Unsteady Flow Analysis of Pump Mode Small Discharge Condition for a Francis Pump-turbine

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Abstract. Unsteady flow phenomena, including vortex flow at runner inlet, helical backflow in the draft tube and numerous vortexes inside the guide vanes, can occur in pump-turbines under off design conditions at pump mode and can impact normal operation of pump-turbines. All of these phenomena cause serious pressure pulsation, which is quite different from cases in normal pump mode. There is also a difference of pressure pulsation frequency and amplitude in different place through the runner. This paper builds a whole flow passage of a model pump-turbine, simulates flow characteristics in runner by CFD technology, analyses pressure pulsation in the runner and explores the origin and mechanism of pressure pulsations. The SST-CC turbulence model is adopted to perform unsteady simulations of the pump-turbine under 0.46Q_BEP small discharge condition at pump mode. Unsteady flow structures are proceeded combined with hydraulic loss and pressure amplitude spectra. The results indicates that there is complicated disordered flow inside the runner under 0.46Q_BEP small discharge condition at pump mode, shows the amplitude and frequency characteristic of pressure pulsations through runner flow passage.

1. Introduction
In recent years, the demand for peaking units is growing with the continuous rise of wind energy, small hydro energy, solar energy, ocean energy and other low-carbon energy sources. Pumped-storage units have abilities like peak shaving, frequency modulation and phase modulation. This system is a safe and effective measure to ensure stable economic operation of power.

There exist some unsteady phenomena in pump turbines under small discharge operating conditions at pump mode, including internal flow instabilities and pressure pulsations. There are numerous researches on running conditions in hump region of pump mode, which is regarded as the most important factor of the pump mode’s instability [1-3]. As the flow rate decreases, pump mode’s operations will become increasingly unstable. Ran Hongjuan performed experiments on pressure pulsations in a pump turbine under 0.84 - 1.0 Q_BEP working conditions at pump mode with monitor points located between the runner and the guide vanes. The results showed that the amplitudes of pressure pulsations are much higher at small discharge conditions [4]. Guo Lei used the CFD technology to elucidate the propagation mechanism of pressure pulsations in a pump turbine at pump mode and indicated that amplitudes and frequencies are not even circumferentially due to the specific stay vane [5]. J Yang carried out an experimental study of a pump turbine in part load conditions and ascertained a second unsteady perturbation and a back flow pattern, which may cause the instability [6]. Zhang Lanjin simulated a
pump-turbine at pump mode by CFD. It was found that there appear vortexes near blades, hub and shroud in the runner and that the intensity of vortexes varies in different areas [7].

In our previous researches, dynamics mechanism of the hump region was explored by numerical simulations of the model pump turbine [8]. Pressure fluctuations in the whole flow passage was studied to some extent [9]. However, the distinction of pressure pulsations in different location of the runner has not been investigated in detail. The specific flow patterns and velocity distribution in the runner have not been analyzed.

This paper performed simulations under $0.46 Q_{BEP}$ small discharge condition at pump mode with the optimal opening. The pressure pulsations were analyzed, combining with internal flow field in the runner, to illustrate the unsteady flow phenomena. Uneven pressure pulsation variations through the runner flow passage were revealed in this research.

2. Simulation model

2.1 Geometry of the model pump-turbine

The whole flow passage of a model pump-turbine was simulated in this paper. The computational flow field is shown in Figure 1. Table 1 is the geometrical parameters of the pump-turbine. The rotating speed of the pump-turbine is 1300 rpm and the designed flow rate is 408 L/s.

![Figure 1. The computational flow field of the model pump-turbine](image)

| Geometrical parameter          | Value |
|-------------------------------|-------|
| Runner diameter $D_r$         | $0.47 m$ |
| Number of runner blades $Z_r$ | 9     |
| Number of guide vanes $Z_g$  | 20    |
| Number of stay vanes $Z_s$   | 20    |
| Optimal opening of guide vanes $A_{opt}$ | 27 mm |

2.2. Mesh generation

Mesh quality has a great impact on simulation results. Improvement of mesh quality will get more accurate results while reducing computation time. The mesh sensitivity analysis was performed and a mesh of 2.8 million cells was adopted in this research. The hexahedral meshes were generated in stay vanes, guide vanes and draft tube domains by Gambit 2.2.30, in which the cell amounts are 0.29 million, 0.34 million and 0.42 million respectively. The hexahedral meshes in runner were produced by TurboGrid 14.5 with 1.2 million cells. The volute domain used tetrahedral meshes in ANSYS Mesh 14.5 with 0.59 million cells.

2.3. Boundary conditions and turbulence model
In this paper, velocity inlet boundary condition was adopted at the draft tube inlet, in order to be consistent with actual situations. Outlet boundary conditions at the volute outlet was set as static pressure for corresponding head. No-slip wall conditions were used in solid boundaries. ANSYS CFX 14.5 is used in unsteady simulations with the SAS-SST turbulence model. The SAS-SST method is based on an improved uRANS formulation and has abilities to adapt the length scale to resolved turbulent flow structures. This method is suitable for solving problems of flow separations and vortexes and could reduce huge amount of computational cost. The selected time step is 1/180 of a rotation period, which means there are 180 steps per revolution. The calculation results were saved in every two steps. A total of 38 runner revolutions were completed and results of the latter 8 revolutions were used for pressure pulsations analysis.

2.4. Monitor points arrangement
For the purpose of pressure fluctuation analysis, monitor points were distributed evenly in the runner to record pressure signals. There were 24 monitor points at pressure side, suction side and mid-passage of 0.5 span in each flow passage. Taking one passage for example, monitor points arrangement is presented in Figure 2.

![Figure 2. Monitor points arrangement in the runner](image)

3. Numerical Results

3.1 Pressure Pulsation Analysis
Unsteady simulations of the pump-turbine were performed under 0.46\textit{Q}_{BEP} small discharge condition at pump mode. In this research, one passage is taken as an example. At pump mode, the water flows from the draft tube to the runner, then passes through double-row cascades and the spiral case. In the runner, the flow direction is from monitor point rv11 to rv18. Different time domains of pressure pulsations at rvM11–rvM18 are presented in Figure 3, which show that pressure signals change periodically with time. Pressure pulsation spectrum is obtained by FFT methods. The comparison of amplitudes and frequencies at rvM11–rvM18 (shown in Figure 4) demonstrates that there are two obvious dominant frequencies less than rotating frequency and indicates that frequency components are more complicated at the outlet. It is preliminarily concluded that there exist some unsteady flow structures in the runner and that the pressure pulsations may be under the influence of flow patterns in vanes and tube.
Figure 3. Time domains of pressure pulsation at rvM11–rvM18
Figure 4. Pressure pulsation spectrum at rvM11–rvM18

Figure 5 shows pressure spectra at suction side, mid-passage and pressure side in the same radius. As seen in Figure 5(a), the first dominant frequencies in rvS11, rvM11 and rvP11 are $0.375f_n$. The amplitude at suction side is higher than the other two points. Differences are noted in both frequency and amplitude at downstream monitor points. From rv12 to rv18, the dominant frequencies at suction side are $0.375f_n$, while the dominant frequencies at pressure side are $0.5f_n$. The dominant frequencies at mid-passage, affected by pressure pulsation at pressure and suction side, are $0.375f_n$ from rvM11 to rvM14 and $0.5f_n$ from rvM15 to rvM18.

Frequencies of $1.125f_n$ appear at upstream monitor points near the tube and disappear gradually downstream. On the contrary, frequencies of $0.875f_n$ are obvious at suction side and mid-passage downstream. The two pressure signal could be propagated from the tube and vanes.

At monitor points rvS18, rvM18 and rvP18, amplitudes of $20f_n$ are prominent, due to the rotor-stator interaction between the runner and guide vanes.
3.2 Inner Flow Structure Analysis
In allusion to the dominant frequency features mentioned above, the flow structures in the tube, runner and double-row cascades are explored in this paper. Figure 6 reveals the inner flow behavior on different span planes of the runner. It is highly visible that the internal flow is quite disorganized. Vortexes appear at different locations of pressure side and suction side. As shown in the streamlines on 0.1 and 0.5 span planes of the runner, there exist vortexes near inlet of the suction side and outlet of the pressure side. The complex flow patterns with low velocity nearby cause blockages and disorder in flow passages, which could be the origin of low-frequency pressure pulsation phenomena in the runner.
Figure 6. Streamline and vector in the different span planes of the runner

Figure 7 shows the streamline on pressure side and suction side of the blade. It is found that a vortex structure exists at inlet of the suction side. It is dumped into the tube under the influence of centrifugal force and cause a back flow in the tube, as shown in Figure 8. The spiral back flow presented at the tube outlet could result in the loss of energy with a strong pressure pulsation.

In the meanwhile, there is a disordered flow pattern at outlet of the pressure side, which is affected by the interaction between runner blades and guide vanes. Figure 9 presents the streamline of vanes at 0.5 span plane. As seen, the flow structure inside double-row cascades is quite disorganized, with a significant damage on a high-speed flow near the inlet of guide vanes. It is demonstrated that the flow field changes over time through a comparison of flow patterns in double-row cascades at different moments. The strengthening and weakening process of vortexes causes low frequency pressure pulsations in vanes, which could propagate to the runner.

Figure 7. Streamline on pressure side and suction side of the blade

Figure 8. Reverse flow patterns inside the tube
4. Conclusions
This paper simulated a model pump-turbine under 0.46Q_{BEP} small discharge condition at pump mode by CFD technology. Pressure pulsation spectra were analyzed through FFT method and some representative dominant frequencies were found inside the runner. The origin and mechanism of pressure pulsations were explored with analysis on unsteady flow structures.

(1) There are vortex structures at both pressure and suction sides of the blade, which cause pressure pulsations with different dominant frequencies lower than the rotating frequency (0.375f_n and 0.5f_n). Amplitudes are higher at monitor points near the vortexes.

(2) Reverse flow patterns exist at outlet of the tube and the inlet of the runner. Pressure pulsations with 1.125f_n frequency are detected at monitor points upstream in the runner.

(3) The strengthening and weakening process of vortexes is observed in double-row cascades. Low frequency (0.875f_n) pressure pulsations appear in vanes and then propagate to the runner. The pressure pulsation with the blade passing frequency (20f_n) is observed at monitor points near vanes, due to the RSI phenomenon.

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