Numerical prediction of the pressure fluctuations on small discharge condition of a pump-turbine at pump mode

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Abstract. The operational stability of the pump turbine at the pump mode will be greatly influenced by large pressure fluctuations when operated in the small-discharge conditions. Therefore, it is significant to analyse the flow characteristic under the small discharge operating conditions deeply. Study of the internal flow in the small discharge condition has been investigate in great detail combined with model experiments in this paper. The SST k-ω turbulence model is adopted to perform three-dimensional numerical simulation of the entire pump-turbine flow passage at optimal guide vanes opening. The numerical simulation results match well with experimental data. Then internal flow under the small discharge condition is analysed. The results show that the dominant frequency inside the flow passage is a relative low frequency. In addition, there are obvious complex flow phenomena inside the draft tube, runner and diffuser domains, such as secondary flow, backflow and even vortex, leading to strong unsteady flow and significant pressure fluctuation.

1. Introduction

With the development of pumped-storage power station technology, the unstable operation issues have drawn more and more attention as pump-turbine’s working condition is relatively complicated. For the pump mode of pump-turbine, the pressure fluctuations become stronger as the discharge gets further from the optimal operating condition, which can seriously affect the operating stability of the pump-turbine [1]. Therefore, it is necessary to analyse deeply on operating conditions in the small discharge conditions of the pump mode.

Study regarding to small-discharge condition’s flow characteristic of pump-turbine in pump mode is not adequate so far. Qu et al [1] analyzed a mixed-flow reversible pump-turbine in pump mode through experiments, finding that the pressure fluctuation is the strongest in the area between the runner and guide vanes and the smallest in the draft tube. Wang et al [2] performed numerical simulation of a pump-turbine, exploring that the vortexes increase and their scales are enlarged as the discharge decreases. Alex et al [3] performed unsteady numerical simulation of a pump-turbine and predicted that the flow field leaving the runner is highly non-uniform under the off-design operating conditions. Ran et al [4] explored pressure fluctuations from experiments, finding that the dominant frequency under the small-discharge off-design condition is a low frequency and the low dominant frequency is believed to be induced by the rotating stall inside the flow passage. Yin et al [5] adopted the DES turbulence model to conduct unsteady numerical simulation, concluding that the low

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dominant frequencies under the small-discharge conditions in the pump mode are mainly caused by the flow separation inside the guide vanes and the fluctuation characteristic is induced by the generation and vanishing frequency. All the researches above lay a foundation for later related study.

In this paper, the small discharge (0.46 \(Q_0\), where \(Q_0\) is the optimal discharge) operating condition of pump-turbine is simulated at the optimal opening. The internal flow field was obtained through numerical calculation and the unsteady flow characteristics at the small discharge operating point were discussed.

2. Simulation model

2.1. Geometry of the model pump-turbine
The three-dimensional entire flow passage of the model pump-turbine is shown in Figure 1. The flow passage is composed of five parts, named draft tube, runner, guide vanes, stay vanes and spiral case. When operated in pump mode, the water begins from the draft tube and flows out from the spiral case. The pump-turbine consists of a runner with 9 blades and a diffuser with 20 guide vanes and 20 stay vanes (including the spiral case node). The runner diameter at outlet is 0.47 m and the optimal opening for the guide vanes is 27 mm. The rated speed of the model pump-turbine reaches 1300rpm. In addition, the discharge of the design point is 408 L/s. In this paper, the discharge is a small one with only 46\% of the design discharge.

![Figure 1. Computational flow domain of a pump-turbine](image)

2.2. Mesh generation
The hexahedral meshes were generated in stay vanes, guide vanes, runner and draft tube domains, while tetrahedral meshes were adopted in the spiral case domain. The grid generation of the entire
flow passage is shown in Figure 2 and the specific cell number of each domain is listed in Table 1. It is obvious that the mesh density of runner, stay and guide vanes is much larger than the draft tube and spiral case domain. The flow speed of the former three domains is high and the internal flow conditions are complicated. Large mesh density is beneficial to capture the complex flow pattern in the three domains in order to improve the calculation precision. While the speed of draft tube and spiral case is comparatively low. It doesn’t have much effect on the overall performance. Small mesh density can save computing resources and raise calculation speed.

**Table 1.** Mesh quantity distribution of the entire flow passage

| Domain   | Draft Tube | Runner | Guide vanes | Stay vanes | Spiral case | Total   |
|----------|------------|--------|-------------|------------|-------------|---------|
| Elements | 418,000    | 1,200,000 | 339,000    | 289,000    | 595,000    | 2,841,000 |

2.3. Boundary conditions and corresponding simulation setup

During the simulation of pump mode, the inlet of the computational domain adopts the velocity-inlet boundary condition so as to get accurate discharge information consistent with the test data. The static pressure is specified at the outlet in order to obtain the pressure distribution consistent with the practical situation. The operating pressure is set as 1 atm. All solid walls are set as no slip. Considering that the SST turbulence model is better for turbine operating conditions \(^{[6-7]}\), the SST turbulence model is chosen ultimately.

The unsteady numerical simulation is on the basis of the steady calculation result. During the simulation process, the time step is set according to both the blade number and the rotating speed. In this paper, 180 transient time steps make up one period calculation. A total of 16 runner revolutions are completed and the data is saved every 2 time steps.

2.4. Recording points distribution

The pressure fluctuations at certain chosen points are recorded during the unsteady numerical simulation of the pump mode. The selected recording points inside the entire flow passage are shown in Figure 3. For the draft tube, a total of 16 recording points are respectively arranged on four planes (named dt14-dt44), as shown in Figure 3(a). On each plane, there are four points uniformly circling around. For the runner domain, there are 8 points rotating with the runner between the two runner blades in the blade passage (named rv1-rv8), as shown in Figure 3(b). In addition, three sets of recording points inside the guide vanes and stay vanes, gv1 to gv4, mv1 to mv4 and sv1 to sv4, are evenly arranged along the circumferential direction on the mid-span plane, as shown in Figure 3(c). For the spiral case, there are five points inside the domain on the mid-span plane (named sp1-sp5), as shown in Figure 3(e).

![Figure 3. Recording points distribution](image-url)
3. Analysis of numerical simulation results

3.1. Hydraulic performance Analysis

Operational parameters were measured at different openings to analyse the operating characteristic of pump mode through a hydraulic machinery rig. Unsteady calculation of the small discharge condition is conducted at the optimal opening in the pump mode of pump-turbine. Table 2 lists both the experimental and calculated head, power and efficiency results of the small discharge. The errors between the simulation and experiment results are also included in the table. The simulated head is close to the experimental result with the error of only -0.58%. The calculated power is relatively smaller than the experimental result with the error of -5.56%. The calculated efficiency is larger than the experimental result as the simulated power is slightly smaller than the practical condition. Generally speaking, the errors between the experimental results and the simulation results are all less than 5.6%. It is feasible to adopt the numerical simulation to analyse the internal flow characteristics under the small discharge operating condition.

Table 2. Experimental and calculated head, power and efficiency of the small discharge

| Parameter | Exp. (m) | Sim. (m) | Err. (%) |
|-----------|----------|----------|----------|
| H         | 53.60    | 53.29    | -0.58    |
| P (kW)    | 159.00   | 150.16   | -5.56    |
|  (%)     | 62.80    | 66.00    | 5.10     |

3.2. Energy loss of the entire flow passage

The energy loss of each domain of the pump mode under the small discharge condition is shown in Figure 4. The energy loss is transformed into head, using m as its unit. The abscissa stands for the five domains, including draft tube, runner, guide vanes, stay vanes and spiral case, arranged according to the flow direction. The energy loss curve shapes like an inverted parabola along the flow direction. The energy loss inside the draft tube is 3.30m. The head loss inside the runner is the largest among the five domains, with the energy loss of 11.34m. The head loss inside the guide vanes is the secondly greatest among the five domains, with the loss of 10.01m, which is close to the runner’s head loss. The energy loss of the stay vanes is much smaller than the guide vanes, with the loss of 2.45m. The energy loss of the spiral case is the smallest among the five domains, with the energy loss of only 0.65m.

Figure 4. Energy loss of each domain

According to Figure 4, the main loss of the whole flow passage lies in the runner and guide vanes domains. The energy loss inside the draft tube is also considerable. The large energy loss is often induced by the special flow inside the passage, such as vortex and backflow, which will cause large pressure fluctuations and influence the operating stability of the pump-turbine.
3.3. Pressure fluctuations analysis

The relative peak-to-peak amplitudes of different points under the small discharge operating condition are shown as Figure 5. For recording points on 4 planes named dt1, dt2, dt3 and dt4 inside the draft tube, the relative peak-to-peak pressure amplitudes are almost the same, around 18%, as shown in Figure 6(a). For the 8 rotating point inside the runner, it can be seen that the amplitude gradually increases along the flow direction from 17.63% to 62.78% as the inference between the runner and guide vanes getting stronger, shown in Figure 6(b). For the 4 points on the pressure side of the guide vanes, the relative peak-to-peak amplitude at point gv1 is larger than the other 3 points with the number of 58.71%, shown in Figure 6(c). Point gv1 is more close to the casing tongue. For the 4 points on the suction side of the guide vanes, the relative peak-to-peak amplitudes are generally smaller than points on the pressure side, as shown in Figure 6(d). The results also show that the pulse amplitudes inside the stay vanes are generally smaller than the guide vanes and the amplitude at point sv2 stand out of the four points, shown in Figure 6(e). In the spiral case, the pulse amplitudes decrease gradually from sp1 to sp5, shown in Figure 6(f). Point sp5 is near the outlet of the flow passage. Its low peak-to-peak amplitude is on account of the given static pressure condition in the outlet.

![Figure 5. Predicted relative pulse amplitudes at different recording points](image)

The pressure pulse spectra are analysed at various points throughout the entire flow passage. The pressure fluctuations both in the time domain and frequency domain at point dt41, rv1, rv4, rv8, gv2, mv2, sv2 and sp2 are shown in Figure 6 for the small discharge condition. The pressure pulse characteristics at point dt14 and rv1 are similar to each other, with the pressure fluctuating between -6 and 6m. The dominant frequency for dt14 and rv1 is a low frequency of 0.35$f_n$. The first dominant frequency for rv4 is also 0.35$f_n$ and the other dominant frequencies are on the multiple of 0.35$f_n$. For point rv8, the 1.41$f_n$, 0.3$f_n$ and 2.81$f_n$ are included and the amplitudes at these frequencies are larger than dt14 to rv4. In addition, the 20$f_n$ corresponding to the number of the guide vanes is not that obvious compared to the low frequencies, indicating that the guide vanes is not the main factor inducing the large fluctuations under the small discharge condition. At point gv2, the 0.35$f_n$ and 0.88$f_n$ are the dominant frequencies. For point mv2, sv2 and sp2, the first dominant frequency changes to 0.53$f_n$.

As the head loss inside the runner and guide vanes is the largest among the five domains and the low frequency 0.35$f_n$ is included inside the draft tube, runner and guide vanes, flow field at certain flow time is adopted and the internal flow changing characteristics is analysed. For the convenience of
analysis, the 4 flow time, respectively 0.62s, 0.66s, 0.70s and 0.74s, are adopted according to the pressure fluctuation at gv2 and named t1-t4, as shown in Figure 6(e).

![Pressure fluctuations both in the time and frequency domain at chosen points](image)

**Figure 6.** Pressure fluctuations both in the time and frequency domain at chosen points

3.4. Internal flow behavior analysis

The streamlines at different flow time on the mid-span plane of the runner, guide vanes and stay vanes are shown in Figure 7. The internal flow of the small discharge condition is highly disorganized. The vortex and secondary flow are obvious inside the guide vanes and stay vanes, even inside the runner. The appearance of these complex flow patterns can block the flow passage and water can’t flow smoothly. In addition, part of the energy will dissipate in the vortexes. These are the factors contributing to the larger energy loss inside the runner and guide vanes. Compared the streamline at different time, it can also be dawn that the internal flow changes with the flow time. From t1 to t2, the flow in the gv2 nearby region becomes less disorganized. While, from t2 to t4, the flow in the gv2 nearby region becomes more and more disorganized. At t4, there even generates vortex flow in the gv2 nearby region. In the mv2 nearby region, the vortex and secondary flow also experiences the
strengthen and weaken process from t1 to t4. The strengthening and weakening of the vortex and secondary flow result in the low dominant frequency inside the flow passage.

Figure 8 shows the streamline on the meridional plane of the runner and draft tube. It can be found that flow separation occurs with backflow flowing from runner toward draft tube. In addition, in the elbow part of the draft tube, there also occur vortexes. The appearance of backflow and vortexes can block the flow passage and increase the flow resistance. Then the mainstream can’t flow into the runner smoothly. In addition, part of the energy of the runner will dissipate in the backflow and vortexes. Therefore, the head loss inside the draft tube is relatively large although the velocity inside it is low. Compared the streamline at different time, it can also be dawn that the flow inside the draft tube changes with the flow time. From t1 to t2, the vortex on the left side migrates downwards. The two vortexes on the right side merge together. From t2 to t3, the vortex on the left side migrates upwards. The vortex on the right side splits into two parts. From t3 to t4, the vortex flow on the left side strengthens and the vortex region enlarges. The vortex on the left side weakens and the vortex region shrinks. The excursion, split and merging of the vortex and the weakening and strengthening of the vortex are all related to the low dominant frequency inside the draft tube.

Figure 7. Streamline in the runner, guide vanes and stay vanes on the mid-span plane

Figure 8. Streamline on the meridional plane of runner and draft tube

4. Conclusions
For the small discharge operating pump condition at the optimal opening, unsteady numerical simulation of the entire passage is conducted adopting the SST $k-\omega$ turbulence model. Conclusions drawn from the analysis are as follows.

1) The simulated results match well with experimental results. It is feasible to adopt the numerical simulation to analyze the internal flow of the small discharge condition.

2) The main energy loss is in the draft tube, runner and guide vanes domain. The large head loss inside the draft tube is induced by the backflow from the runner and the vortex inside the draft tube. The large head loss inside the runner is caused by the backflow in the runner inlet region and the
secondary flow in the runner outlet region. The large head loss inside the guide vanes is due to the large number of vortex and secondary flow inside the passage.

3) The strengthening and weakening of the vortex and secondary flow result in the low dominant frequency inside the runner downstream flow passage. The excursion, split and merging of the vortexes and the weakening and strengthening of the vortexes are all related to the low dominant frequency inside the draft tube.

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