Research and optimization of performances of a pump turbine in pump mode

P Xue¹, ZP Liu¹, L Lu¹, YJ Tian¹, X Wang¹ and R Chen¹

¹China Institute of Water Resources and Hydropower Research, Beijing 100038, China

Email: xuepeng09iwhr@163.com

Abstract. For reversible pump turbines, a sufficient hump safety margin has to be provided under small-flow and high-head conditions to avoid instability resulting from operations in the hump-shaped zone. On the other hand, under large flow conditions, cavitation easily occurs on the low-pressure edge of runner blades due to effects of low pressure and flow separation, sharply reducing the efficiency and lift head. Therefore, the hydraulic design of the pump turbine in pump mode is crucial for the properties of energy, cavitation and stability of the whole unit. A model runner (runner A) designed for a pumped storage power plant in China was demonstrated with insufficient hump safety margin and poor cavitation performance. To improve those deficiencies, optimization design was carried out on the geometric dimensions of the runner A, based on the computational fluid dynamics (CFD) analyses and model test results. The final model runner B was designed with less blade number Z and large blade wrap angle θ, to both alleviate the flow separation around the lower pressure edge of the blade and the non-uniform velocity distribution in the outlet area around high pressure edge. The model test results show that in comparison with the original runner A, the performances, such as efficiency, hump safety margin, cavitation and stability are improved in runner B.

1. Introduction
Pumped storage power stations (PSPS) play an increasingly important role in maintaining the stability of the power grids and improving the utilization of energy. The pump turbine is the core equipment in a modern PSPS and its performance directly determines the economic benefits, safety and stability of the power station. The hydraulic design and calculation of pump turbines are usually carried out in pump mode, and the runners are checked in turbine mode [1]. During the design, the performances, including efficiency, cavitation, and stability in pump mode and turbine mode should be both taken into consideration. A matching relationship of the energy parameters is also required in the two modes. These targets interact with and sometimes even contradict each other; therefore it is very difficult to design a model with excellent performances in all aspects [2]. At present, the general method used is to design an initial runner at first and then optimize the runner through multiple times of numerical simulation, model test and optimization to achieve the final design objectives [3].

For the design of a pump turbine in pump mode, it is more difficult to achieve the high level performance due to the divergent flow passage. A pump turbine generally has a narrow high-efficiency range in pump mode because of the instability flow pattern in both small and large flow conditions. Cavitation tends to occur on the low-pressure edge of runner blades under large-flow conditions caused by effects of low pressure and flow separation, thus leading to a sharp decrease in the efficiency and lift head. Under the small-flow and high-head conditions, the flow-head curves show a positive slope section [4] that is known as the hump-shaped zone (as displayed in Figure 1).
When generating units are operated in this zone, the pressure oscillates greatly. For this reason, a sufficient hump safety margin has to be reserved for the flow-head characteristics of the runner, so as to avoid operating in the hump-shaped zone. Therefore, during the design of the pump turbine in pump mode, extensive studies must be carried out to expand the high-efficiency operating zone, which is crucial for performances of the whole unit in energy efficiency, cavitation and stability.

Studies of the rotating stall phenomenon and pressure pulsation characteristics in the hump-shaped zone in pump mode have been carried out through model tests and numerical simulations [5]. Wang et al. [6] studied the pressure fluctuation characteristics of a pump turbine in the hump-shaped zone in pump mode on a high-precision model test platform and found that pressure pulsation in vaneless space increases as guide vane opening increasing. By injecting bubbles into the model, Yang et al. [7] observed the rotating stall structure in the flow channel of guide vanes in the hump-shaped zone through model tests and investigated the relationship between the structure and pressure pulsation. Ran et al. [8] conducted a model test and researched on the double-hump phenomenon and the corresponding characteristic of pressure pulsation of a pump turbine appearing in the small flow rate zone near the optimal working condition. In the test, the first hump of the flow-head curve appears in the zone where the flow rate is 0.85~0.9 times of that in the optimal efficiency condition in the pump mode. In the meanwhile, the amplitude of pressure pulsation in the vaneless space is significantly larger than that in the stable mode. Many studies have analyzed the pressure pulsation and flow patterns in the vaneless space of pump turbines in the normal operating mode [9-11] and in the hump-shaped zone in unstable operating mode [12, 13] based on numerical simulation. The results of those studies indicate that operating in the hump zone is harmful to the unit and even the power station, thus sufficient hump safety margin must be provided. The hump-shaped zone could be closely related to the matching relationship between the flow velocity at the high-pressure edge of runner blades and the angle of guide vanes.

A runner (runner A) designed for a pumped storage power plant in China was studied to improve its deficiencies, such as insufficient hump safety margin and poor cavitation performance at the large flow rate working condition demonstrated by the model test. The numerical analysis of the flow patterns in the channels between runner blades, guide vanes and stay vanes were conducted. Analyzing the simulation results complemented with model test results, optimization was carried out and finally runner B was obtained. The model test results showed that the runner B presented better performances, like efficiency, hump safety margin, cavitation and stability compared to the original runner A.
2. Numerical analysis and optimization

2.1. Parameters of the origin pump turbine

The given design parameters of the pump turbine in the PSPS are shown in Table 1. The ratio of the maximum head to the minimum head is $H_{\text{max}}/H_{\text{min}}=1.43$, which is currently the largest ratio of fixed-speed pumped-storage units in China. In addition, the power station combines conventional units with pumped-storage power generating units and the maximum submerged depth in the design is -16m. All these are challenges to the hydraulic design.

| $n_1$ (r/min) | $Q_1$ (m$^3$/s) |
|--------------|-----------------|
| 65.24        | 0.76            |

According to the given design parameters, the runner A was designed at the first. The model test results indicate that the head entered the hump-shaped zone before reaching the maximum pumping head required by the project. In the large flow rate working condition, cavitation occurred at the low-pressure edge of runner blades such that the efficiency dropped sharply, accompanying with disordered flow pattern and large pressure pulsation. To meet the project requirements, CFD simulations were carried out to analyze the causes of the insufficient performances of runner A.

2.2. Numerical analysis

2.2.1 Numerical simulation. In order to explore the causes for the occurrence of hump-shaped curve in pump mode of runner A, the simulation of three-dimensional turbulent flow through whole flow passage was carried out in the model size dimension. The internal flow field in the area around guide vanes was analyzed. The computational domain can be seen in Figure 2 and dimensions are in Table 2.

![Figure 2. Computational domain of the pump turbine model.](image-url)
Table 2. Dimensions of model A.

| Model                  | Runner A                  |
|------------------------|----------------------------|
| Diameter of high pressure edge D₁₅₀ (m) | 0.400                     |
| Diameter of low pressure edge D₂₅₀ (m)  | 0.280                     |
| Blade number Z         | 9                          |
| Spiral case            | Wrap angle 349.5°          |
| Stay ring              | Number of stay vane 20     |
| Guide vane             | Height (mm) 74             |
|                        | Number 20                  |
| Draft tube             | Length (m) 2.76            |

Table 3. Mesh information of each component of the model pump turbine.

| Component                      | Spiral case, guide vane and stay vane | Runner | Draft tube | Total     |
|--------------------------------|----------------------------------------|--------|------------|-----------|
| Number of nodes                | 6634081                                | 7524771| 3088073    | 17246925  |

SST k-ω turbulence model was used for CFD computation to close Reynolds-averaged Navier-Stokes (RANS) equations through commercial soft ANSYS CFX. The SST k-ω turbulence model can acquire the separated flow of a pump turbine at the small flow rate[14]. Taking the influence of boundary layers into account, y+ was controlled in the range of 30~300 and the value of major flow regions was about 100 to meet the stability requirement of the SST k-ω turbulence model and numerical values.

The governing equation was discretized by using finite volume method. The source item and diffusion item were discretized by applying the second-order central difference scheme while the second-order upwind difference scheme was utilized to discretize the convective term.

When determining the boundary condition of numerical calculation, the inlet velocity was given and the turbulent energy at the inlet was designed as 5% of the average kinetic energy. The average static pressure was considered as the boundary condition of outlet and the relative pressure was set as 0 Pa. Non-slipping wall condition was adopted for the boundary condition of solid wall. Moreover, the values of the velocity, k and ω near wall region were calculated by using the wall function.

2.2.2 Results analysis. The calculation was conducted under the working point in the hump zone in pump mode according to the model test results. The guide vane opening a₀ is 20mm and the flow coefficient φ is 0.164. Figure 3 includes the velocity profile in the runner and flow channels between guide vanes and stay vanes.

As shown in the calculation results: 1. Large scale vortices appeared in large areas near the suction side of guide vanes and stay vanes, which increases losses; 2. The incidence angle to the inlet of low pressure edge is large, especially the part near the rim of the runner, where large scale vortices can be found; 3. non-uniform velocity distribution formed at the outlet area of the runner, from the suction side of one blade to the pressure side of the adjacent blade. Low velocity flow occupied considerable part of the flow passages on the suction side of blades. The combined effects of these observations lead to hydraulic loss and insufficient circumference velocity, which makes runner A fail to satisfy the design requirements. Meanwhile, disordered flow patterns and large differences between flow velocities among the flow channels of guide vanes can also be observed, which eventually causes instability of generating units.
2.3. optimization
Blade number, wrap angle, inlet angle and outlet angle were changed during the optimization. Different combinations of the blade number and wrap angle were chosen based on the previous design models and statistic data. The analysis was conducted by comparing the guide vane angle with the outflow angle in the vaneless space in different height sections, which were calculated based on numerical simulation. Based on the comparative study and model test analysis, the high-pressure edge angle and low-pressure edge angle of the blade were optimized, to enhance the head where the humping phenomenon occurs. However the performance in the turbine mode also requires to be considered during the optimization. In addition, high efficiency, favorable cavitation performance, optimal pressure pulsation characteristic and acceptable S-shaped characteristic zone for the runner have to be guaranteed when improving the flow-head characteristics. By conducting multiple rounds of optimization design, numerical simulation and assessment, the runner B were consequently designed. The runner B has 7 blades and its blade wrap angle on the crown stream surface is increased from 95° to 111° compared to runner A. The inlet diameter $D_{2m}$ is slightly increased to 0.288 m.

3. Test results and analysis

3.1. Model test platform and test conditions
Comprehensive model test of runner B was carried out on the high-precision universal hydraulic model test platform TP1 in Hydraulic Machinery Laboratory of China Institute of Water Resources and Hydropower Research. The test platform TP1 was a closed circulating system and can be used to test performances of all kinds of water turbines, pumps, and pump turbines. The TP1 was equipped with measurement instruments including high-precision electromagnetic flow-meters, differential pressure sensors, absolute pressure sensors, torque sensors, as well as in-situ calibration systems. The forward and backward in-situ calibrations for the electromagnetic flow-meters were conducted before and after the model test by discharge calibration system. The load transducers of the discharge calibration system were finally calibrated by the standard weight. According to the calibration, the flow rate measurement error was less than 0.13%. The measurement accuracy of all test parameters and operation stability all satisfied the requirements of relevant codes stipulated by international standard IEC60193. The comprehensive error of the platform in the model efficiency test was less than ±0.2%. The assembly model equipment on the test rig was shown in Figure 4. There are totally 7 measurement points of pressure fluctuation on the spiral case, head cover, draft-tube and elbow tube. During the tests in the pump mode, the rotation speed is 1200 r/min and the Reynolds number is $6 \times 10^6$. 

![Figure 3. Steamlines in the runner (a) and velocity profile in flow channels between guide vanes and stay vanes (b).](image-url)
3.2. Comparative analysis of the test results

The results of the model test were converted into non-dimensional parameters such as the flow coefficient $\phi$ and the pressure coefficient $\psi$ [4]. By selecting the guide vane opening as 20 mm at which the runner A presented the optimal efficiency, important performances of runner A and runner B were compared, including the flow-head characteristics (Figure 5A) and efficiency in pump mode (Figure 5B), efficiency in turbine mode (Figure 6) and pressure fluctuation measured in vaneless space in the pump mode at on-cam relationship working condition (Figure 7).

It can be seen from the figures that the runner B was improved in terms of the hump safety margin and the stable operation range under the large flow condition. In the pump mode, the runner presented obviously higher efficiency than the runner A in partial flow rate zones. The two runners showed basically same efficiency in the turbine mode, and the optimal efficiency of the runner B was higher.
than the runner A. By comparing the pressure pulsations measured in vaneless space in on-cam operation condition in pump mode (Figure 7), we found that the amplitude of pressure pulsation of the runner B reduced apparently compared with the runner A in almost all zones except near the rated head, where it was 1% higher. Therefore, the pump turbine showed greatly improved stability.

4. Conclusion
The disadvantage of insufficient hump safety margin was studied on a preliminary designed model runner (runner A) for a PSPS in China. Numerical simulations were carried out. The results showed that serious flow separation and large scale vortex emerged in the blade channels and on the suction side of guide vanes and stay vanes. Consequently losses increased significantly, which contribute to the formation of positive slope section in the flow-head curves. Specifically, velocity distribution at the outlet area of runner A was extremely non-uniform which increased losses and the average circumference velocity at this region was beyond the design objectives. The combined effects mentioned above caused the insufficiency of hump safety margin, which made the pump turbine fail to satisfy design requirements.

Based on this study, optimization design was carried out for multiple rounds and an alternative model runner B was developed. The blade number was reduced and the wrap angle was increased compared to those of the runner A. Finally, the runner B was tested on the high-precision model test platform for hydraulic machines. The test results demonstrated that lift head of the runner B was higher than the preliminary design when the hump phenomenon occurs. In the mean while the performances of cavitation and pressure fluctuation were also be improved with evidences.

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