Performance variation study on bulb turbines with different water head under considering free surface and water gravity

To cite this article: Yaping Zhao et al 2019 IOP Conf. Ser.: Earth Environ. Sci. 240 022057

View the article online for updates and enhancements.

You may also like
- Energy model of pumped hydro storage station
  Huafeng Li, Zhizhong Guo and Zhe Ding
- The simulation analysis and operation conditions optimization of low-head water bulb tubular turbine based on different guide vane outlet angle
  Z G Li, B Wang, B Ma et al.
- Flow characteristics on the blade channel vortex in the Francis turbine
  P C Guo, Z N Wang, X Q Luo et al.
Performance variation study on bulb turbines with different water head under considering free surface and water gravity

Yaping Zhao 1*, Zhihua Li 1,2, Weili Liao 1, Xiaobo Zheng 1, Xingqi Luo 1

1. Institute of Water Resources and Hydro-electric Engineering, Xi’an University of Technology, Xi’an 710048, China
2. Xi’an thermal power research institute co., ltd, Xi’an 710054, China

E-mail: zyp0168@163.com

Abstract. The maximum and minimum water head difference of bulb turbine is only a few meters to ten meters, and the performance is highly sensitive to the change of water head, especially in low head operation, the bulb turbine performance deteriorates and vibration intensifies, this seriously affect the normal operation of the power station. Therefore, considering the free surface and water gravity, a numerical method is adopted to study the performance of prototype bulb turbines at different water head, by analysing the distribution of flow parameters with water head in hydraulic turbine, it is revealed that the stress of blade is uneven and the cause of cavitation and vibration at low head operation. The results show that: with the decrease of water head, the cavitation performance of the runner becomes worse, the low pressure area moves from the suction side to the pressure side of blade; the water torque of the guide vanes and the runner blades in different positions is different, which could cause the guide vane and blade angle asynchronism during adjustment; with the decrease of water head, the torque difference generated by the blades at different positions on the rotating shaft of the hydraulic turbine increase, the blades experience dynamic stresses, It threatens the strength of the blade and the stable operation of the unit.

1. Introduction

In recent years, the development of the water resources has gradually turned to the low head and ultra-low head. Due to the unique advantages, the bulb turbine has become a good model for the development of extra-large flow, low head and ultra-low head hydraulic resources and tidal energy, it also has obvious advantages and broad application prospects in the development of river cascades [1].

At present, the optimal design of flow passage components [2,3] and the internal flow characteristics analysis [4-6] for bulb turbine is mainly conducted by the experiment or numerical simulation for the model turbine. Although the model research can improve the design and operating performance of the bulb turbine, the Insufficient output, severe vibration and other issues still occur frequently in various tubular power stations. The emergence of these problems is mainly due to the fact that it is difficult to meet the geometric similarity and flow similarity in model test, the operational performance of the turbine obtained by the model test is different from the actual situation. Even though the real machine size was used for numerical simulation research, due to neglecting the effect of free surface and water gravity for the turbine unit, the inlet flow of the turbine unit is uniform, the inlet flow of the runner has a good axial symmetry, there are still differences between research results and the real situation, such as the cavitation field only appears on the top of shroud in real machine,
but in the model test, the cavitation may involve the whole runner \[7\]. The literature \[8\] shows that the change of flow state caused by the inlet inclination of unit can directly alter the combination relationship of the turbine. It is clear that the accurate prediction of performance for bulb turbine is still an important issue for the hydraulic machinery researchers. In this, taking the exploration of boundary conditions that are in line with the actual situation as an entry point, the author of this paper carried out a numerical research method to investigate the performance of a real bulb turbine with considering the water gravity and free surface, and conducted relevant test verification, the results are consistent with the actual situation \[9-10\].

The bulb turbine has large diameter and low head, the maximum and minimum water head difference of bulb turbine is only a few meters to ten meters, and the performance is highly sensitive to the change of water head. Therefore, based on the previous research results, the performance numerical investigation of prototype bulb turbine in different water head is conducted with considering the free surface and water gravity. The cause of the uneven stress on blade, the cavitation and vibration at low head operation is revealed by analyzing the distribution of flow parameters with the change of water head in hydraulic turbine.

2. CFD methods

2.1. Fluid control equation

Continuity equation, momentum conservation equation and energy conservation equations are the basic equations to describe the flow regularity. Generally, water is regarded as an incompressible fluid flow, the heat exchange is very small, and the energy conservation is often ignored in the flow that water is taken as the medium. Based on these reasons, for the complex three-dimensional incompressible viscous flow in the turbine, the basic equations can be described as:

Continuity Equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0 \tag{1}$$

Where: \(\rho\) is the density, kg/m\(^3\); \(t\) is time, s; \(u, v, w\) are the components of the fluid velocity vector along the coordinate axis, m/s.

For incompressible fluids, the density \(\rho\) is a constant; the mass conservation equation can be simplified to:

$$\frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0 \tag{2}$$

When the fluid is incompressible and the viscosity coefficient is constant, the momentum equation is simplified to the following equation:

$$\frac{\partial u}{\partial t} + \text{div}(uU) = \text{div}(\nu \text{grad}u) - \frac{1}{\rho} \frac{\partial p}{\partial x} + F_x \tag{3}$$

$$\frac{\partial v}{\partial t} + \text{div}(vU) = \text{div}(\nu \text{grad}v) - \frac{1}{\rho} \frac{\partial p}{\partial y} + F_y \tag{4}$$

$$\frac{\partial w}{\partial t} + \text{div}(wU) = \text{div}(\nu \text{grad}w) - \frac{1}{\rho} \frac{\partial p}{\partial z} + F_z \tag{5}$$

Where: \(U\) is the fluid velocity vector, m/s; \(\nu\) is the hydrodynamic viscosity, Pa·s; \(\nu\) is the viscosity of the second molecule of the fluid; \(\nu\) is the viscosity of the fluid motion, m\(^2\)/s; \(p\) is the pressure, Pa; \(F_x, F_y, F_z\) and \(F_i\) is the physical force on the microelement.

It is not closed when describing the turbulent motion by averaging N-S equations and the turbulence model should be introduced to make the equations closed. The standard k-\(\epsilon\) turbulence model is stable, simple and economic, and has sufficient accuracy in a wide range of applications, including boundary layer flow, pipe flow, shear flow, so it has been widely used. Therefore, the
standard k-ε turbulence model is used to simulate the flow characteristics of the computational domain in this paper.

2.2. Free surface tracking

The flow simulation with the free surface focus on how to track the free surface, there are many ways to solve the problem, such as steel lid law, marking particle method, the height function method, VOF method. The advantage of VOF (Volume of Fluid) method is the only one can describe a variety of complex changes in the free surface. So, in this paper, VOF method is used to solve the water-air two-phase flow in bulb turbine with considering free surface, and the position of the free surface is determined by establishing and solving the transport and diffusion equation of the volume function.

The transport and diffusion equation of the volume function:

\[
\frac{\partial V}{\partial t} + u \frac{\partial V}{\partial x} = 0
\]  

(6)

\(V\) is equal to volume of fluid in the element divided by volume of the element. When the volume function \(V\) is equal to 1, the element is filled with liquid; when the volume function \(V\) is equal to 0, there is no liquid in the element; when the value of \(V\) is between 0 and 1, the element is half-filled with liquid.

3. Geometric model and mesh generation

In this paper, a horizontal bulb type hydropower station was set as the research object, the geometry of computational domain is shown in Figure 1; and table 1 shows the basic parameters of the bulb turbine.

| Basic parameter       | Value  |
|-----------------------|--------|
| Runner diameter       | 7.2m   |
| Number of guide vane  | 16     |
| Number of blades      | 3      |
| Hub-tip ratio         | 0.35   |

Table 1. Basic parameter of bulb turbine.

![Figure 1. Geometric model](image)

High-quality structured grids for all components of the turbine units are created by the commercial software ANSYS ICEM-CFD with multi-block templates. The grids of the whole computational domain and turbine local region are shown in figure 2 and the grids distribution of each components of computing domain are shown in table 2.

![Figure 2. Grid distribution.](image)

Due to the large density difference between water and vapour, the phase interface is often clear at the free-surface, and the sparse grids near the free-surface will cause a wider phase interface and a greater error of the results, so the grids are refined near the free surface. Meanwhile, the volume
fraction gradient of the water and vapour changes larger at the phase interface; denser grids can help the better development between the water phase and the vapour phase. Therefore, the grids near the free surface is encrypted during generating the grids, this method not only can prevent the calculation speed slow down caused by the dense grids of the entire computational domain effectively, but also can make a good simulation for the phase interface.

Table2. Distribution of grid number.

| Flow components                        | Number of grid Nodes/10^4 | Number of grid elements/10^4 |
|----------------------------------------|---------------------------|-----------------------------|
| Inlet section and upstream of the reservoir | 115.3656                  | 111.5892                    |
| Guide vane                             | 83.8080                   | 78.4160                     |
| Runner                                 | 43.6912                   | 41.0184                     |
| Draft tube and downstream of the reservoir | 52.1569                   | 50.5212                     |
| Total                                  | 295.0217                  | 281.5448                    |

4. Calculated operating points and boundary condition
The basic parameters of each operating point are shown in table 3.

Table 3. Basic parameters of the operating points.

| Operating points                        | Water head(m) | rotate speed(r/min) | Flow rate (m³/s) |
|----------------------------------------|---------------|---------------------|------------------|
| Operating point 1                       | H1=7.3        | 75                  | 298              |
| (Optimum operating point)              |               |                     |                  |
| Operating point 2                       | H2=6.1        | 75                  | 406              |
| (Rated operating point)                |               |                     |                  |
| Operating point 3                       | H3=3.5        | 75                  | 320              |
| (minimum head operating point with rated output) |           |                     |                  |

Tubular power stations are typical low-head hydroelectric plants without storage, the water level in the upper reservoir remains unchanged, and the water level in the downstream reservoir changes according to the flow rate of turbines under different operating conditions. Therefore, for the tubular power plant with three-blade, the upper stream level remains normal water level and unchanged at the three different operating conditions, and the water head under different operating conditions is satisfied by the different downstream water levels, as shown in figure 3.

The initial flow field is shown in figure 4. The red region indicates the initial position of the water, and the blue region indicates the initial position of the air. Boundary conditions are set as followings:
Inlet of the upstream reservoir: free surface height and hydrostatic pressure, the flow rate is a very small value.
Outlet of downstream reservoir: free surface height and hydrostatic pressure.
Top of reservoir: opening surface (water volume fraction is 0).
Rotating components: the runner is rotating component, the rotating speed is given. Solid wall: solid wall with smooth no-slip boundary. Medium: water and air.

5. Result of analysis

In order to research the pressure change on runner blade during rotating, four monitoring points are defined (Shown in figure 5(a)). Figure 5(b) shows the pressure fluctuation of different monitoring points on the blade surface. It can be found that the pressure on blade is fluctuating periodically during rotating in each operating points, shown as Figure 5.

Due to the direction of rotation, the initial phase of pressure fluctuation increase gradually from the leading edge of blade to the trailing edge of blade. The variation of pressure fluctuation at each monitoring points is different. Due to the different entrance flow condition at different operating points, the biggest pressure fluctuation is in the leading edge of blade, and the pressure fluctuation in blade near the hub is smaller.

At operating point 1, the Minimum pressure area on the blade is located near the hub. Since the pressure on the pressure side of the blade is greater than the saturation vaporization pressure of water (3754[Pa]), there is no cavitation. At operating point 2, when the blade moves to the top of the shroud, the pressure on the pressure side of the blade approximately to the saturation vaporization pressure of water, so the cavitation occurs firstly at the leading edge of blade. At operating point 3, when the blade moves to the top of the shroud, the pressure is less than the saturation vaporization pressure of water, the cavitation has occurred at the leading edge of blade at this operating point. Thus, with the decrease of the water head, the cavitation position of bulb turbine moves from the blade head to the pressure side of blade, and the cavitation performance deteriorate rapidly. It is obvious that the frequencies for the pressure fluctuations are equal to the rotational frequency of the turbine.

The bulb turbine is dual-adjustable. When the operating condition changes, the operating force required for adjusting the guide vane opening and blade rotation angle is generated by the respective servomotor, the operating torque of the servomotor is also affected by the hydraulic moment acting on guide vanes and blades. Figure 6 and figure 7 shows the hydraulic moment coefficients along the respective rotation axis of guide vanes and runner blades at different positions.

For guide vanes, affected by the free surface and water gravity, the hydraulic moments of the guide vanes in different positions vary in the circumferential direction, and with the water head increasing,
the hydraulic moment difference of guide vanes in different positions becomes larger. For runner blade, the higher the water head, the worse the homogeneity of the hydraulic moment distribution on the blade in different positions. The hydraulic moment of the blade located at the top of the runner is the largest, while the moment of the blade located at the center of the runner is the lowest.

**Figure 6.** hydraulic moment coefficient of guide vanes on their rotor shaft at different head

**Figure 7.** hydraulic moment coefficient of blades on their rotor shaft at different head

In order to ensure the smoothness and evenness of the water flow, in the design and operation of the bulb turbine, the guide vane openings and blade rotation angles in different position are asked to be the same. However, for the horizontal cross-flow type unit, due to the free surface and the water gravity, the hydraulic moments of guide vanes and the runner blades in different positions are different, It is easy to cause asynchronism for servomotor in the guide vane opening and blade rotation angle adjustment, which causes the axial symmetry of the water flow in the unit to deteriorate, and has a great adverse effect on the stability of the unit.

**Figure 8.** Torque distribution produced by the different positions blades to the shaft.

Figure 8 shows the torque distribution of blade at different positions when the heads are different. Under different heads, there are differences in the torque produced by blades of different positions on the turbine shaft, and the torque difference also varies with the change of water head. When H = 3.5m, the difference in the torque is the largest, and the difference between the maximum torque and the minimum torque is about 2.6%. When H = 7.3 m, the torque is relatively close, and the difference between the maximum and minimum values is 0.9%. It can be seen that the difference in torque of the blades at different positions on the turbine shaft increases with the water head decreasing. During the rotation of the turbine, the blade will experience a dynamic stress with larger change of the magnitude, which threatens the strength of the blade and the safe and stable operation of the turbine.

6. Conclusions
In this paper, considering the free surface and water gravity, a numerical method is adopted to study the performance of prototype bulb turbines at different water head, by analysing the distribution of
flow parameters with the changes of water head in prototype bulb turbine, the cause of the uneven stress on blade, cavitation and vibration at low head operation is revealed. the main conclusions are as follows:

1. With the decrease of the running head, the low pressure area in the runner moves from the back of the blade to the front of the blade, and the cavitation performance of the runner becomes worse;
2. Because of the free surface and the gravity of the water flow, the water torque of the guide vanes and the runner blades in different positions is different, the difference increases with the increase of the head, easily cause the guide vane and blade angle asynchronism during adjustment, lead the axial symmetry of the water to become worse;
3. With the decrease of water head, the torque difference generated by the blades at different positions on the rotating shaft of the hydraulic turbine increase, the rotation of the runner causes the blades to experience dynamic stresses of greater magnitude; it threatens the strength of the blade and the stable operation of the unit.

Acknowledgments
This work was supported by the National Natural Science Foundation of China (51679196), National Natural Science Foundation of China (51339005), and the Scientific Research Project of Education Department of Shaanxi Provincial Government (17JK0568).

References
[1] Simon P., Neill, Emmer J. The impact of tidal stream turbines on large-scale sediment dynamics [J]. Renewable Energy, 2009, 34: 2803 – 2812.
[2] Yaping Z., Weili L., Zhihua L., Hui R., Xingqi L. Flow field performance of bulb turbine with C-type or S-type blades[J]. Transactions of the Chinese Society of Agricultural Engineering, 2013, 29(17): 47-53 +294.
[3] Guojun Z., Pengcheng G., Xingqi L., et al. Optimal design of runner blade in bulb turbine base on multidisciplinary feasible method[J]. Transactions of the Chinese Society of Agricultural Engineering, 2014,30(2): 47 − 55.
[4] Vigfús Arnar Jósefsson. Flow analysis of a rotating tidal turbine blade using a dynamic mesh[D]. Master’s Thesis of University of Iceland. 2011.
[5] V Guénette, S Houde, G D Ciocan, G Dumas, J Huang, C Deschênes. Numerical prediction of a bulb turbine performance hill chart through RANS simulations[C]// Beijing, 26th IAHR Symposium on Hydraulic Machinery and Systems. 2012.
[6] Haithacha Sudsuansee, Udomkiat Nontakaew, Yodchai Tiaple. Simulation of leading edge cavitation on bulb turbine[J]. Songklanakarin J. Sci. Technol. 2011, 33 (1): 51-60.
[7] Shutang T. Analysis and Evaluation of Operation Performance for Bulb Turbine[J]. Dongfangdianji, 1994, 1: 79-85.
[8] M Benišek, I Božić, B Ignjatović. The comparative analysis of model and prototype test results of bulb turbine[C]// Romania, 25th IAHR Symposium on Hydraulic Machinery and Systems, 2010.
[9] Yaping Z., Weili L., Haoda F., Hui R., Xingqi L. Experimental and numerical study on inlet and outlet conditions of a bulb turbine with considering free surface[C]// Beijing, 26th IAHR Symposium on Hydraulic Machinery and Systems, 2012.
[10] Yaping Z., Weili L., Zhihua L., Hui R., Xingqi L. Numerical study on the performance of bulb turbine by using runaway condition to flood discharge and sediment dredging[C]// Montreal, 27th IAHR Symposium on Hydraulic Machinery and Systems, 2014.