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Flow Characteristics Analysis of a High-Head Prototype Pump-Turbine During Turbine Start-Up: Effects of the Clearance

Flow Characteristics Analysis of a High-Head Prototype Pump-Turbine During Turbine Start-Up: Effects of the Clearance

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Abstract. The dramatic changes in the internal flow and the corresponding structural behavior during the turbine start-up transient process of pump-turbines are extremely complex. The clearances in the upper crown chamber and bottom ring chamber affect the results of the flow field and structural field of the pump-turbine runner. Most of the previous studies ignored the effects of the clearance flow field to simplify the numerical simulations. In this study, numerical calculations were performed on the entire flow passage of a high-head prototype pump-turbine during the start-up in turbine mode, and the model with and without the clearance are analysed respectively. The causes of the flow field characteristic difference and external characteristic difference caused by the existence of intermediate clearance in the model are studied in detail. The results show that the clearance flow field has a great influence on the axial forces on the runner, which is mainly due to the higher pressure of the clearance flow field compared with the flow field in the runner; At the same time, because the existence of clearance flow field only has a small effect on the distribution of internal flow field, the hydraulic torque of runner considering clearance effects is basically the same as the one without clearance.

1. Introduction

Pumped hydro energy storage is a well-established and commercially-acceptable technology for utility-scale electricity storage and has been used since as early as the 1890s[1]. The reversible pump-turbine (PT) units are the key components of pumped storage power stations (PSPS) that can operate for power generation in turbine mode and energy storage in pump mode. So the PSPSs can effectively absorb unstable wind power and solar power, and improve the stability of the power grid via frequency modulation, phase modulation, and peak shaving. However, to meet the requirements of the power grid, PTs must experience more start-stops per day, and the turbine start-up transient process is a very challenging operation condition for the unit and is a major fatigue damage factor for the turbine unit[2].

During the transient process of pump-turbines, the axial force on the runner, shaft and support bracket changes strongly and complexly. If the upward axial hydraulic force is too strong to exceed the unit weight the unit lifting happens and sometimes it can lead to serious accidents [3-5]. To explore the influencing factors of the axial hydraulic force, many studies have focused on the runner clearance in the upper crown chamber and bottom ring chamber. It is been explored that the axial hydraulic force is caused by the pressure difference between the inner and outer surfaces of the runner's crown and the band and between the runner blade's suction and pressure sides [6,7]. The clearance flow can also cause hydraulic losses by viscous dissipation effects of the vortex flows [8,9]. But there are still many issues that need to be studied in depth.

In this paper, the start-up transient process in turbine mode of a high-head prototype pump-turbine was studied by a coupled one-dimensional-three-dimensional numerical calculations method. The model with and without the clearance is analysed respectively to compare the performance characteristics and the axial hydraulic force. The influence of clearance flow during the start-up process is discussed.

2. Numerical method

2.1. Governing equations and turbulence model

During the start-up transient process, the flow in the pipeline system of the power station is simplified into the one-dimensional flow and solved by using the method of characteristics (MOC). The control equation of a one-dimensional pipeline system is

$$\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x} + g\frac{\partial H}{\partial x} + \frac{f}{2D}V|V| = 0 \tag{1}$$

$$V\frac{\partial H}{\partial x} + \frac{\partial H}{\partial t} + \frac{a^2}{g}\frac{\partial V}{\partial x} - V\sin\alpha = 0 \tag{2}$$

where V is the average velocity at cross-section (m/s), H is the pressure head (m), t is the time (s), x is the distance along the pipeline axis (m), g is the gravitational acceleration (m/s^2), f is the Darcy–Weisbach friction factor, D is the pipeline diameter (m), a is the water-hammer wave speed (m/s), α is the angle between pipeline axis and horizontal plane (degrees).

Surge tanks, pipes with branches and other components in the hydraulic system are also simplified and solved.

To study the internal flow performance and stress characteristics of the unit, the external characteristics calculated by MOC are used as the boundary conditions of three-dimensional calculation. In the three-dimensional calculation process, the flow in the unit is regarded as a three-dimensional incompressible flow. The governing equation is

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{3}$$

$$\frac{\partial u_i}{\partial t} + u_j\frac{\partial u_i}{\partial x_j} = f_i - \frac{1}{\rho}\frac{\partial p}{\partial x_i} + \nu\nabla^2 u_i \tag{4}$$

where u_i and u_j are the index form of velocity components, x_i and x_j are the cartesian coordinate components, f_i is the body force component, ρ is the density of water, p is the pressure, and ν is the kinematic viscosity. In this paper, the Shear-Stress Transport (SST) k-ω model is used to calculate the turbulence in the pump-turbine. It can simulate both the shear flow and adverse pressure gradient accurately and is particularly useful in engineering simulations.

2.2. Calculation model and boundary conditions

The one-dimensional calculation model consists of reservoirs, surge tanks, tunnels, pipelines, pump-turbine units, and valves. At the beginning of the start-up transient process, the water levels of upstream and downstream reservoirs are 867 m and 306 m respectively. The guide vane opens at a fixed angular speed until the rotating speed reaches the rated value, the movement of the guide vane is controlled by the speed controller.

The three-dimensional model of the pump-turbine of a pumped storage power station is shown in Figure 1, which mainly consists of spiral casing, stay vane, guide vane, runner and draft tube. The clearance upper crown is between the hub and the head cover and the clearance lower band is between the shroud and the bottom ring. The model with the clearance and the model without clearance are simulated respectively. The mesh of the flow field is composed of tetrahedral mesh and hexahedral mesh, which is shown in Figure 2. The mesh sensitivity was studied by comparing the relative torque results in Figure 3. The variation of these cases is less than 7.5%. As the research unit is under construction, there is a lack of comparison between the simulation and experimental results, which will be supplemented in further research. The chosen mesh has 2,258,125 nodes in the whole fluid domain and the number of elements is 5,738,537. The domain of runner has 335,724 nodes and the clearance has 689,073 nodes. The mesh details of the runner and clearance are shown in Figure 4.

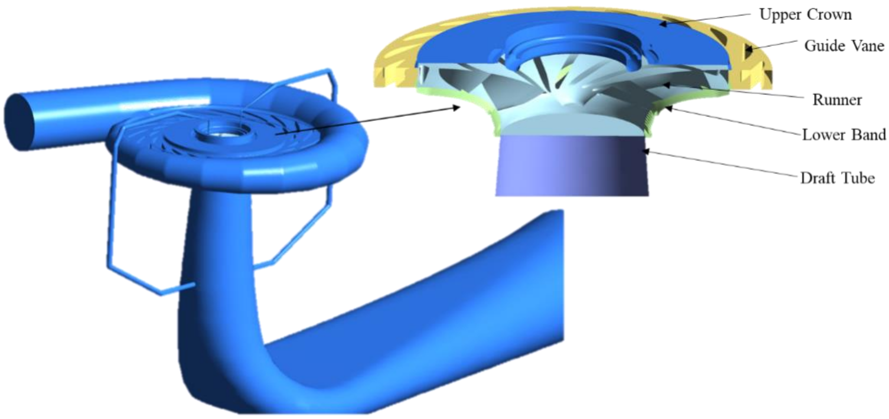


Figure 1. Three-dimensional model of the pump-turbine.

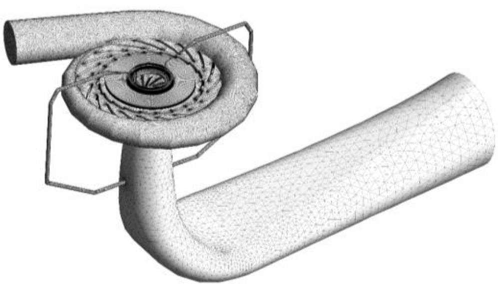


Figure 2. Mesh of the flow domain.

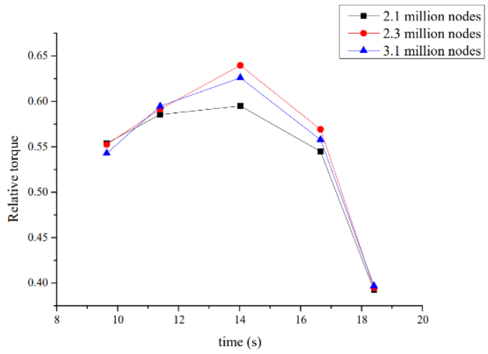


Figure 3. Mesh independence test.

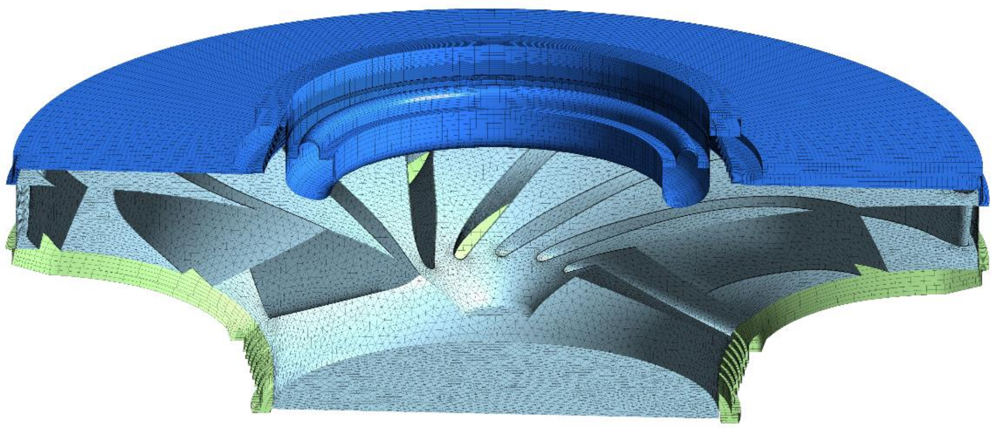


Figure 4. Mesh of runner and clearances.

The boundary conditions calculated by MOC are shown in Figure 5. The internal flow of the pump-turbine unit at typical time points was calculated. The relative pressure coefficient C_p is defined as

$$C_p = \frac{p}{\rho g H} \tag{5}$$

where p is the pressure, H is the rated head, which is 545 m in this paper. Total pressure P_{in} was set at the inlet of the spiral casing, and static pressure P_{out} was set at the outlet of the draft tube. The no-slip wall boundary conditions were set at all walls. Rotor-stator interfaces were used between the runner and static domains.

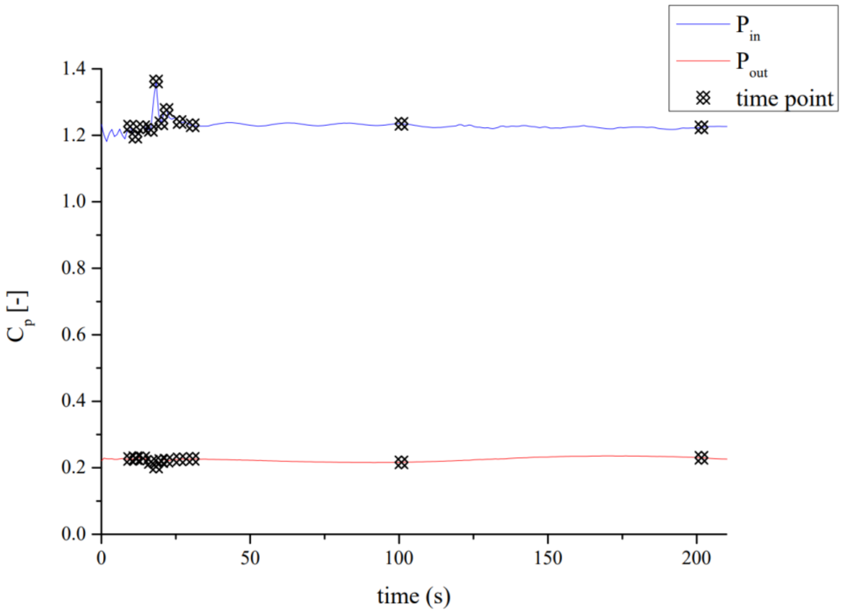


Figure 5. Inlet and outlet boundary conditions.

3. Results and discussion

3.1. Performance characteristics

The relative rotation speed, relative flow rate and relative guide vane opening during the start-up process calculated by MOC are shown in Figure 6. In three-dimensional calculation, typical time points are set at the turning point of these curves to characterize the start-up process. When the opening of the guide vane increases the flow rate increases, which resulted in the increase of the torque acting on the runner, then the speed rose rapidly to the rated speed. Then the opening of the guide vane decreased and the flow rate decreased to the no-load operation area. At 100 s, the guide vanes reopened. The unit reached the rated working condition at 200 s.

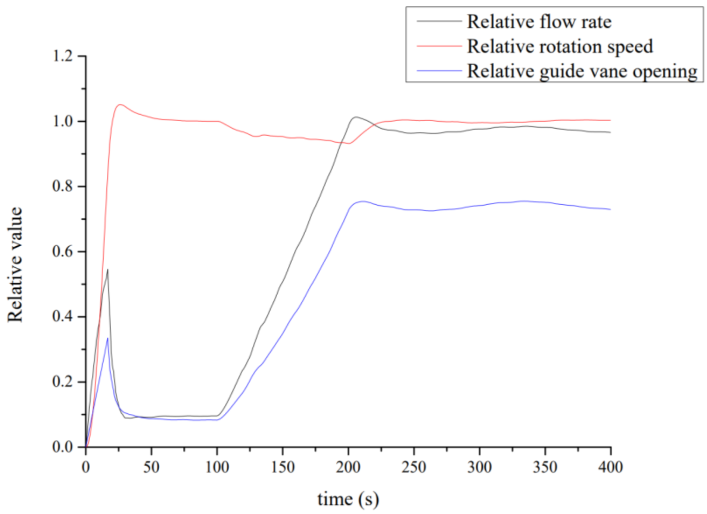


Figure 6. The relative flow rate, relative rotation speed and relative guide vane opening of the unit.

A comparison of the relative torque results calculated for the model with and without clearances is shown in Figure 7. The trend of torque variation is consistent with the prediction of the analysis. There is no significant difference between the results of the two models. The relative pressure distribution results at typical time point 201.59 s are shown in Figure 8 and Figure 9. These results indicate that the clearance has little effect on the internal flow in the runner. Therefore, pressure distribution on the runner blades is very close in the two results and is the main factor affecting torque. The inner surface of the upper crown and the lower band, which is ignored in the model without clearances, is almost no torque loading.

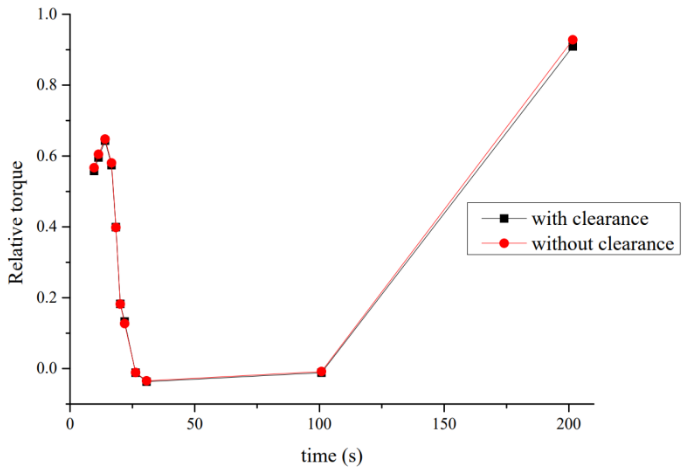


Figure 7. The relative torque of the model with and without clearances.

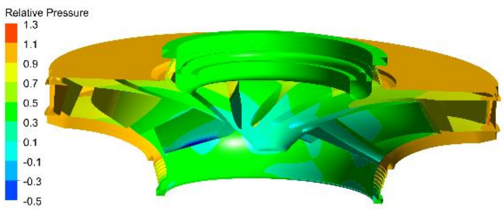


Figure 8. Relative pressure of the model with clearance

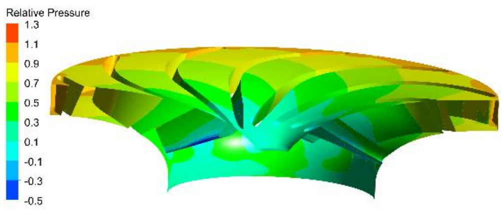


Figure 9. Relative pressure of the model without clearance

3.2. The axial hydraulic force comparison

As shown in Equation (6), the relative axial hydraulic force F_z^* is defined. F_z is the axial hydraulic force calculated by three-dimensional CFD calculations. For calculation with clearance, F_z consists of forces on the fluid domain of runner, upper crown and lower band. For calculation without clearance, F_z represents only the axial force on the runner.

$$F_z^* = \frac{F_z}{mg} \tag{6}$$

where m is the total mass of rotating components of the unit.

According to the definition of the +z direction, the upward axial force is defined as positive. As shown in Figure 10, there is a great difference of the axial hydraulic force between the two results during the start-up process. The flow surface of the runner can be divided into three parts: the inner surface of the upper crown (F_z-UC), the inner surface of the lower band (F_z-LB) and the whole surface of the internal runner flow passage (F_z-R). The hydraulic force on the three parts of flow surfaces and the comparison between F_z-R and F_z-without clearance is shown in Figure 11. F_z-UC is downward and F_z-LB is upward. The absolute values of both forces increase with the increase of rotating speed. The change trends of F_z-R and F_z-without clearance are basically the same as the torque. As the flow rate of the unit decreases during the no-load operation area, the pressure difference between the front and back of the blade decreases, and the torque of the blade decreases. The force on the upper crown and the lower band is several times the force on the internal runner, therefore, it plays a major role in the performance of the axial hydraulic force and the model without clearance cannot simulate the real stress surface of the runner.

In addition, the flow pressure in the clearance is significantly higher than that in the internal runner, especially when the opening of the guide vane is small. The tangential velocity of the water flowing into the runner is high, forming a "water ring" that rotates at high speed in the vaneless area, which increases the pressure in the vaneless area and the pressure in the clearance connecting the vaneless areas These results are shown in Figure 12.

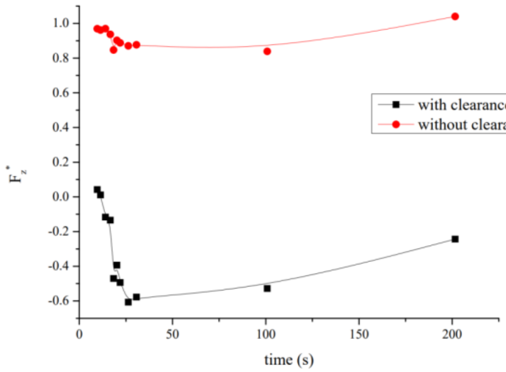
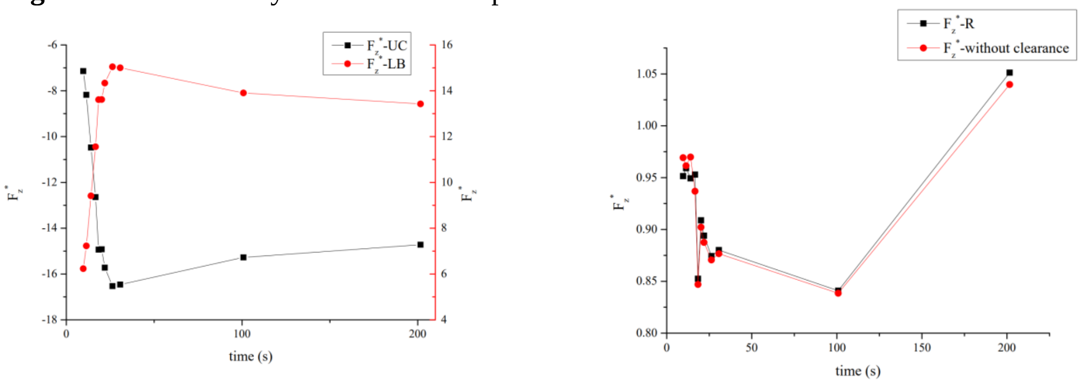
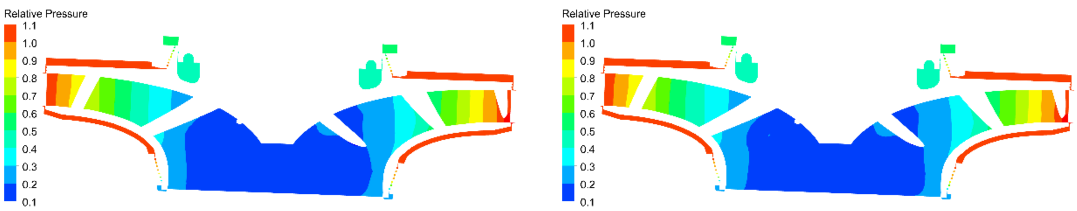


Figure 10. The axial hydraulic force comparison of the model with and without clearances.

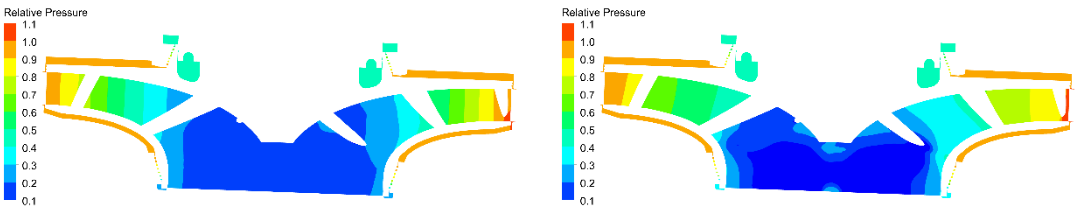


(a) F_z-UC and F_z-LB (b) F_z-R and F_z-without clearance

Figure 11. The axial hydraulic force decomposition.



(a) 26.29 s (b) 30.68 s



(c) 100.80 s (d) 201.59 s

Figure 12. Pressure on the section of clearance and runner.

4. Conclusion

In this paper, a coupled one-dimensional-three-dimensional numerical calculations method is used to explore the clearance effects on the flow characteristics in a high-head prototype pump-turbine during the start-up process in turbine mode. The model with clearance and without clearance is simulated respectively in the three-dimensional calculations to obtain and compare the external characteristics and the internal flow characteristics of the unit. It is found that the results of the hydraulic torque of runner are close. The clearance has little effect on the internal flow in the runner and the inner surface of the clearance loads little torque. The axial hydraulic force on the runner is analysed. There is a great difference of pressure inside and outside the runner. The axial hydraulic force on the inner surface of the clearance is significant. According to this study results, further simulation can reasonably simplify the model of clearance.

In this study, the clearance effects during the pump-turbine start-up process in the time domain are clarified. However, the transient frequency characteristics are difficult to express in this way. Further work will focus on this aspect.

Acknowledgments

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