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Mechanism of high amplitude low frequency fluctuations in a pump-turbine in pump mode

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ABSTRACT

Pumped storage technology has become the most important energy storage technology in industry today. The instabilities of pump-turbines as key parts of pumped storage power plants have become critical issues in development of pump storage technology. High-amplitude pressure fluctuations are one of such instabilities. In this study, unsteady numerical simulations were carried out in the pump mode of a pump-turbine at using the shear stress transport (SST) k-ω turbulence model. Performance characteristics agree well with experimental data. Unsteady characteristics of typical operating points were obtained. The results show that there are high-amplitude pressure fluctuations at the partial operating points. The frequency characteristics and the propagation law of an at-large partial operating point ($0.74Q_{BEP}$) were determined using time and frequency domain analysis methods (bispectrum, coherence) in combination with the flow field. The analysis results reveal that high-amplitude pressure fluctuations at point $0.74Q_{BEP}$ result from the rotation of Dean Vortices in the draft tube. Due to the rotation of the Dean Vortices, the blocking intensity of the two regions in the guide vane and the stay vane passages are periodically changed at a frequency $0.58f_n$, which results in a shock phenomenon in the guide/stay vanes.

1. Introduction

Hydropower is the most abundant renewable energy resource in the world and has become the important part in power systems [1]. The most well-developed method currently available to store energy from a power system is pumped storage, which is a key part of hydropower. This method has been recognized as the most ideal method for large-scale peak shaving and power storage in power system [2–4]. Pumped-storage power plants obtain dramatic increase due to massive investment of the wind energy, solar energy, and nuclear energy into power systems.

With the rapid development of pumped-storage power plants, pump-turbines, which are the key components of pumped-storage power plants, trend to higher heads, larger capacities, and higher specific speeds [5]. Hydraulic instability problems of pump-turbines have become more prominent, such as hump characteristic and S-shaped characteristic [6,7]. These instability problem are usually related with high amplitude frequency fluctuations. Pressure fluctuation may induce mechanical vibrations, which lead to mechanical failures in some cases [8]. These high-amplitude pressure fluctuations always occur in turbine and pump modes and usually come from rotor–stator interaction [9], rotating stall [10,11], and motion of vortices [6]. These conditions significantly limit the safe and stable operation range of the pump-turbines and are therefore extremely harmful to the whole unit. It is rather important to study the generation mechanism and the propagation law of pressure fluctuations.

In recent years, a significant number of researchers have studied the pressure fluctuation in pump-turbines through experimental and numerical methods. Li et al. [12] studied the influence of the guide vane oscillating process on pressure fluctuations in vaneless space of a pump-turbine in turbine mode. Guo et al. [13] investigated the pressure fluctuation in a pump-turbine operating in pump mode under a low-head condition, and revealed the propagation law in the off-design, low-head condition. Ran et al. [14] performed an experimental study of the pressure fluctuation in a

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pump turbine in pump mode at large partial flow conditions, and believed that 0.2 times the runner rotational frequency in the flow passage is induced by rotating stall in the guide vanes. Sun et al. [15] predicted the pressure fluctuation in a prototype pump-turbine in pump mode through numerical simulation using the SST k-ω turbulence model, and obtained a propagation law for the blade passing frequency and its harmonic frequencies. Li et al. [16,17] conducted a comparison of pressure fluctuations in a pump-turbine in pump mode between a large eddy simulation (LES) and unsteady Reynolds-averaged Navier–Stokes (URANS). They concluded that LES could obtain better results if the number of nodes is sufficient. Zhang et al. [18] carried out experimental study of load variation on pressure fluctuations in a prototype reversible pump-turbine. Zhao et al. [19] performed the numerical simulation on the non-axisymmetric flow characteristic and analysed the generation of low frequency ($0.375f_n$) at pump off-design condition of a pump-turbine.

In a summary, the above studies reveal that, in turbine mode, high amplitude pressure fluctuations mainly come from rotor-stator interaction at normal conditions. At part load condition, they are mainly from vortex rope in the draft tube. At extremely off-design conditions, they originate from back flow vortices in the draft tube, motion of the vortex in the runner and guide/stay vanes, as well as rotating stall in vaneless space. In pump mode, the pump-turbines operates similar to pumps, high amplitude pressure fluctuations mainly come from motion of the vortex (rotating stall, separation vortices, leakage vortices) and rotor-stator interaction. Furthermore, the amplitude of pressure fluctuations at part load are dozen times of that at normal condition [20–24]. Part load operations are becoming more frequent with the introduction of renewable energy such as wind energy and solar energy [25]. The instabilities at large partial operating points are limited to the safe and stable range of the pump-turbines. How to restrict the instabilities has become a key issue in the development of pump-turbines.

Recently, the generation mechanism of rotating stall has become a major focus for an increasing number of researchers. Dynamic pressure measurements and a high-speed flow visualization of injected air bubbles were carried out by Yang et al. [26] to characterize the unsteady phenomena in a low specific speed pump-turbine operating in pump mode. An analysis of the frequency and time-frequency domains reveals that a rotating stall structure occurs in the diffuser when the pump-turbine operates at partial load. Pacot et al. [10] carried out a numerical simulation using the LES approach to predict the rotating stall of a pump-turbine operating in pump mode, and obtained results that were in excellent agreement with the experimental data. Cavazzini et al. [27] characterized the rotating stall phenomena by applying a frequency and time-frequency analyses to a pump-turbine in turbine mode.

Although the studies above have obtained interesting information for pressure fluctuations, there are still many steps to solve instability problems and significantly enlarge the safe and effective operating ranges of pump-turbines. Higher head, larger capacity, and higher specific speed pump-turbines are needed to be investigated. Advanced signal processing methods are needed to determine the characteristic frequency. More detailed analysis is required to obtain the frequency propagation in the entire flow passages of pump-turbines.

In addition, the authors found that not all high-amplitude low frequency fluctuations come from rotating stall. The goal of the present study is to ensure the source of the high-amplitude low-frequency fluctuation in order to lower the level of the pressure fluctuations at large partial discharge conditions. Hence, the authors chose the largest capacity pump-turbine installed in China and tried to carry out unsteady simulations at different operating conditions based on the experimental validation. To obtain the more accuracy results, the number of the mesh was increased over ten million. To determine the dominant characteristic frequency, the analysis method of bispectrum was novelty adopted. To identify the relation of the characteristic frequencies in each part, the analysis method of coherence was innovatively used. Combining with time and frequency analyses, through the variation of discharge at the different passages, velocity field, as well as pressure flied, the generation mechanism of the nonlinear characteristic frequencies in every part was analysed. The high-amplitude low frequency was confirmed from an interesting shock phenomenon at large partial discharge operating condition ($0.74Q_{BEP}$).

2. Numerical model and schemes

2.1. Pump-turbine model

The specific speed of the pump-turbine is 30.7 (m, m^3/s) and it is calculated using Eq. (1). It is a low-speed, high-head, and large-capacity pump-turbine. The various parameters of the pump-turbine are listed in Table 1. The entire computational domain is shown as Fig. 1. A commercial software program, ANSYS ICEM 14.0, is implemented to generate the grid for each part using a structured grid. The entire computational domain is divided into four parts: the draft tube, runner, guide/stay vanes, and spiral casing. The average y^+ at the wall is less than 1.5. Detailed information regarding the mesh is given in Table 2.

$$n_q = \frac{n\sqrt{Q}}{H^{3/4}} \tag{1}$$

2.2. Numerical schemes and boundary conditions

The finite volume method is chosen to solve the incompressible Unsteady Reynolds-averaged Navier-Stokes (URANS) equations. The two-equation turbulence model (SST k-ω) is adopted to close the URANS equations. The high-resolution scheme is chosen for all the terms. In addition, the time step corresponding to 1° of runner revolution and a convergence criterion, $RMS_{max} < 10^{-6}$, with 10 internal coefficient loops are selected. Moreover, the spiral casing-stay vane interface is General Grid Interface (GGI), while the rotor-stator interfaces (guide-vane runner and runner-draft tube) based on the GGI are set as Transient Rotor-Stator. The accuracy of the numerical model and schemes has been validated in previous published papers [28–30].

The static pressure (0 Pa) at the draft tube inlet is set and the discharge at the spiral casing outlet is specified according to the experimental data. No-slip wall conditions are set at the wall.

Table 1
Parameters of the pump-turbine.

Parameter	Symbol	Value
Specific speed	n_q	30.7
Runner blade number	Z_r	9
Guide vane number	Z_g	20
Stay vane number	Z_s	20
Runner outlet	D_{1m}	0.524 m
Runner inlet	D_{2m}	0.274 m
Guide vane height	B_0	0.04577 m
Guide vane distribution diameter	D_0	0.61046 m
Rotation speed	n	1000 rev/min