Influence of nonlinear pulsating characteristic on the simulation of the load rejection process in a pump-turbine

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Abstract. During the load rejection process, the nonlinear pulsation characteristic and its influence are significant. However, in most previous simulations, the effects of nonlinear pulsation are not fully considered. In this study, the load rejection process was simulated respectively with two different methods. In the average simulation method, the boundary conditions at the inlet and outlet of the pump-turbine were determined with the average filtered unsteady experimental data. The turbulence flow was simulated through Reynolds-averaged Navier-Stokes (RANS) with the renormalization group (RNG) k-ε model. While, in the pulsating simulation method, the boundary conditions at the inlet and outlet of the pump-turbine were directly determined with the original unsteady experimental data. The turbulence flow was simulated using the large eddy simulation (LES) method. Then, the numerical results obtained from the two methods were compared with the experimental data. Results show that the performances of the pulsating simulation method are generally better than those of the average method. However, large computer resources have to be consumed using the pulsating simulation method. Therefore, the two simulation methods should be selected correctly and flexibly according to the actual requirements in the simulation of the load rejection process.

1. Introduction

In recent years, lots of clean and renewable energies e.g., wind power and solar power etc., are introduced into the power grid, which are intermittent and uncontrollable [1]. The intermittent and uncontrollable power will cause the unbalanced loads and frequency fluctuations in the power grid [2]. Nowadays, the pumped-storage power technology is an available grid-scale energy storage technology [3]. In order to balance the variation of the load and adjust frequency in the power grid, the pumped storage power stations frequently undergo a series of transient conditions. The load rejection process is one of the most typical transient processes of pump-turbines [4]. For the normal operating conditions in turbine mode of pump-turbines, it will go into the load rejection process after the pump-turbine disconnecting from the power grid or load. In the load rejection process, the pump-turbine frequently transforms operation modes among the turbine runaway mode, the turbine braking mode and the reverse pump mode. Moreover, the operation of pumped-storage power stations is very dangerous and the flow in pump-turbines is rather complicated in the load rejection process. Many researchers have studied this transient process with different methods. However, up to now, it is still not very clear. It is necessary

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and valuable to accurately simulate the transient flow in pump-turbines with three-dimensional (3-D) methods for researching the transient characteristics of the load rejection process.

In order to accurately simulate the load rejection process, relevant researchers have carried out lots of explorations. Reasonable boundary conditions and turbulence solutions are two key issues which influence the numerical accuracy of the load rejection process. In the early stage, the constant boundary conditions were established at the inlet and outlet of hydraulic turbines [5]. However, the boundary conditions are transient in the actual operation. In order to determine reasonable transient boundary conditions, the pipeline systems upstream and downstream were also included in the 3-D computational domains [6]. However, in the full 3-D simulation methods, the effects of hydraulic acoustic in pipelines were ignored. In addition, the computational effort is too large to fast simulate the transition processes with available computer resources. Later, one-dimensional (1-D) method of characteristic (MOC) was used to determine the transient boundary conditions at inlet and outlet of hydraulic turbines [7]. Based on the 1-D MOC, a 1-D and 3-D (1-D-3-D) coupling method has also been proposed to determine the transient boundary conditions [8]. However, all the numerical results of 1-D method are average. So, neither nor the 1-D MOC and 1-D-3-D method could accurately simulate the intense nonlinear pulsation. Recently, some transient processes were simulated using the real boundary conditions based on the experiments [9,10]. The intense nonlinear pulsation could be accurately predicted in a certain degree.

The turbulence solution is another important issue which influences the numerical accuracy of the load rejection process. At first, the simple $k$-$ε$ turbulence model was adopted to predict the turbulence flow in hydraulic turbines during the transient processes [5,6,11]. Later, considering the advantages of shear stress transport (SST) $k$-$ω$ model in simulating the flow near walls, it was used to predict the transient processes of pump-turbines [12]. However, studies reveal that the SST $k$-$ω$ turbulence model was still a linear eddy viscosity model, while the $\overline{v^2} - f$ turbulence model is a nonlinear eddy viscosity model, which is more suitable to accurately simulate the complex nonlinear turbulence flow in transient processes [13]. Further, a more senior scale-adaptive simulation (SAS)-SST turbulence model has been adopted to simulate the transient processes. Researches pointed out the von Karman scale allows the SAS-SST turbulence model to dynamically adjust to resolved structures in the unsteady Reynolds-averaged Navier-Stokes (RANS) simulation. Thus, in the unsteady flow regions, its simulation effect is similar to that of the large eddy simulation (LES). Meanwhile, this turbulence model is able to provide standard RANS capacities in the stable flow regions [14]. In addition, the detached eddy simulation (DES) method has also been used to simulate the transient processes of pump-turbines. The DES method combines the advantages of RANS (for the near-wall regions) and LES (used in the free-stream). It is potential to provide a better accuracy than RANS and use less cost than LES [15].

Through previous summaries on simulations of the transient process, it could be found that nonlinear pulsating characteristics of boundary conditions and turbulence model obviously influence the numerical accuracy of the load rejection process in pump-turbines in a certain degree. However, few systematic studies have been carried out about the influence of nonlinear pulsation on the simulation of the load rejection process in pump-turbines.

In this paper, firstly, the load rejection process of a pump-turbine was simulated respectively with the average method and the pulsating method. In the average method, the boundary conditions at the inlet and outlet of the pump-turbine were unsteady but filtered in average, the turbulence flow was simulated using the RANS method with the renormalization group (RNG) $k$-$ε$ turbulence model. While, in the pulsating method, the boundary conditions were unsteady and pulsating, the turbulence flow was predicted with the LES method. Then, numerical results with two different methods were compared systematically and in detail to analyze the influence of nonlinear pulsation on the simulation of the load rejection process in a pump-turbine.

2. Computational Models and Boundary Conditions

2.1. Computational domain and mesh generation
In this study, the computational domain is the whole flow passage of the pump-turbine, which includes the spiral casing, 20 stay vanes, 20 guide vanes, the runner with 9 blades, draft-tube and the runner clearance between the runner and the stationary components as shown in figure 1. The pump-turbine has 20 guide/stay vanes and 9 runner blades. The diameters of the runner inlet and outlet are respectively 524mm and 274mm. Its rated rotational speed is 375r/min, rated head is 48.3m, and rated discharge is 0.12m$^3$/s. The hexahedron grids were generated in all components except the guide vane. The hybrid grids (wedge and hexahedron elements) were generated in the guide vane as shown in figure 1. The number of the nodes, grid quality in each component and maximum value of $y^+$ on the solid wall in boundary layers are listed in Table 1.

![Figure 1. Computational domain and grids.](image)

**Table 1.** Nodes distribution and grids quality in each component.

| component                   | Spiral-casing | Stay vane | Guide vane | Runner | Runner clearance | Draft-tube |
|-----------------------------|---------------|-----------|------------|--------|-----------------|------------|
| Nodes number ($\times 10^6$) | 1.14          | 1.50      | 1.67       | 1.76   | 1.10            | 1.20       |
| Maximum $y^+$               | 132           | 8         | 30         | 5      | 10              | 83         |
| Quality                     | 0.27          | 0.43      | 0.45       | 0.38   | 0.72            | 0.66       |

2.2. **Boundary conditions and numerical solutions**

In the average method, the original experimental unsteady pressure signals at the pump-turbine inlet and outlet were filtered using the weighted-adjacent-average method. The obtained filtered average pressure signals were adopted to determine the boundary conditions at the pump-turbine inlet and outlet as shown in figure 2. While, in the pulsating method, the original experimental unsteady pressure signals were directly used to determine the boundary conditions at the pump-turbine inlet and outlet as shown in figure 2. On the other hand, the simple RNG $k$-$\varepsilon$ turbulence model which has a linear eddy viscosity model was adopted to simulate the turbulence flow in the average method. In order to more accurately simulate the nonlinear pulsating characteristics, the LES method was applied to simulate the turbulence flow in the pulsating method.
Figure 2. Unsteady pressure boundary conditions at the inlet and outlet of the pump-turbine: (a) Original and filtered average pressure experimental data at the spiral-casing inlet (b) Original and filtered average pressure experimental data at the draft-tube outlet.

In addition, in these two simulation methods, the dynamic mesh technology was both applied to simulate the closing process of the guide vanes according to the closing law of the guide vanes as shown in figure 3. Meanwhile, the sliding mesh method was both adopted to simulate the speed-changing rotation of the runner with a user-defined function (UDF) according to the equation (1) of angular momentum.

\[
M = J \frac{d\omega}{dt}
\]

where \( M \) is the resultant torque on the rotor; \( J \) is the inertia of the rotor; \( \omega \) is the angular speed of the rotor, \( t \) is the time. The time step size was both 0.0017 s for the two unsteady simulations. The runner rotates 3.825° in a time step at the initial rated speed condition. The loop iterations were both no more than 25 in per time-step. In this study, the initial computation fields of the two unsteady simulations were both the converged steady numerical results of the critical condition point at which the pump-turbine just enters into the load rejection process.

Figure 3. Closing law of the guide vanes during the load rejection process.

3. Results and Analyses

3.1. Comparisons between the simulation results and experimental data

Rotational speed is an important parameter and evaluation index in the load rejection process of a pump-turbine. The normalized rotational speed is defined as equation (2).

\[
n_r = \frac{n}{n_0} \times 100\%
\]
where \( n_t \) is the normalized rotational speed, \( n \) is the rotational speed, \( n_0 \) is the rotational speed at the initial instant of the load rejection process. The simulated rotational speeds were compared with the experimental tests as shown in figure 4. On the overall trend, numerical results from the two methods basically consist with the experiments. The deviations between the two methods and experiments are both less than 5% except in the region \( E_3 \). In the region \( E_3 \), the deviation between experiments and the numerical results of the average method is larger and more than 5%. Whereas, there is a good agreement between the experiments and the numerical results of the pulsating method in this region. It indicates that the pulsating method is more suitable to simulate the reverse pump process than the average method in this study. In addition, the numerical results of the average method are systematically lower than the experiments. Considering the neglected resisting torque on the rotor, it is obvious that the lower systematic deviations are not reasonable. While, almost in entire simulation process, there always remains a good agreement between the experiments and the numerical results of the pulsating method except in the region \( E_3 \). Actually, the local higher deviations between the experiments and the numerical results of the pulsating method are also closely related to the neglected resisting torque on the rotor. The region \( E_3 \) is close to the runaway condition point. Close to the runaway condition point, the hydraulic torque on the runner is almost zero and smaller than that at other all conditions. Meanwhile, the rotational speed of the rotor is also close to maximum runaway speed and larger than at that other all conditions. Therefore, in the region \( E_3 \), the simulated rotational speeds of the pulsating method are slightly higher than the experiments. At the same time, the instant at which the rotational speed is the maximum runaway speed is later than that of experimental test. Hence, except in the region \( E_3 \) which are caused by the measurement error, all numerical results are reasonable, predictable and expected. From the point of rotational speed analyses, it is clear that the numerical results of the pulsating method are more accurate. The above analyses also indicate that it is acceptable to ignore the smaller and uncertain resisting torque during the simulation of the load rejection process.

![Figure 4. Comparison of rotational speeds between simulations and experiments.](image)

The hydraulic pressure signals are more suitable to reflect the nonlinear pulsation characteristics of a pump-turbine during the load rejection process. In this study, the pressure signals at the end of the spiral casing (SP), at the inlet of the draft-tube (DT), in the stay vanes (SV) and in the vaneless space (VL) were selected to analyse the nonlinear pulsation characteristics of the pump-turbine during the load rejection process as shown in figure 5. Firstly, for the monitor points SP and SV far from the runner, the pressure signals obtained through two methods are both basically consistent with the experiments. However, for the monitor points VL and DT close to the runner, the deviations of the pressure signals between the experiments and the average simulation method are larger than those between the experiments and the pulsating simulation method. It is because, closer to the runner, the effect of rotor-stator interaction (RSI) is more significant. Besides, near the runner, especially in the vaneless space and at the draft-tube, the unsteady unstable flows are quite complex. The pressure pulsations are closely related to the RSI and the periodic motion of the unstable flow. So, it is more difficult to accurately simulate the pressure pulsations at the monitor points VL and DT than those at the monitor points SP and SV. The comparison results indicate that the pulsating simulation method is more suitable to
accurately simulate the intense nonlinear pressure pulsations caused by the RSI and the complex unsteady unstable flows. From the figures 3 and 5, at the initial stage of the load rejection process, the larger deviations of the pressure signals between the experiments and the average simulation method illustrate that the faster closing process of the guide vanes could not be accurately simulated through the average simulation method. Furthermore, during the time period about from 6s to 9s, the performances of the two simulation methods are predicted not very well. During this time period, the pump-turbine mainly operates in the brake mode and reverse pump mode. According to the previous studies [4,14,16], it is quite possibly related to the intense instantaneous impact of the reverse back-flow in the reverse pump mode or the reverse water hummer and the S-shape instability in the brake mode. The detailed flow evidences of this explanation will be given in the next research.

Figure 5. Comparison of pressure signals between simulations and experiments at different monitor points: (a) At the end of the spiral-casing (SP) (b) In the stay vane (SV) (c) In the vaneless space (VL) (d) At the inlet of the draft-tube (DT).

3.2. Comparisons between the results from two different numerical methods

In order to further illustrate the differences between the numerical results obtained from these two different methods, some other parameters obtained from these two different simulation method were also compared each other as shown in figure 6. For the discharge, head and hydraulic torque on the runner, the overall trends of the three parameters obtained from two methods are basically consistent. However, the parameters curves obtained from the pulsating simulation method show more intense nonlinear pulsation characteristics and higher pulsation amplitude than those of the average simulation method. It indicates that the average simulation method could not accurately simulate the detailed nonlinear pulsation during the load rejection process. It is worth noting that there exist very large differences between the hydraulic thrust obtained from two different simulation methods. On the one hand, the pulsation frequency and amplitudes of the hydraulic thrust on the runner obtained from the pulsating simulation method are both higher than those of the average simulation method. On the other hand, the magnitudes of the hydraulic thrust on the runner obtained from the pulsating simulation method are also both larger than those of the average simulation method. It shows again that the pulsating simulation method has much better performance than the average method in predicting the nonlinear pulsation characteristics during the load rejection process.
Figure 6. Comparison between some parameters obtained from the two different simulation methods: (a) Discharge ($Q$) (b) Head ($H$) (c) Hydraulic torque on the runner ($T$) (d) Axial hydraulic thrust on the runner ($F_z$) (e) Radial hydraulic thrusts on the runner ($F_x$ and $F_y$).

4. Conclusions

In this study, the load rejection process of a pump-turbine was simulated respectively using the average method and the pulsating method. The rotational speed and some pressure signals obtained from the two simulation methods were respectively compared with those of experiments. Further, some other parameters obtained from the two different methods were also compared each other. Finally, some conclusions were obtained as follows:

In the load rejection process, the nonlinear pulsation characteristic of pump-turbines and its influences are quite significant and should not be neglected. The pulsating simulation method could more accurately simulate the load rejection process than the average method. It has a better performance than the average method in predicting the nonlinear pulsation characteristics.

The average simulation method could only effectively predict the overall trends of some parameters during the load rejection process. It has a weak performance in predicting the nonlinear pulsation characteristic during the load rejection process. Especially, the average method could not accurately predict the nonlinear pulsations caused by the complex unsteady unstable flow in the brake mode and reverse pump mode. Meanwhile, it could also not effectively predict the faster closing process of the guide vanes.
The numerical results of the pulsating method are generally more accurate than those of the average method. However, it will consume too much computer resources to be adopted widely in engineering nowadays. By contrast, the average method could effectively approximately predict most parameters and flow without the great limitation of the computer resources. Therefore, the two different simulation methods should be selected correctly and flexibly according to the actual requirements during the simulation of the load rejection process.

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