Numerical investigation of blade tip cavitation on large discharge condition of diagonal turbine

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Abstract. A series of numerical simulations was carried out to estimate the cavitation characteristics for a diagonal flow turbine. The interaction between the tip leakage flow and the cavitation bubbles was investigated using the obtained numerical data. The evaporation and condensation process for cavitation phenomenon was described by using a simplified Rayleigh-Plesset equation. A two-phase homogeneous model was adopted to calculate the mixture of gas and liquid phases. Three types of cavitation with different origins were obtained around the blade tip region. The incipient cavitation was well-known tip clearance cavitation. In the present study, the occurrence of tip clearance cavitation was limited in the position where the clearance between blade tip and discharge ring spread, irrespective of Thoma number. Tip clearance cavitation was found out not to contribute to the decrease of the turbine efficiency. On the contrary, the turbine efficiency increased with the growth of tip clearance cavitation. When Thoma number decreased, the other type cavitation arose on the discharge ring surface. The development of the discharge ring cavitation led to the decrease of the turbine efficiency. In addition, so-called tip vortex cavitation obtained in the lower Thoma number condition than the appearance of discharge ring cavitation. These three types of cavitation were also observed in the experimental approach. The correlation between the decrease of the turbine efficiency and the development of discharge ring cavitation was also verified in the model test.

1. Introduction
Diagonal flow turbine (diagonal turbine, hereafter) is consisted with adjustable blades and the blade angle varies to be suited for each operating point. Because of these blades, the diagonal turbine achieves high turbine efficiency not only at the design point but also at partial loads and overloads. On the overload conditions, the blades are widely opened as the large amount of water flows through the runner. Therefore, the trailing edges of blades move to downstream and stick out from the spherical range of the discharge ring. In this case, the clearance between the blade tips and the discharge ring keeps constant distance in the upstream region and spreads in the vicinities of the trailing edges. The sectional view of the diagonal turbine was shown in figure 1. The upstream region of the discharge ring shapes spherical to keep the clearance between the blade tip and the discharge ring constant. The downstream region of the discharge ring is conical. The spherical and conical parts are smoothly connected. Therefore, the inflection point exists in the end of the spherical part. The clearance in the upstream region of the inflection point is constant, and the clearance in the downstream region of the inflection point spreads as going to the downstream as shown in figure 1.

The flow in the clearance between the blade tips and the discharge ring has been investigated frequently. In those works, two types of cavitation were reported[1]. One was tip vortex
cavitation that occurred at the core of the rolled up vortex on the blade suction side[1, 2, 3]. Another was tip clearance cavitation that arose in the clearance[4]. The cavitation phenomena cause the decrease of the turbine efficiency. Furthermore, it may damage the components of the machine and shortens the service life.

To design a turbine avoiding the problems caused by the cavitation, the proper understanding about the phenomena is required. Most of the past studies considered as the clearance was almost constant from the leading edge toward the trailing edge. As mentioned above, the tip clearance of the diagonal turbine blade is not constant, but spreading around the trailing edge. This paper mainly describes the effects of the spreading blade tip clearance toward the flow patterns and the cavitation characteristics in the large discharge conditions using the numerical approaches. The numerically estimated turbine efficiency profiles and screenshots of cavitating region were compared with the experimental data on model cavitation test in order to confirm the accuracy of the Computational Fluid Dynamics (CFD) results. The relation between the cavitation phenomena and decrease of the turbine efficiency was discussed based on the obtained numerical and experimental data.

2. Numerical procedures
A diagonal turbine scaled model of specific speed \( N_{QE} = 0.15 \) is simulated in the present study. Here, the specific speed \( N_{QE} \) is based on the IEC standards, and is defined as

\[
N_{QE} = \frac{n \cdot \sqrt[3]{Q}}{E^{\frac{2}{3}}},
\]

using the rotational speed \( n [s^{-1}] \), the flow rate in the design point \( Q [m^3/s] \) and the specific hydraulic energy \( E [J/kg] \). The turbine contained 18 guide vanes and a runner. The runner consisted of 8 blades. The reference diameter of the runner was 325.8[mm].

The simulations were carried out with the use of the commercial CFD software ANSYS CFX 17.2. This software is known as a coupled solver, which solves velocity and pressure equations simultaneously. The effects of turbulence were taken into account by using the shear stress transport (SST) turbulence model developed by Menter[5]. A first and second order hybrid differential scheme peculiar to the software was adopted as the spatial discretization method.

In order to save the computational time, the steady simulations with a mixing plane approximation[6] were carried out in the one flow path around a guide vane and a runner.
blade. The subject of the present simulation was bordered by the broken line in the figure 1. To stabilize the CFD, the inlet and outlet parts were elongated. An example of computational domain and grid was shown in figure 2.

The evaporation and condensation process for cavitation phenomenon was described by using a simplified Rayleigh-Plesset equation

$$\frac{dR_b}{dt} = \sqrt{\frac{3}{2}} \frac{p_b - p}{\rho},$$

where \(R_b\), \(t\), \(p_b\), \(p\) and \(\rho\) were the radius of vapor bubble, time, instantaneous static pressure in vapor bubble, instantaneous static pressure at local point and density of liquid water, respectively. The averaged radius of vapor bubble \(R_b\) was set as \(2.0 \times 10^{-6}\) [m]. The evaporation \((F_{eva})\) and condensation coefficients \((F_{cnd})\) were equal to the typical default values in the literature [7] by Bakir et al., that is \(F_{eva} = 50\) and \(F_{cnd} = 0.01\). A two-phase homogeneous model was adopted to calculate the mixture of gas and liquid phases.

The Thoma number \(\sigma\) was varied from 0.140 to 0.465 to obtain the cavitation characteristics. The flow rate depended on the Thoma number. Therefore, the total and static pressures were specified to the inlet and outlet boundaries so as the computed turbine head was equal to that in the non-cavitation condition. The flow rate was obtained as the computational result for each Thoma number. The computational conditions were summarized in table 1. The speed factor \(Q_{ED}\) and the discharge factor \(Q_{ED}\) were defined as follows;

$$n_{ED} = \frac{n \cdot D}{\sqrt{E}}, \quad Q_{ED} = \frac{Q}{D^2 \cdot \sqrt{E}},$$

The computations were judged as converged when the root mean squares (RMSs) of residuals reached to the value lower than \(1.0 \times 10^{-6}\). In the cases with the higher RMSs of residuals than \(1.0 \times 10^{-6}\), the following two criteria were used to confirm whether the simulations converged. One was that the level of the RMSs of residuals were fell down around \(1.0 \times 10^{-4}\). Another was
Table 1. Computational conditions.

| Flow rate | Small | Large |
|-----------|-------|-------|
| Speed factor $n_{ED}$ | 0.453 | 0.453 |
| Discharge factor $Q_{ED}$ | 0.183 | 0.214 |
| Guide vane opening (GVO) [%] | 70.0 | 90.0 |
| Runner blade angle (BA) [deg] | 25.0 | 25.0 |
| Thoma number $\sigma$ | $0.140 - 0.390$ | $0.195 - 0.465$ |

that the integrated variables such as flow rate, total or static pressure at boundary surfaces were oscillating around fixed values during not less than 1000 iterations. The turbine characteristics were derived using the fixed values.

3. Results and discussion

3.1. Cavitation characteristics

The relation between Thoma number and turbine efficiency for small and large flow rate conditions are plotted in figures 3 and 4, respectively.

In the present study, the computational domain was quite different from the model test rigs in not containing several components such as the spiral casing, the stay vanes and the bent draft tube. Owing to the difference of evaluated regions, the numerically estimated turbine efficiency and Thoma number were not equivalent to the experimentally obtained ones. Therefore, the CFD and model test were compared qualitatively. The efficiency was normalized by the maximum value. The Thoma number $\sigma$ was normalized by that in just beginning of the efficiency drop condition $\sigma_{dr}$.

In the small flow rate condition, the CFD result showed good agreement with the model test within the range of Thoma number $\sigma > 0.18$. On the other hand, the points for the large flow rate condition seemed to shift to large Thoma number. However, the difference between the CFD and model test was small. In addition, the decrease rate of obtained turbine efficiency as decrease of Thoma number agreed with the model test, especially below the efficiency drop point. This result indicated that the present CFD was adequate for qualitative evaluation of the relation between the cavitation phenomena and the turbine efficiency.

The cavitating regions were visualized in figure 5 for three typical conditions, they were the high Thoma number condition(a), just beginning of the efficiency drop condition(b) and below the efficiency drop condition(c). The photographs taken in the model test were compared. In the CFD results, the line consisted of inflection points (inflection line, hereafter) was drawn.
Figure 5. Relation between the turbine efficiency and the cavitating regions under the large flow rate condition.

The envelope of the section which was vertical to the turbine axis and has minimum area size was also shown.

In the high Thoma number condition, (a) and (A), the incipient cavitation arose in the clearance between the blade tip and the discharge ring surface. This cavitation was well-known as the tip clearance cavitation[4] (TCC, hereafter). TCC grew toward the circumferential and the trailing edge directions as the decrease of the Thoma number but was not elongated toward the leading edge direction. This is because the TCC was caused by the spreading clearance. Therefore, TCC was located just after the inflection line on the discharge ring. This phenomenon will be discussed in the following section 3.2.

In the condition just beginning of the efficiency drop, (b) and (B), a unique cavitation appeared in the region between the inflection line and the envelope of the minimum section. This cavitation attached to the surface of the discharge ring. In present paper, authors called this cavitation phenomenon as Discharge Ring Cavitation (DRC, hereafter). DRC was enlarged
as the decrease of the Thoma number as shown in figure 5(c) and (C). And the turbine efficiency decreased as development of the DRC. The turbine efficiency in the small flow rate condition was also confirmed to decrease as the development of DRC (not shown in present paper). This result indicated that the cavitation characteristics for the diagonal turbine depended on the DRC. The detailed description about DRC will be discussed in the section 3.3.

While the tip vortex cavitation (TVC, hereafter) was estimated by the CFD in the low Thoma number condition, it did not get the pictures in the model test. This is because TVC was intermittent phenomenon. During the model test, the instantaneous appearance and disappearance of TVC were often verified with the visual observation. In the present work, the steady flow was assumed. Therefore, it was expected that TVC phenomenon was overestimated as the steady. This problem will be solved using the unsteady CFD. The unsteady CFD is scheduled as one of the future works in order to obtain the intermittent TVC phenomenon.

3.2. Tip clearance cavitation
For the detailed analysis, some variables were extracted from the tip clearance region. The data extraction was performed in many lines defined in the clearance region (see figure 6). The extracted data were averaged in the direction from the blade tip toward the discharge ring. The obtained averaged velocity profiles for non-cavitation and just beginning of the efficiency drop (b) conditions were plotted in figures 7 and 9. The averaged pressure profiles for non-cavitation and just beginning of the efficiency drop (b) conditions were drawn in figures 8 and 10. The dynamic and total pressures were derived in the static system. All pressure components were normalized by the total pressure difference between the inlet ($P_{t,in}$) and the outlet ($P_{t,out}$) as follows;

$$C_p = \frac{p - P_{t,out}}{P_{t,in} - P_{t,out}}.$$  (4)

Here, $p$ was the target of the normalization, $C_p$ was the pressure coefficient. The blade surface static pressure was also plotted for comparison. The blade surface static pressure was obtained in a line which internally divided the surface from the hub side edge toward the blade tip by 9:1.

In the non-cavitation condition, all velocity components increased just before the position where the blade tip clearance started to spread. The velocity magnitude had maximum value in the beginning of the clearance spread (BCS position, hereafter). This is because the spread of the clearance caused the reduction of friction loss. The loss due to sudden expansion was also decreased in the suction side of the blade.

The velocity magnitude decreased just after the BCS position (see the chordwise position 0.8 - 1.0). The radial and axial direction components were mainly decreased. The reason why the velocity magnitude decreased, the pressure difference between the pressure and suction sides of blades got smaller as going toward the trailing edge (as shown in figure 8). In addition, the
spreading clearance accelerated the mixture of fluid. And the accelerated mixture uniformed above mentioned static pressures on the pressure and suction sides of blades. The total pressure profile in non-cavitation condition also pointed out the spreading clearance caused the uniform pressure profile (see chordwise position 0.85 - 1.0 in figure 8).

On the other hands, the dynamic pressure increased around the BCS position. Therefore, the static pressure drastically fell down around the BCS position. This was the main cause of the Tip Clearance Cavitation (TCC). Hence TCC was related to the BCS position strongly, the upstream edge of the TCC was fixed as described in section 3.1. The comparison of figures 8 and 10 proved that the significant decrease of the static pressure was independent of Thoma number and took place in the same position, just before the BCS position. The minimum static pressure obtained in the chordwise position 0.75 - 0.92 in figure 10 was corresponding to the saturation vapor pressure.

The peak value of the velocity magnitude in the operating condition (b) was lower than that of the non-cavitation condition (figure 9). The resistance produced by the cavitation bubbles was assumed to decelerate the fluid in the clearance.

3.3. Discharge ring cavitation

In order to investigate the Discharge Ring Cavitation (DRC) in detail, the vortex structures around the DRC were visualized in figure 11. The visualization was performed to the just beginning of the efficiency drop condition ($\sigma / \sigma_{dr} = 1.0$). The gray isosurface was corresponding to the region where the second invariant of velocity gradient tensor $Q/Q_{max} > 0.00112$. Here, $Q$ was normalized by the maximum value of $Q$ in whole computational domain. The variable $Q$ was widely used for the vortex identification[8]. The light blue isosurface was the region where the volume fraction of vapor $\gamma_v > 0.5$ and equivalent to the caviting region.
Figure 11. Vortex structure around the discharge ring cavitation under large flow rate condition.

Figure 12. Helicity contour crossing the blade tip and secondary vortices under large flow rate condition.

Figure 13. Definition of lines crossing the blade tip vortex.

Figure 14. Monitoring lines projected onto the meridional plane.

Figure 11 indicated that the blade tip vortex was detached from the discharge ring and DRC occurred in the interstice between the blade tip vortex and the discharge ring. In addition, another vortex structure was appeared in the downstream region of the blade tip vortex. Harvey & Perry[9] confirmed that the vortex approaching the wall rebounded in their experiments. According to them, the boundary layer was developed as the approach of the vortex to the wall. And the secondary vortex appeared in the boundary layer. The secondary vortex prevented the primary vortex to progress. As the results, the primary vortex rebounded from the wall. In the present case, the above described another vortex was the equivalent of the secondary vortex. The contour of helicity was shown in figure 12. Helicity has two meanings; those are the strength and the direction of the spiral motion. Warm colors were corresponding to the strong counter-clockwise spiral motions and cold colors were the strong clockwise ones from the downstream point of view. The blade tip vortex was founded to be rotating toward counter-clockwise direction and the secondary vortex toward clockwise direction as drawn in figure 12. Two vortices combined and high speed region turned out in the middle region between those vortices.

The secondary vortex reached to the inflection line before the primary vortex. Owing to the destabilization effect of the curvature of the discharge ring, the secondary vortex was strengthened. The static pressure in the core region of the secondary vortex was decreased by the strengthening. This was the reason why the DRC occurred in the region between the blade tip vortex and the secondary vortex. The large magnitude velocity generated by the
composition of two vortices also one of the reason why the static pressure decreased and the DRC appeared.

To confirm the mechanism for occurring DRC, some variables were obtained in the lines crossing the blade tip vortex. The lines in which variables were extracted were visualized in figures 13 and 14. These lines were across the blade tip vortex and the axis of the rotation as shown in figure 13. The angles formed by the lines and the horizontal plane were set as 45 degrees (see figure 14).

The axial velocity components for non-cavitation and (b) conditions were compared in figures 15 and 17. Here, the local maxima and local minima were corresponding to the up-streamwise and down-streamwise velocity fluctuations generated by the blade tip vortex and secondary vortex. The lines in upstream region (lines 1 to 4) had only one local maximum. And the minimum value of the axial velocity obtained near the end of the curves. The ends of the curves meant the discharge ring surface. These results suggested that the secondary vortex was thin and touched the discharge ring surface in the upstream region. However, the lines in downstream region (lines 5 to 7) had two local maxima. This is because the counter rotating motion caused by the secondary vortex developed. The line 7 in figure 15 indicated that the local minimum around the radial position 0.4 corresponding to the negative direction velocity fluctuation was enhanced compared with the positive fluctuation. This negative direction velocity fluctuation was the large magnitude velocity generated by overlapping of the blade tip vortex and the secondary vortex, mentioned in figure 12. Line 7 in figure 17 pointed out that the secondary vortex was more strengthened under the cavitating conditions compared with the non-cavitation condition. The local maximum in the end of the curve corresponding to the positive fluctuation of the secondary vortex was higher than the local maximum caused by the blade tip vortex at
radial position 0.39.

The static pressure profiles were plotted in figures 16 and 18. The minimum values of the pressure were obtained at the right end of the curves for the upstream regions (see lines 1 to 4 in figure 16). These results meant that the decrease of the pressure was occurred near the secondary vortex. As the growth of the secondary vortex, the minimum values of the pressure sited toward inner direction. Some lines of the static pressure in figure 18 had a constant value. This value was equal to the saturation vapor pressure. The comparison of figures 16 and 18 again indicated that the vortex structure was strengthened under the cavitating condition and the minimum values of the pressure were lower than those of the non-cavitation condition while the values were almost same at the location 0.34.

4. Conclusion

A series of numerical simulations for a diagonal turbine was carried out to estimate the relation between the blade tip leakage flow and the cavitation phenomena. To confirm the accuracy of the simulations, model tests for the corresponding operating points were also performed. The derived conclusions were as follows;

• Three types of cavitation phenomena were observed around the blade tip clearance.
• The incipient cavitation was the well-known tip clearance cavitation.
• The spread of the clearance led the decreases of the losses by the friction and the sudden expansion at suction side of the blade. Because of the decreasing losses, the dynamic pressure increased and the static pressure decreased. The tip clearance cavitation appeared owing to the decrease of the static pressure.
• The tip clearance cavitation did not affect to the decrease of the turbine efficiency.
• The decrease of the turbine efficiency was caused by the cavitation occurred on the surface of the discharge ring.
• The discharge ring cavitation was strongly related to the rebound phenomenon of the blade tip vortex. The main sources of the discharge ring cavitation were the strengthened secondary vortex by the curvature of the discharge ring and accelerated velocity by the composition of the blade tip and the secondary vortices.

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