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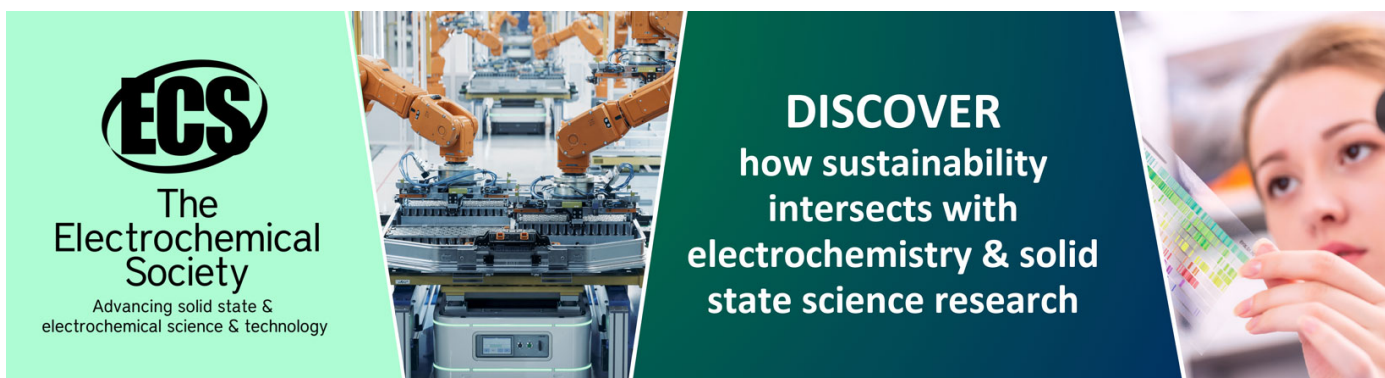
Steady and transient performance of a model pump-turbine at off-design conditions: turbine mode

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Steady and transient performance of a model pump-turbine at off-design conditions: turbine mode

J H Hu¹, J D Yang¹ and W Zeng²

¹State Key Laboratory of Water Resources and Hydropower Engineering Science, Wuhan University, Wuhan 430072, China

²School of Civil, Environmental and Mining Engineering, University of Adelaide, SA 5005, Australia

jin honghu@whu.edu.cn

Abstract. Model Experiments including steady-state and hydraulic transient experiments are essential to study the hydraulic characteristics of pump turbines in turbine mode. They can be used as verification of CFD calculations. Moreover, some special and complicated transient processes can only be examined through experimental research. This paper reports a series of steady-state and hydraulic transient experiments performed on an open-loop pumped storage test rig. First of all, steady-state experimental results were presented, including the measured characteristic curves by adjusting the sphere valve, characteristics of pressure pulsations, and the hill-chart. Subsequently, three kinds of transient experiments were conducted under a fixed guide-vane opening (GVO) to dynamically investigate characteristic curves. The first kind of dynamic tests was conducted with continuous operation of the sphere valve while the unit rotated at the rated value by connecting to the power grid. This was followed by the load rejection experiments. In the third type of transient experiments, the recirculation pump delivering water to the upstream and downstream tanks was shut down when the unit was in designed operation and synchronized to the power grid. As the water level in the upstream tank continued to decrease, the unit experienced a special dynamic process. In this process, the flow discharge and the water head were oscillating periodically while the pump-turbine was switching back and forth from the turbine mode to the reverse rotation pump mode with a constant rotational speed. This transient test was completely different from the others and was regarded as an individual method to investigate the characteristic curves dynamically. Finally, characteristics curves under a constant GVO obtained by these tests were compared and analyzed.

1. Introduction

Pumped storage units can operate in both pump and turbine modes. This flexibility allows them to change their operating conditions to meet the needs of the power grid. However, the units are required to frequently experience transient operations. Taking a pumped-storage power station in China as an example, the number of startups and shutdowns per year of the four units was beyond 100 [1]. For a pump-turbine, issues during transient process are often related to the S-shaped characteristic curve, which not only affects the startup, but also affects the water hammer pressure during emergency load rejection [2]. This significant difference distinguishes pump-turbines from conventional Francis turbines and has drawn much attention from researchers in the last decade.

Since the pump turbine is a compromise between a pump and a turbine, it is naturally assumed that the hydrodynamic phenomena in S-shaped characteristic region were inherited from the pump,



such as rotating stalls. Vesely et al. [3] measured the pressure field of a medium-head model pump-turbine in both pump and turbine conditions. Cavitation nucleus of the rotating stall under pump conditions was visualized. They also stated that the stall cell behaved differently in pump mode and turbine mode and that the radial force produced by the stall was high. By bubble injection and flow visualization, Hasmatuchi et al. [4] experimentally confirmed the existence of the rotating stall in a pump-turbine. The relative frequency of the stall was obtained through pressure measurement. Using the PIV technique, Ciocan et al. [5] experimentally observed that, under off-design conditions, the non-uniform distribution of velocity and local pressure at the outlet of the guide vane resulted in an increase in the pressure within flow channel of guide vanes. Guggenberger et al. [6] confirmed that the flow field distribution between the rotor and guide vanes was not uniform in turbine brake mode. He also detected a stall cell by pressure-phase analysis and that the normalized frequency was about 0.65. Once this phenomenon had been experimentally confirmed, numerical simulations could be used to investigate the flow pattern inside the runner in detail, such as the research by Widmer et al. [7], Xia et al. [8], Jacquet et al. [9], Cavazzini et al. [10] and Genter et al. [11]. Not only the instantaneous flow and pressure fields in the runner were presented in these papers, but also the turbulence analysis, and characteristics of pressure pulsations. The effect of unsteady flow patterns on the dynamic stress of the runner was also evaluated.

The paper reports steady and transient experiments on an open-loop pumped storage test rig. Through the adjustment of a sphere valve, characteristic curves from the nominal point to the zero flow point were obtained by point-to-point measurements. Pressure pulsation in the vaneless space between the guide vanes and the runner were also analyzed. Subsequently, three transient experiments, including continuous adjustment of the sphere valve, load rejection with guide vanes stalling, and a dynamic oscillation experiment at a constant rotational speed, were performed under a fixed GVO. Finally, the dynamic characteristics obtained by different methods were compared. Differences in the measured characteristic curves among these tests were analyzed.

2. Experimental Setup

The schematic of the open-loop pumped storage test rig at Wuhan University is shown in figure 1 (a) with the 3D view of the pipe system shown in figure 1 (b)

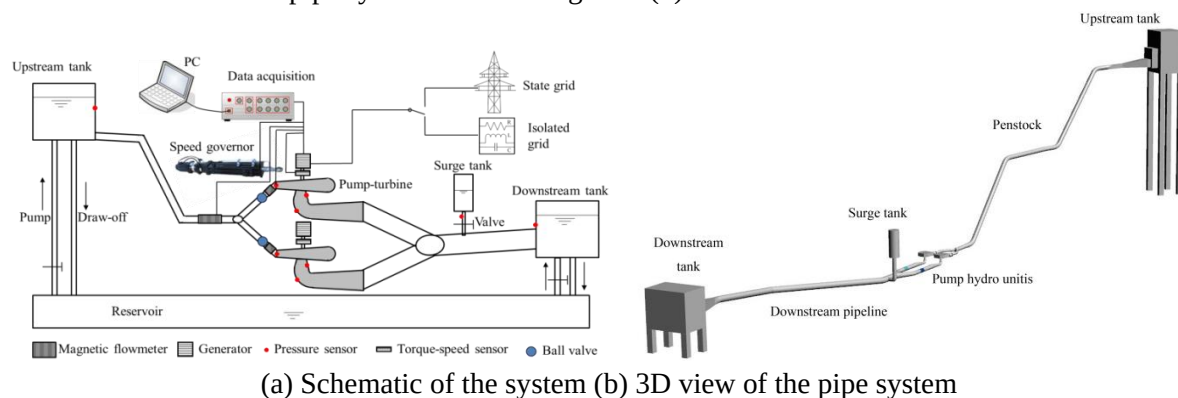


Figure 1. Open-loop pumped storage test rig.

Recirculation pump delivered water from an underground reservoir to the upstream and downstream tanks. The water levels in both tanks are maintained constant through overflow. The whole system was designed according to IEC 60193 [12] and the principle that no cavitation would occur during transient experiments.

In steady state operations, the unit rotated steadily as connected with the power grid. In Table 1, geometrical parameters and rated operating parameters of the model unit are presented. On the test rig, there are two model runners, whose characteristics are different in turbine braking mode but similar in designed operating conditions. The runner with gentle S-shaped characteristics was selected for the experimental studies in this paper. The other unit was not in operation during experiments. Table 2 displays the specifications of the sensors used to measure mechanical and flow parameters. Eleven sensors as shown in figure 2 were installed on each pump turbine with three listed in Table 2 used in

this paper. For the measurement of pressures in the spiral case and the draft tube cone, Micro MPM absolute pressure sensors calibrated with Fluke 754 were used. The PCB dynamic pressure sensors, calibrated by the manufacturer, were used to measure the pressure pulsations in the vaneless region.

The control of the spherical valve is of great importance in the tests. In the steady tests, the hydraulic loss of the test rig was changed by adjusting the spherical valve, which was previously proposed by Doerfler et al. [13] to stabilize the prototype pump turbine at no-load conditions. The physical mechanism of the operation can be explained by changing the system's characteristic curve [14] and increasing the resistance of the system [15]. This method has been employed to experimentally explore the steady-state performance of pump turbine in turbine brake mode by Guggenberger et al. [6]. For dynamic tests, the spherical valve was closed or opened continuously to obtain the dynamic performance curves at off-designed conditions.

Table 1. Model unit parameters.

Specific Speed N_{QE} ($\text{m}^{0.75} \text{s}^{-1}$)	Inlet diameter D_1 (mm)	Outlet diameter D_2 (mm)	Guide vane height (mm)	Blade number Z_b	Guide vane number Z_g	Rated rotational speed n (rpm)	Rated head H (m)	Rated flow discharge Q (L/s)	Full rotational inertia (N.m)
37.91	280	146.34	24.44	9	20	1000	10.54	49.1	66.4

Table 2. Specifications of the different sensors.

Sensor type	Positions	Measurement range	Accuracy degree (%)	Uncertainty(%)	Signal
Pressure sensor (Micro MPM 480)	1# Spiral case inlet (point 1 in figure 2)	0–200 (kPa)	0.25	0.109	4–20 mA
	1# Draft tube inlet (point 8 in figure 2)	–20–50 (kPa)		0.083	
Pressure sensor PCB 112A	1# Vaneless space (point 6 in figure 2)	/		/	
Flow meter (OPTIFLUX 2000)	1# branch pipe	–70–70 (L/s)		/	
Torque sensor (HLD09)	Between 1# turbine and 1# generator	–100–100 (N.m)	0.2		4–20 mA /Frequency

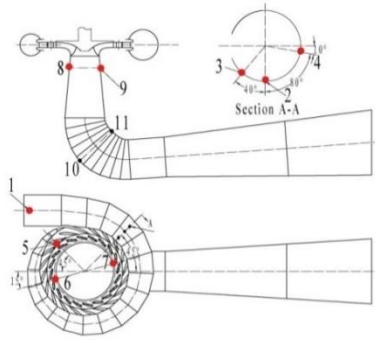


Figure 2. Location of pressure sensors.

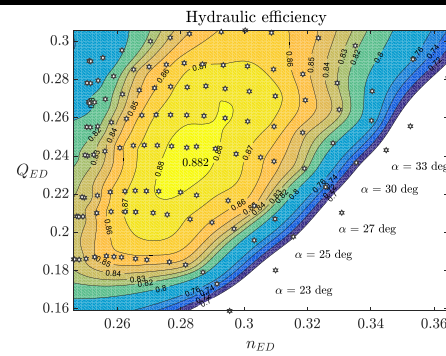


Figure 3. Hill-chart of the pump-turbine.

3. Results and discussions

In both steady and transient tests, the sampling rate was set to 1000Hz. For steady experiments, each operating point was sampled over 8 s to ensure reasonable averaged results. In the transient experiments, the Savitzky-Golay FIR filter [16] was applied to the measured flow discharge, torque and pressures in the spiral case and draft tube. Characteristic curves of the pump-turbine were described with the dimensionless parameters n_{ED} , Q_{ED} and T_{ED} defined as

$$n_{ED} = \frac{nD}{E^{0.5}} \quad Q_{ED} = \frac{Q}{D^2 E^{0.5}} \quad T_{ED} = \frac{T}{\rho D^3 E^{1.5}} \quad (1)$$

where D is the reference diameter of the pump-turbine (D_2 in this paper), E is the specific hydraulic energy, ρ is the water density and T is the torque.

3.1. Steady performance

At the beginning, the unit was started to the speed no load at the nominal rotational speed 1000 rpm. Then, the GVO was adjusted to the target value. Under the rated head, the head loss was increased by closing the sphere valve, whereas the flow discharge was reduced. At the same time, the operating condition of the unit would gradually reach the part load mode, turbine brake mode, and finally the zero flow condition. Figure 3 plots the hill-chart of the pump-turbine with results of steady tests using sphere valve adjustments. As an example, the characteristic curve with $GVO = 23 \text{ deg}$ was selected. The distribution of the amplitude of pressure fluctuation in the vaneless space is plotted using the scatter diagram in figure 4, with the standard deviation of pressure fluctuation defined in equation (2).

$$\tilde{C}_p = \frac{1}{\rho E} \sqrt{\frac{1}{N} \sum_{i=1}^N (p_i - \bar{p})^2} \quad (2)$$

As shown in figure 5, ten operating points from the designed condition to zero flow condition were selected to study the pressure pulsations in the vaneless space. The pressure pulsation coefficient is defined as equation (3).

$$C_p = \frac{p - \bar{p}}{\rho E} \quad (3)$$

The results showed that the amplitude of the pressure pulsation with the blade passing frequency $9f_0$ and $18f_0$ gradually increased as the unit operated into the turbine brake mode. In addition, a sub-synchronous frequency with a gradually increasing amplitude could be observed, especially at the operating point (No. 9), where the normalized frequency was 0.5883. However, the amplitude of this pressure pulsation reduced at zero flow condition (No. 10).

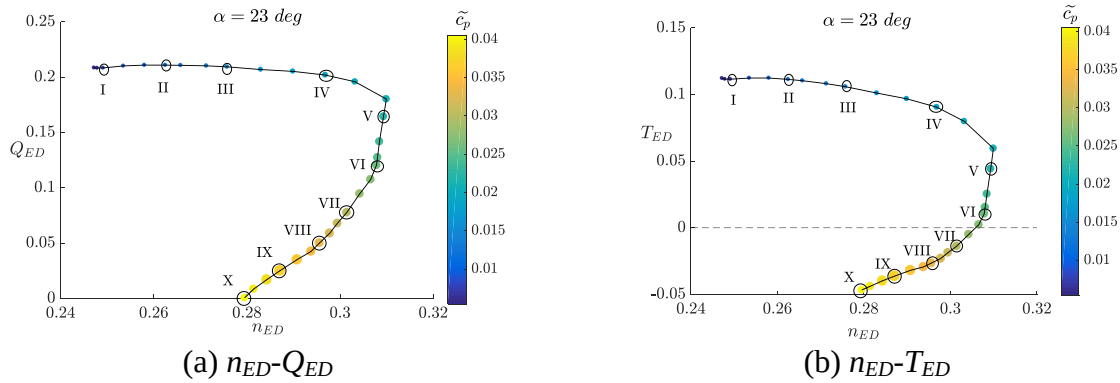


Figure 4. Characteristic curves and the distribution of the amplitude of pressure fluctuation in the vaneless space ($GVO = 23 \text{ deg}$).

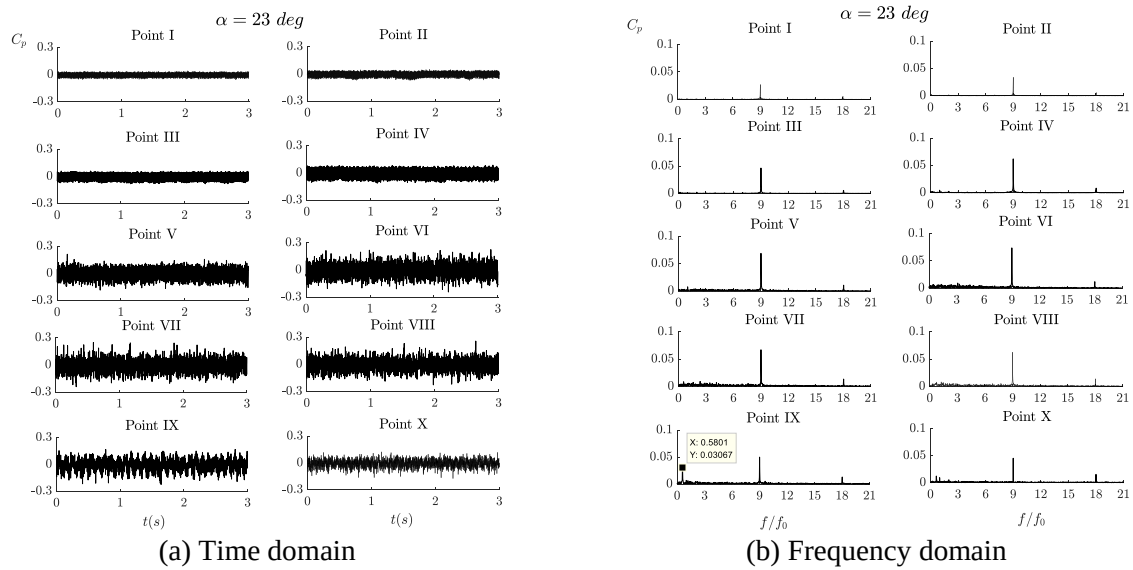


Figure 5. Pressure pulsations in the vaneless space.

3.2. Transient performance

3.2.1 Continuous closing/opening of the spherical valve. Hydraulic transient tests could be divided into spontaneous tests and manually controllable tests. Typical spontaneous transient tests include emergency load rejection in turbine mode and power failure in pump mode. In these scenarios, the unit is driven by an unbalanced torque, and transient parameters of the unit would change nonlinearly. By contrast, transient parameters were often linearly adjusted in the manually controllable tests to investigate the dynamic performance of a pump turbine. For instance, the linear adjustment of the rotational speed is often adopted to measure the characteristic curves of hydraulic turbines. However, on this open-loop test rig, the unit was connected to the grid, so the rotational speed could not be linearly changed. Therefore, the sphere valve was utilized to change the flow discharge of the unit with a constant GVO.

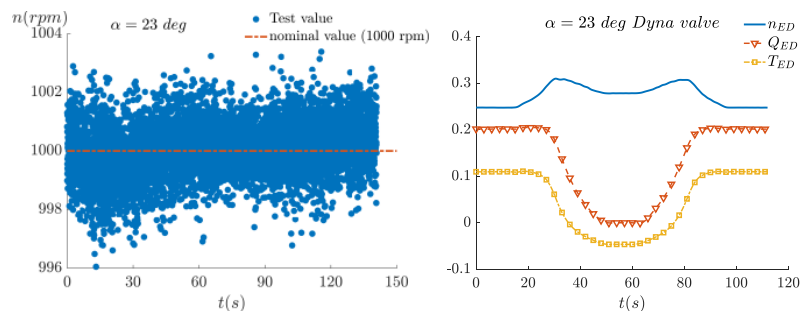
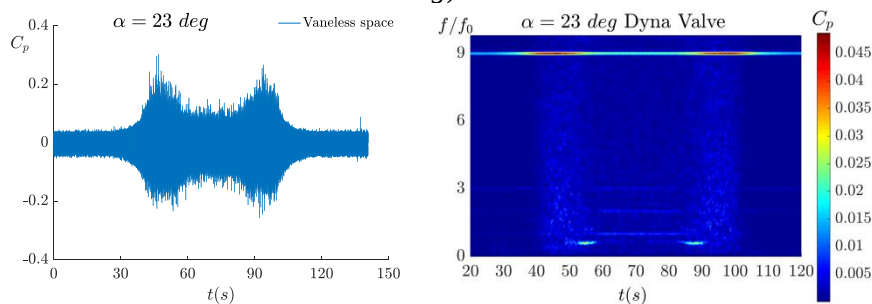


Figure 6. Variation of transient parameters by continuous adjustment of spherical valve (GVO = 23 deg).



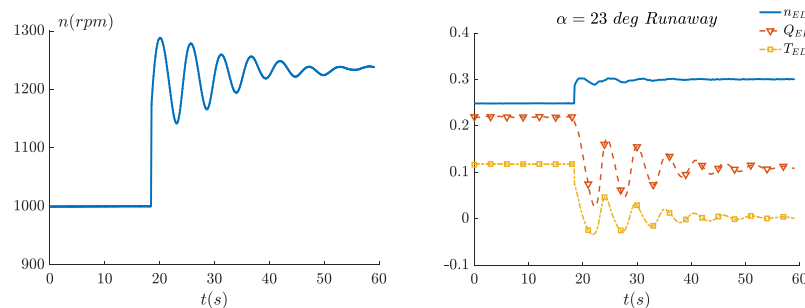
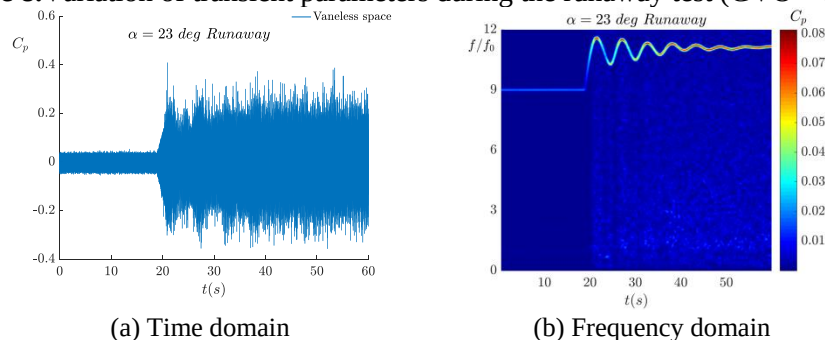
(a) Time domain

(b) Frequency domain

Figure 7. Pressure pulsations in the vaneless space by continuous adjustment of the spherical valve (GVO = 23 deg).

A dynamic experiment to obtain performance curves of the pump-turbine with GVO = 23 deg was carried out by continuously regulating the spherical valve. The spherical valve was linearly closed from the fully open state to fully closure within $T_s = 30$ s. After remaining closed for 20 s, it was reopened linearly in 30 s. The variations of the rotational speed and the dimensionless turbine parameters are shown in figure 6. Characteristics of the pressure pulsations in the vaneless region are shown in figure 7. It can be observed that the frequency of the dominant component of the pressure pulsations is always the blade passing frequency $9f_0$. However, the amplitude of this rotor-stator interaction induced component changes significantly as the operation condition varies. Especially when the pump-turbine approaches turbine brake mode, the amplitude increases sharply. In addition, another low frequency component with a large amplitude in the turbine brake mode could be observed, which was in consistent with the results obtained in steady state measurements.

3.2.2 Runaway Test. The runaway test which means emergency load rejection with guide vanes stalled was conducted to measure pump turbine dynamic characteristic curves. After several cycles of fluctuation, the rotational speed and the dimensionless turbine parameters as shown in figure 8 finally stabilize at the some specific levels with the turbine operates at the no-load condition. In terms of the pressure pulsations in the vaneless space as shown in figure 9, the amplitude of the dominant component (frequency = $9f_0$) maximizes when the unit reaches the zero torque condition ($dn/dt = 0$). However, the intensity of low frequency part in figure 9 was much less distinctive compared with that in figure 7. The main reason is that the operating point moves too fast and stays in the turbine brake mode in a short period, which might be not enough to form low frequency hydrodynamic instabilities.

**Figure 8.** Variation of transient parameters during the runaway test (GVO = 23 deg).

(a) Time domain

(b) Frequency domain

Figure 9. Pressure pulsations in the vaneless space during runaway test (GVO = 23 deg).

3.2.3 Constant-speed oscillation (CSO). A strategy to measure the characteristic curves using a closed-loop test rig is to adjust the water head of the hydraulic turbine. However, the water head can be only slightly changed for open-loop test rigs if the upstream and downstream tanks are filled by the recirculation pump to remain constant water levels. To achieve a large

change of water head, the recirculation pump was shut down while the unit still generated and was connected with the grid. As can be seen from figure 10 and figure 11, the operating condition of the unit switches back and forth from turbine working mode to the reverse rotation pump mode and across the turbine break mode. But the rotational speed almost kept constant as shown in figure 10 since the unit is connected with the power grid.

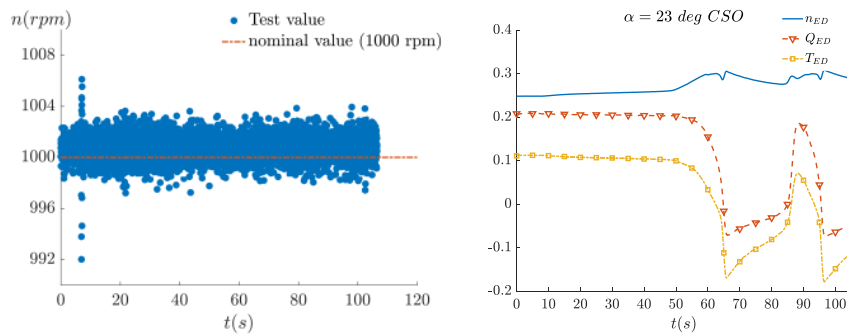


Figure 10. Variation of flow parameters during CSO test (GVO = 23 deg).

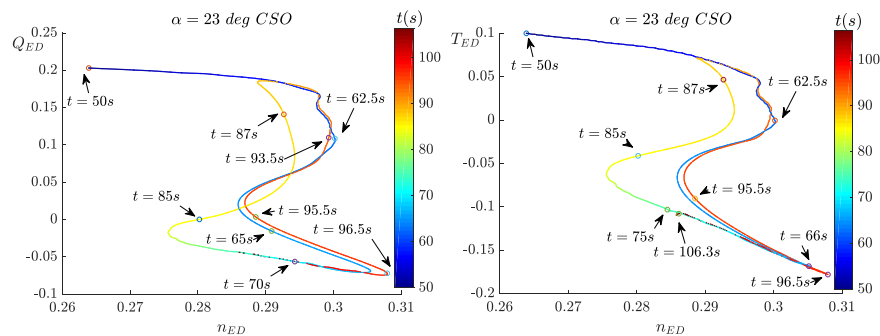


Figure 11. Dynamic trajectories during CSO (GVO=23 deg).

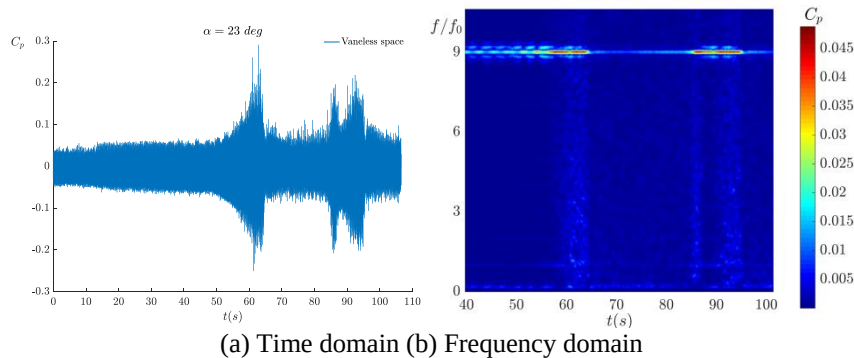


Figure 12. Pressure pulsations in the vaneless space during CSO test (GVO=23 deg).

According to figure 11, a significant difference between the forward path and backward path of the n_{ED} - Q_{ED} (T_{ED}) curves can be observed. This can be attributed to the significant dynamic characteristics as transient parameters vary sharply. Figure 12 shows the pressure pulsations in the vaneless space in the time domain and the corresponding spectrum in the frequency domain. The dominant component (frequency = $9f_0$) is significant in the deep part load conditions and the turbine brake mode. In addition, it can be found that a strong modulation, which is caused by the instability of the excitation system, occurs with the frequency equal to $9 \pm 0.17 f_0$.

Possible reasons for the oscillation during the CSO test are discussed. As the pressure in the upstream tank decreases, the net head across the pump-turbine spontaneously declines. Due to the unstable characteristics in the S-shaped region, the pump turbine is forced to operate unstably under a

low water head. While the friction of the system was not enough to stabilize the unit since the sphere valve is fully open, thus a periodic interaction of energy between water and the grid is formed.

3.3. Comparison of characteristic curves obtained by different methods

In figure 13, characteristic curves obtained by different methods with a same GVO are compared. A distinctive difference is observed that the dimensionless turbine parameter n_{ED} of the characteristic curves in the S-shaped region is relatively larger when measuring the characteristic curves by operating the sphere valve. The factor resulting in the difference can be explored using the analysis of the dimensionless Navier-Stokes Equation (equation (4)) [17].

$$n_{ED} \frac{dv}{dt} = \frac{1}{Q_{ED}^2} \nabla p + \frac{1}{Reynolds} \nabla^2 v \quad (4)$$

For two operating points with the same Q_{ED} , the one with a much lower net head should have a lower rotational speed, which results in a lower Reynolds number based on the outlet peripheral speed. Then, the actual n_{ED} would be enlarged according to Eq.4. As the water head across the turbine would be greatly reduced by closing the sphere valve, the n_{ED} would be relatively larger according to the analysis.

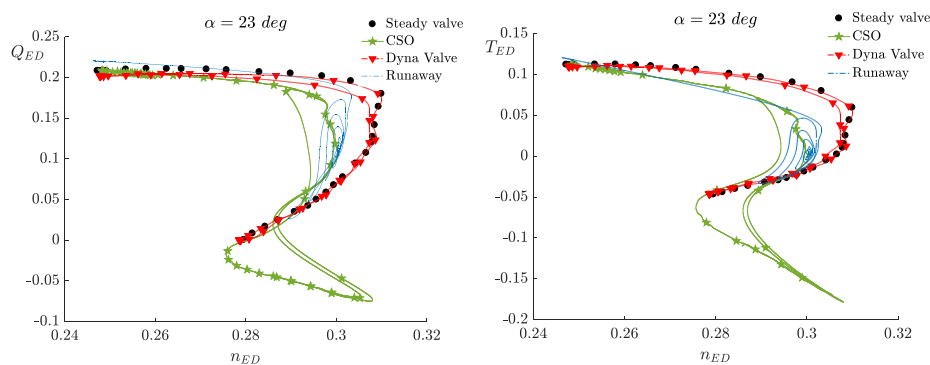


Figure 13. n_{ED} - Q_{ED} and n_{ED} - T_{ED} characteristic curves obtained by different methods.

4. Conclusions

Steady-state and hydraulic transient experiments have been conducted on an open-loop pumped storage test rig. Through the steady-state hydraulic performance experiments with adjusting the sphere valve point-by-point, the steady-state characteristic curve of the model pump turbine can be obtained. Similar characteristic curves with some small loops can be obtained through dynamically closing or opening the sphere valve. Compared with these measured characteristic curves, the curves measured by conducting runaway and CSO transient scenarios are distinctively different. Large dynamic loops can be observed in these two scenarios due to the dynamic characteristics of the pump turbine during transient scenarios. Further stability analysis and CFD simulation are going to be conducted to further understand the dynamic performance of pump-turbines.

Acknowledgments

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