Influence of runner clearance on efficiency and cavitation in Kaplan turbine

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Abstract. Since the clearance between the runner shroud and the hub can change with the angle of the blades, it has a great influence on the hydraulic characteristics and cavitation performance in the Kaplan Turbine. In this paper, the CFD will be used to solve the turbulence model based on the k-ωSST model in the entire flow field of the turbine and the Zwart-Gerber-Belamri model will be used to predict the cavitation. We compare the hydraulic characteristics and cavitation performance of the Kaplan Turbine under the conditions of the runner clearance exist or the runner clearance absence. Finally, we analysis the influence of the clearance width in the hydraulic performance and the relationship between the clearances and the turbine characteristics.

1. Preface

In the Kaplan Turbine, the match relationship between the blade and the guide vane causes the turbine have a wide range of high efficient operating areas, thus the Kaplan Turbines are widely used in medium head and low head hydropower stations. The angle of the blade can vary with the operating conditions. In the blade angle change process, the runner hub clearance also changed but the runner shroud clearance remain unchanged. Wu Hua¹ using CFD to study the effect of hub clearance flow on Kaplan Turbine’s internal flow structure, it prove the hub clearance lower the hydraulic efficiency but reduce the hydraulic loss in draft tube. Xiao Wei [²] made an overall and systematic numerical calculation of the inner flow behavior of Kaplan Turbine including the rim clearance in the case of different blade angles and each guiding vane openings, proving the basis for the further improving of this type turbine as well as the stable operation of the hydropower station. Tomas [³] considering the influence of the runner blades and the hub clearance flow in the optimized design, optimizing the blades to improve the efficiency of the runner. Liao WL [⁴] made numerical simulation and experimental research on the characteristics of the clearance cavitation in Kaplan Turbine, mainly analysis the relationship in between the leaking vortex, secondary flow vortex cavitation and erosion.

In this paper, we use numerical simulation method to calculate the hydraulic characteristics and cavitation performance for an actual operation Kaplan Turbine in the power plant, considering no clearance of the runner and considering the clearance of the runner in the same working condition. At the same time, we also change the width of the clearance in different simulation groups in order to study the influence of different clearance width on the performance of the turbine.
2. Geometric model and simulation method

The geometrical parameters and operating parameters of the Kaplan Turbine used in this paper are shown in Table 1, the runner diameter is 5000 mm and the rated rotating speed is 107.1 rpm.

| NO. | Name                  | unit | parameter |
|-----|-----------------------|------|-----------|
| 1   | Runner diameter D     | mm   | 5000      |
| 2   | Hub diameter D\textsubscript{hub} | mm   | 2140      |
| 3   | Rotating speed n      | r/min| 107.1     |
| 4   | Blade number Z\textsubscript{i} | /    | 5         |
| 5   | Stay vane number Z\textsubscript{c} | /    | 11+1      |
| 6   | Guide vane number     | /    | 24        |
| 7   | Draft tube middle pier | /    | 1         |

The turbine is divided into volute section, stay vane section and guide vane section, runner section and draft tube section. We compute all these sections in the simulation, the fluid domain is shown as figure 1:

\[ \frac{\partial}{\partial t}(\rho \mathbf{u}) = 0 \quad (2.1) \]

\[ \frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} \left[ (\rho u_i) \right] = \frac{\partial}{\partial x_j} \left[ (\mu + \mu_s) \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + f_i \quad (2.2) \]

Since this study calculates the flow in the rotating runner, the average N-S equation in the rotating coordinate system is as follows:
\[
\frac{\partial}{\partial x_i} (w) = 0
\] 
\[\frac{\partial w_i}{\partial t} + \frac{\partial}{\partial x_j} (w_j w_i) = - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \frac{\mu + \mu_t}{\rho} \left( \frac{\partial w_i}{\partial x_j} + \frac{\partial w_j}{\partial x_i} \right) \right] + f'_i
\] 
\[\dot{u} = \dot{w} + \dot{\omega} \times \dot{r}
\] 
\[F' = -2\dot{\omega} \times \dot{r} - \dot{\omega} \times (\dot{r} \times \dot{r}) + f
\]

As the calculation is non-compressible flow field, SIMPLEC is used to solve the velocity field and the pressure field in the solution of RANS.

The boundary conditions are as table 2:

| Table 2. Boundary conditions |
|-------------------------------|
| Inlet: Total pressure, 136927 Pa |
| Outlet: Relative pressure, 0 Pa |
| Boundary conditions: Solid wall with no slip boundary conditions |
| Fixed interface: General interface |
| Static and dynamic interface: Stage interface |
| Turbulence model: SST k- |
| Cavitation model: Zwart-Gerber-Belamri model |
| Convergence residual: 10e-5 |
| Gravity: 9.81 m/s^2 |

After grid-independent check, we decide to use the following grid number: the volute section 500000, the guide vane section 4800000, the runner section 1300000 and draft tube section 1000000. When considering the runner clearance, we need more meticulous grid to reflect the clearance so the runner section grid number is 3000000. Details of the grid is show on figure 2:

3. No clearance/clearance exist result contrast

Calculate conditions of Head = 12.9m, guide vane opening 49.4 degrees, blade opening 29.3 degrees, rim clearance 3mm, hub clearance corresponding to the maximum opening of the blade (34 degrees) is 2mm. Compare the former calculation results with results that on the same operating conditions but do not take into account the clearance, as shown in the following table 3:

| Table 3. Results comparison |

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(a) runner grid
(b) shroud clearance grid circumferential direction
(c) shroud clearance grid radial direction(20 layers mesh)

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Figure 2. Details of the runner grid

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3. No clearance/clearance exist result contrast

Calculate conditions of Head = 12.9m, guide vane opening 49.4 degrees, blade opening 29.3 degrees, rim clearance 3mm, hub clearance corresponding to the maximum opening of the blade (34 degrees) is 2mm. Compare the former calculation results with results that on the same operating conditions but do not take into account the clearance, as shown in the following table 3:
According to the results, it can be seen that the efficiency and output power decrease after the consideration of the runner clearance, but the flow rate $Q$ and water vapor volume rise. The relative deviation is smaller than 1%.

The cavitation of axial flow turbine is mainly divide as vane cavitation and clearance cavitation. The vane cavitation refers to the cavitation caused by the water flow through the blade surface then local pressure decrease. The clearance cavitation is due to the water flow through some narrow channel or gap, causing the local flow rate increases, the pressure reduction caused by a whirlpool cavitation. In our computation case, refer to the installation position of the turbine to determine the suction height $H_s = -3.5m$, the height of the center of the runner to the outlet of the tail pipe is $H_d = 6.702m$. Check the properties of the water. At 25 °C, the gasification pressure $P_{va} = 3168Pa$, the local altitude 128m atmospheric pressure is $99736Pa$. Use water vapor volume fraction 10% as the cavitation surface, the distribution of cavitation cavities on a single blade is shown in the following figure 3 (clearance exist) and figure 4 (no clearance):

(a) Pressure surface cavitation cavities  (b) Suction surface cavitation cavities  (c) shroud clearance cavitation cavities  (d) hub clearance cavitation cavities

**Figure 3.** Blade cavitation cavities (clearance exist)
Figure 4. Blade cavitation cavities (no clearance)

In both cases above, cavitation appears only on suction surface. In the case which no clearance, there are two cavitation zone in the backside of the leading edge and in the middle of the connection between hub and blade.

In the case which runner clearance exist, cavitation are mainly distribute in suction surface in three positions: the leading edge near the rim position, the middle of the shroud clearance position, the middle of the hub clearance position. The cavitation in the shroud position have the largest volume, but the volume of the three cavitation cavities are not large relative to the runner, which indicates that the cavitation performance of the runner is good. The cavitation of the shroud and the hub can be classified as clearance cavitation due to the water flow through the gap, causing the local flow rate increases, the pressure reduction caused by a whirlpool cavitation. The leading edge cavitation cavity is due to the change in the flow pressure caused by the blade shape.\textsuperscript{[7][8]}

According to the calculation results of the turbine operating point, the pressure distribution of the blade is obtained, as shown in figure 5:

Figure 5. Blade pressure distribution

As can be seen from the above figure 5, the pressure on the front surface gradually decreases from the inlet side to the water outlet side. As a result of the centrifugal force, the pressure contour on the rim side is greater than the pressure at the hub side\textsuperscript{[9][10]}. The pressure in the blade along the direction of water flow first drop and then rise. In the case which runner clearance exist, the low pressure zone is located in the middle of the blade, and the lowest pressure zone is located at the inner edge and the outer edge of the middle part of the blade, there is also a small low-pressure zone at the outer edge of the water head. The three low-pressure zones are most likely to become cavitation zone, this is consistent with the cavitation cavities described above. In the case which no clearance, the middle shroud edge doesn’t have a low pressure zone, so no cavitation cavity is formed in that place.

Figure 6 shows the water velocity streamline in the runner clearance in both relative coordinate system and absolute coordinate system. The water flow along the clearance from the front side to the backside of the blade. Due to the circumferential velocity of the runner, the velocity of the clearance flow at the shroud is greater than that in the hub, and the flow direction is almost parallel to the blade shape in the shroud clearance.
Using the unsteady method to calculate the water vapor volume change during the rotation of runner for one period, result shown in figure 7:

\[ \text{water vapor volume (mL)} \]

In one rotation period (0.56s), the maximal water vapor volume is 4.04 mL, and the cavitation position remains unchanged, no other cavitation occurs in the rest of the runner, the runner anti-cavitation performance is fine.

4. Influence of the clearance width in the hydraulic performance

In this chapter, we focus on the influence of the width of the runner clearance on the hydraulic characteristics of the turbine. To reduce the interference factor, we only change one parameter value of the runner clearance at a time.

First, assuming that the shroud clearance value is zero, the influence of the runner clearance is studied by changing the width of the hub clearance. The width of the different hub clearances are listed in the following table 4. The calculation is based on the previous meshing method and the same boundary conditions. The calculated results are also shown in the following figure 8.

| Table 4. Hydraulic characteristics with different hub clearance | 3.7 | 3.75 | 3.8 | 3.85 | 3.9 | 3.95 | 4 | 4.05 |
|---|---|---|---|---|---|---|---|---|
| water vapor volume (mL) | 0.00 | 0.14 | 0.28 | 0.42 | 0.56 | 0.00 | 0.14 | 0.28 | 0.42 | 0.56 |
It can be seen from the figure, with the width of the clearance at the hub increases, the Turbine's output power and efficiency have a small decline, while the water vapor volume and flow increased. But the flow increase is quite small, in the process of clearance change from 0mm to 6mm, the maximum flow of only change 0.3%. As the clearance width increases, water flowing through the gap increases, resulting in increased flow, and the part of the water from the gap directly flow away, did not push the rotor blade work, resulting in the decline of turbine efficiency. Power is equal to the product of flow and efficiency, the rate of increase in flow is less than the rate of decline in efficiency, resulting in a decline in output power. Since the hub clearance is only increased to 6mm, it is only 0.12% relative to the diameter of the runner 5000mm, so the flow in the gap still produces a clearance cavitation and, as the flow rate increases, increases the volume of water vapor.

Using the same method, close the hub clearance, change the clearance width at the shroud to study its effect on the hydraulic characteristics of the turbine. The results are as following table 5 and figure 9:

Table 5. Hydraulic characteristics with different shroud clearance

| Shroud clearance (mm) | 0.0  | 2.0  | 2.3  | 2.6  | 3.0  | 3.4  | 3.7  | 6.0  | 9.0  |
|-----------------------|------|------|------|------|------|------|------|------|------|
| η (%)                 | 88.00| 87.54| 87.59| 87.46| 87.44| 87.42| 87.37| 86.97| 86.45|
| Q(m3/s)               | 184.40| 184.41| 184.40| 184.42| 184.42| 184.41| 184.40| 184.91| 185.54|
| Power(kW)             | 20554| 20475| 20475| 20446| 20439| 20437| 20423| 20385| 20328|
| Vapor volume(mL)      | 2.07 | 3.42 | 3.40 | 3.43 | 3.43 | 3.41 | 3.44 | 3.42 | 3.52 |
Figure 9. Parameters trends with shroud clearance change

In general, the effect of the shroud clearance on the hydraulic characteristics is consistent with that of the hub clearance. As the width increases, the efficiency and output power of the turbine decrease, and the flow and water vapor volume rise. But the volume of water vapor almost remain unchanged after the shroud clearance greater than 3mm.

5. Conclusion

In this paper, the influence of runner clearance on the hydraulic characteristics and cavitation performance of the Kaplan Turbine is studied by numerical simulation. The calculation method used in the simulation is reliable, it has been verified on other Kaplan turbine after comparison between the experimental data and simulation data. Calculation is done at the rated operating point. After considering the clearance of the runner, the efficiency of the turbine is reduced by 0.79%, the flow rate is increased by 0.25%, the power is reduced by 0.65%, and the cavitation water vapor volume is increased by 86.71%, indicating that a large part of cavitation occurs at the clearance of the runner. During the process of changing the runner clearance width to the original width of 100% to 300%, the efficiency and output power will decrease, the flow and cavitation water vapor volume will rise as the width increase.

When the shroud clearance of the runner is enlarged to about one thousandth (about 5 mm) of the diameter of the runner, the efficiency of the turbine is reduced by 0.5% relative to the normal clearance (3 mm). In a power station when the turbine is running for a long period of time, due to the wear and tear, the shroud clearance will be larger than the initial value, contributing to one reason in the turbine output decline phenomenon.

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