface of the finite-thickness blade tip section. This gridding approach was successfully used for many compressor calculations which dealt
The grid used for the present calculation (called the embedded grid) is shown in Fig. 3b. In addition to distributing the volume grid in the flow domain, the grid is extended to the blade interior. This grid not only assures a smooth grid transition from the blade tip to the tip-clearance region, but also provides a means to apply the nonslip boundary condition on the blade tip surface.

The overall features of the computational grid are shown in Figs. 1 and 3. The embedded grid consists of 54 nodes in the blade-to-blade direction, 47 nodes in the spanwise direction and 101 nodes in the streamwise direction. The actual rotor tip geometry was gridded with eight nodes from the suction surface to the pressure surface of the blade. Ten nodes were located between the rotor tip and the casing.
Rotor Flow Field at Design Condition

Figure 4 shows the predicted circumferentially averaged inflow conditions compared with the five-hole-probe data at 82.3% IGV chord upstream of the rotor tip leading edge. The comparisons indicate that the rotor inlet swirl varies parabolically from the hub to the tip and that the maximum swirl occurs near 20% of the span from the hub. The axial velocity comparison shows that the inflow boundary layer matches the measurement well.

Passage Flow on Suction Side

Figure 5 shows predicted particle traces on the rotor suction side using the embedded grid approach. The trailing-edge flow resembles the experimental observation and shows a two-dimensional structure except near the hub and the casing. A comparison of the chordwise pressure distributions at 30% and 70% of the span on both sides of the blade is shown in Fig. 6. The agree-

Figure 4. Pump rotor inlet flow comparison.

Figure 5. Predicted flow on the pump rotor suction surface using embedded grid calculation.

Figure 6. Pump rotor blade surface pressure comparison.
ment between the prediction and the measurement is good except on the suction side at 30% of the span. This discrepancy is related to a fillet at the rotor hub between the endwall and the blade surfaces which was neglected in the computational model.

A dramatic difference between the predicted results using the regular grid and the embedded grid can be seen in Fig. 7, which shows three-dimensional perspective views of both predicted particle traces on the rotor suction side and in the wake. Figure 7 also shows the tip-vortex trajectories which will be discussed in the next section. A measured plane downstream of the rotor is displayed with the predicted axial-velocity contours to trace the vortex motions produced by the upstream trailing-edge vortex and the tip-clearance vortex. The solution from the regular grid produces a large recirculating region near the hub and an incorrect large wake on the downstream plane. The solution from the embedded grid, however, shows a clear trailing-edge separation vortex driven in the radial direction by the centrifugal force.

Tip Flow

The predicted tip-clearance flows are also shown in Fig. 7. Due to the interaction of the tip vortex and the trailing-edge vortex shown in the embedded grid solution, the tip vortex travels farther tangentially as it is convected downstream than it does in the regular grid solution. The downstream plane shows that the tip vortex from the embedded grid solution stays close to the suction surface and is entirely separate from the trailing-edge vortex.

Figure 8 shows a comparison of the experimental and the predicted interactions between the tip vortex and the trailing-edge vortex. The photo of the experiment was obtained by lowering the tunnel pressure to let both vortices cavitate at the core of the vortex. Inoue and Kuroumaru [19], Storer and Cumpsty [20] and Zierke et al. [7] located the onset of the tip vortex roll-up at the point of minimum pressure on the suction surface. The photo shows only the last 30% of the blade chord. The two vor-
tices pass one another at different radii and do not intersect. Based on the sizes of the cavitated vortices, the tip-clearance vortex seems to have a larger strength than the trailing-edge vortex. The predicted results from the embedded grid indicate that the tip vortex passes over the tip section from the pressure side to the suction side at about 15% of the chord, which was also observed by Zierke et al. [7]. The calculation also correctly predicts the relative sizes and the points of crossover of both vortices.

**Wake Flow**

Figure 9a shows the measured axial-velocity contours in the rotor wake at the downstream plane shown in Fig. 7. The corresponding predicted contours using both grids are shown in Figs. 9b and 9c. Figure 10 shows the predicted secondary flows at the same rotor wake plane, which were constructed by subtracting the mean component of the plane velocity at each radius from the local absolute velocity. Since the radial velocity was not measured, Fig. 10 does not include the corresponding measured secondary-flow structure. The zigzag contours from the measurement shown in Fig. 9a are due to the sparse measuring locations in the radial direction. As was pointed out earlier with Fig. 7, the regular grid solution produces an unrealistically large hub-vortex wake. This vortex (the T.E. vortex) is also shown in Figs. 9b and 10a. On the other hand, the embedded grid solution shown in Figs. 9c and 10b has a blade wake which resembles the measured blade wake shown in Fig. 9a. A similar phenomenon can be observed for the tip-clearance vortex. For the regular grid solution shown in Fig. 9b, the high-gradient area of the axial-velocity contours is near the tip of the suction side. This suggests the existence of vortex activity. For the embedded grid solution shown in Fig. 9c, however, the high-gradient area moves tangentially (opposite to the direction of rotation) toward the pressure side. The connection between the tip and T.E. vortices shown in Fig. 10b and the upstream vortex motion was demonstrated in Fig. 7.

Figure 11 shows a quantitative comparison of the axial and the tangential velocities at the rotor midspan. The predicted velocity agrees well with the measured velocity, but the calculation predicts a deeper wake defect velocity. The coefficients of the harmonic decomposition of both velocity components are also shown in Fig. 11. Although the experimental data contains extensive noise, particularly in the high-frequency range, general agreement exists between the experiment and the prediction up to the fourth harmonics.

**CONCLUSIONS**

The current study compares numerical solutions with measured vortical structures in a high-Reynolds-number rotor flow, with particular emphasis on the rotor tip-clearance flow. Solutions obtained from using the regular and embedded grids were compared and prove that the approximate tip-clearance treatment of the regular grid approach is not adequate for the present pump rotor geometry and its tip clearance. The embedded grid solution, on the other hand, not only predicts the correct trailing-edge separation vortex associated with the centrifugal effect, but also
Figure 10. Secondary flows in the pump rotor wake. The authors are indebted to Messrs. W. C. Zierke, P. D. Taylor and W. A. Straka for providing their rotor data. The first author was funded under the Applied Hydrodynamics Program and the Block Program for Hydrodynamics and Hydroacoustics of Internal Flow. The program monitor at the Office of Naval Research is Dr. P. Purcell.

ACKNOWLEDGMENTS

The correct trend for the interaction between the trailing-edge vortex and the tip-clearance vortex. Further comparisons also indicate that the formation of the tip vortex is primarily due to the local pressure gradient across the rotor tip for the current rotor.

Figure 11. Comparisons of axial and tangential velocities in the rotor pump wake.
REFERENCES


