Effects of Pump-turbine S-shaped Characteristics on Transient Behaviours: Experimental Investigation

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Abstract. A pumped storage stations model was set up and introduced in the previous paper. In the model station, the S-shaped characteristic curves was measured at the load rejection condition with the guide vanes stalling. Load rejection tests where guide-vane closed linearly were performed to validate the effect of the S-shaped characteristics on hydraulic transients. Load rejection experiments with different guide vane closing schemes were also performed to determine a suitable scheme considering the S-shaped characteristics. The condition of one pump turbine rejecting its load after another defined as one-after-another (OAA) load rejection was performed to validate the possibility of S-induced extreme draft tube pressure.

1 Introduction

The S-shaped characteristics of the pump turbine are the most critical factor determining the transient characteristics of pumped storage stations. Static tests of pump turbines [1,2] and computational fluid dynamics numerical methods [3,4] can reveal the complex hydraulic phenomena of the internal runner when a pump turbine is operating in the S region, such as stationary vortices, unsteady vortex, and the rotating stall that results in the blockage effect; all these phenomena are fundamental causes for the inversion of the S-shaped characteristic curve. Moreover, the S-shaped characteristics also lead to unstable runaway or no-load operations [5-8], as well as water-hammer pressure surge after load rejections.

Due to the S-shaped characteristics, the dynamic processes of a pumped storage station is complicated: 1) The unstable pump turbine S-shaped characteristics make it difficult to measure the S-shaped curves, which can, therefore, be obtained by the transient method [9,10]; 2) pump turbines often show an unstable runaway and no-load operations, which are close to the form of S-shaped curves [5-8]; 3) in the load rejection process, the water-hammer pressure is also related to the S-shaped curves; 4) in some extreme conditions, such as OAA load rejections, numerical simulations have shown that the water-hammer pressure, especially in the draft tube, is fairly significant [11,12]; 5) to control the water-hammer pressure and runaway speed during the load rejection process, multi-phase closing schemes can be used to guarantee the safety of a pumped storage station [13]. Therefore, based on the transient issues described above, the relevant verification experiments were conducted to clarify the effects of S-shaped characteristics on hydraulic transients.
2 Transient Experiments

The characteristic curves of pump-turbines are usually represented using three dimensionless parameters. To standardize the equation derivation, three relative dimensionless parameters are defined as follows:

\[ n_{ed} = \frac{n_{ed}}{n_{ED,r}} = \sqrt{\frac{H}{n}} \frac{n}{\sqrt{H}} \quad Q_{ed} = \frac{Q_{ED}}{Q_{ED,r}} = \sqrt{\frac{H}{Q}} \frac{Q}{\sqrt{H}} \quad M_{ed} = \frac{M_{ED}}{M_{ED,r}} = \frac{H}{M} \frac{M}{H} \]  

(1)

In which, \( n \) is the rotational speed, \( H \) is the water head, \( Q \) is the flow, \( M \) is the torque, the subscript ED means the dimensionless parameters and \( r \) means the rated parameters.

2.1 Load rejection tests with guide vanes stalling

2.1.1 Dynamic measurement of S-shaped curves. In the turbine braking zone, the pump turbine characteristic curves tend to have a positive slope, as shown in Fig. 1. The region meets the following condition:

\[ \frac{dQ_{ed}}{dn_{ed}} > 0 \]  

(2)

It will turn into an unstable region, and the operating point is difficult to stabilize within the area. When the point reaches the higher singular point \( S \), it always moves directly to the lower singular point \( S' \) and skips the inflection region [8].

![Figure 1. Pump turbine S-shaped curve](image)

Figure 1. Pump turbine S-shaped curve

The transient method can be used to measure the characteristic curve in this area, i.e., measure the dynamic flow, speed, torque, and head of the pump turbine after load rejection with guide vane stalling. The flow inertia in the pump turbine (from the spiral case inlet to the draft tube outlet) of the general test rigs has a large proportion compared with the piping system, and therefore, there will be a significant dynamic loop in the S-shaped region [9]. Nielsen studied the dynamic characteristics of the Francis turbine and found the following correction equation for the dynamic head \( H_{dyn} \)

\[ H_{dyn} = H - I \frac{dQ}{dt} \]  

(3)

Here, \( I \) is the flow inertia between the defined unit inlet and the outlet.

In this pumped storage system model, the proportion of the flow inertia in the pipe system is much greater than that in the turbine. Moreover, the draft tube inlet is defined as the unit outlet to reduce the flow inertia in the pump turbine. Thus, the transient method to measure the dynamic S-shaped characteristic will lead to a significant reduction of the dynamic effects. As each pump turbine rejected its load, and we measured the spiral case pressure (SCP), draft tube pressure (DTP) after filtering, torque, flow, and rotating speed. Thus, the dynamic characteristic curves in the S-shaped region can be plotted based on these results, as shown with the dotted line in Fig. 2. After the modification by Eq. (3), the dynamic loop can be substantially eliminated, shown in Fig. 2 with the solid line.

2.1.2 Runaway stability. The runaway stability of the pump turbine is directly related to the slopes of the runaway point on the characteristic curves. Theoretical derivations illustrate that the stability criteria under the assumption of a rigid water column is [7]
In which, $T_a$ is the time scale of machinery, $T_w$ is the time scale of flow and $K_f$ is the friction loss coefficient. When excluding the system friction, the runaway stability condition is

$$\frac{dM_{cd}}{dn_{cd}} > \frac{2T_a}{Q_{cd}T_w} \quad \text{or} \quad \frac{dM_{cd}}{dn_{cd}} < 0$$

(5)

The slope $dM_{cd}/dn_{cd}$ of the BQ pump turbine at the runaway point at the rated opening is smaller than that of the XJ pump turbine. Therefore, when the piping system is without a surge chamber, XJ displays a convergence (Fig. 3 (a)), while BQ exhibits a uniform amplitude oscillation (Fig. 3 (b)).

As the timescale of water inertia $T_w$ increases, for example, when the valve between the surge chamber and the pipe is turned off, the speed variations of BQ with a surge tank connected and disconnected to the pipe are shown in Figs. 3 (b) and 3 (c), respectively. In such a situation, $2T_a/Q_{cd}T_w$ reduces in Eq. (5), and the stability criterion is more easily satisfied; therefore, the unit gradually stabilizes under a small flow, as shown in Fig. 3 (c).

When the unit rotating inertia is reduced, for example, when 60% of the rotating inertia in BQ is eliminated, the measured speed variation is as shown in Fig. 3 (d). According to Eq. (5), this situation is more likely to meet the stability criterion; therefore, the unit gradually stabilizes under the flow.

![Figure 2. Dynamic loop and modified curve in the S-shaped region](image)
2.2 Load rejection tests of different pump-turbines

In addition to the runaway instability induced by the S-shaped curve, water-hammer pressure in the load rejection process is also closely related to it. The unit flow rapidly changes in the S-shaped region and results in the sharply declined flow in the transition process and a greater water-hammer pressure.

The S-shaped curve in Domain 2 of the XJ unit is outward relative to the BQ unit (Fig. 2), which may improve the unit’s stability under the runaway condition. However, because of the outward characteristic curve at the large guide vane opening, the trajectory of the operating points of XJ may be more inclined in Domain 2 when the guide vanes are closed, resulting in a greater water-hammer pressure.

Load rejection tests for the two units were performed with guide vanes straightly closed in 7 s. The measured $H_s$, $H_d$, $n$, and $Q$ are shown in Fig. 4. The maximum $H_s$ of BQ (14.56 m) is less than that of XJ (14.82 m) by 0.26 m, while the minimum $H_d$ of BQ (1.78 m) is greater than that of XJ (1.57 m) by 0.21 m, and the maximum $n$ of BQ (1268 rpm) is less than that of XJ (1290 rpm) by 22 rpm. Therefore, XJ is more dangerous during transients, wherein the major difference in the transient pressures $H_s$ and $H_d$ between BQ and XJ is from Domain 2, the grey region in Fig. 4. According to the trajectories shown in Fig. 5, one can see that the trajectory of XJ in Domain 2, especially in the braking zone, is more inclined, which is adverse for the water-hammer pressure. The difference in the maximum rotating speed comes from the turbine area, especially owing to the larger $n_e$ at the runaway point.

![Figure 3](image-url)  
**Figure 3.** Measured rotating speed under runaway operations (1) $Q_0 = 49.1$ L/s; (2) $Q_0 = 37$ L/s; (3) $Q_0 = 24$ L/s

![Figure 4](image-url)  
**Figure 4.** Measured transient parameters, BQ vs XJ

![Figure 5](image-url)  
**Figure 5.** Trajectory, BQ vs XJ
2.3 Load-rejection tests with different guide vane closing schemes

After the station being construction, the most effective and economical engineering measure to reduce the transient pressures and speed is improving the guide vane closing scheme. Detail theoretical analyses of guide vane closing scheme considering the pump turbine S-shaped characteristics were conducted in the paper [13]. Here, three types of closing schemes (shown in Fig.6) based on the theory were compared through experiments.

For the BQ unit, the variations in $H_s$, $H_d$, and $n$ under a straight closing scheme at 7 s are shown in Fig. 7 with solid lines. Under this scheme, because the guide vanes close in the braking region, the corresponding water-hammer pressure increases significantly (maximum $H_s$ 14.56 m, minimum $H_d$ 1.78 m). In addition, the closed guide vanes in the turbine working region reduces the input torque, thus the runaway speed is rather low (maximum $n$ 1268 rpm).

Compared with the straight closing scheme, the two-phase closing scheme (closing the guide vanes after stalling) can effectively control the water-hammer pressure, in which the guide vanes start to close when the operating point reaches the lower singular point S' (Fig. 1). The corresponding experimental results (the dashed lines in Fig. 7) validate the statements (maximum $H_s$ 13.84 m, minimum $H_d$ 2.12 m). However, because the guide vanes are not closed in the turbine working region, the torque decreases slowly and the runaway speed increases significantly (maximum $n$ 1290 rpm).

In comparison with the straight closing scheme, the three-phase closing scheme (guide vanes close first then stall (or open), and then close again) can not only reduce the water-hammer pressure but also the runaway speed. The first turning point of the three-phase closing scheme is when the operating point reaches the higher singular point S, and the second is when the operating point reaches the lower singular point S'. The corresponding experiments (the dashed lines in Fig. 7) show the advantages of the three-phase closing scheme in controlling both water-hammer pressure and runaway speed (maximum $H_s$ 13.99 m, minimum $H_d$ 2.03 m, maximum $n$ 1253 rpm).

![Figure 6 Guide-vane closing schemes](image)

![Figure 7. Variations in the measured transient parameters with different guide vane closing schemes](image)
2.4 OAA load rejection tests

To reduce project investment, pumped storage stations generally use the layout of multiple machines sharing the same main pipes, which leads to hydraulic connections between pump turbines during the transition process. When one pump turbine rejects its load, it would lead to overload of another unit, which in turn will reject the load (defined as OAA load rejection). Owing to the positive slope of the pump turbine characteristic curve in Domain 2 (fig. 1), hydraulic connections between the units under OAA conditions may lead to a higher water-hammer pressure, directly threatening the safety of the pumped storage station.

However, this possibility of an extreme water-hammer pressure under such conditions can only be determined by numerical analyses [12]. Because of the risks involved with this transient condition, it has never been tested in fields. Thus, model experiments under such extreme conditions were conducted in the pumped storage model. The surge chamber inlet was opened in the experiments, and the XJ pump turbine rejected its load after BQ. The time intervals of the load rejection start times of the two pump turbines are 0, 0.2 ... 2.6 s. The results are shown in Table 1. It can be seen that in the interval of 1.0 s, the draft tube pressure is extremely unfavorable. The variations in $H_s$ and $H_d$ under intervals of 0, 1.0, and 2.0 s are shown in Fig. 8. As can be seen, under OAA load rejection, the water-hammer pressure change is very obvious in Domain 2.

![Figure 8](image)

**Figure 8.** Variations in the measured transient pressures under OAA conditions

| Table 1. Extrema of spiral case pressure and draft tube pressure |
|---------------------------------------------------------------|
| $T$ (s) | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 | 2.2 | 2.4 | 2.6 |
| $H_s$ | 14.81 | 14.85 | 14.87 | 14.84 | 14.80 | 14.79 | 14.70 | 14.64 | 14.59 | 14.44 | 14.38 | 14.25 | 14.10 | 14.07 |
| $H_d$ | 1.75 | 1.69 | 1.65 | 1.66 | 1.60 | 1.56 | 1.61 | 1.60 | 1.61 | 1.60 | 1.67 | 1.71 | 1.83 | 1.89 |

3 Conclusions

In the present study, a series of transient experiments were conducted on the pumped storage model station. The following conclusions can be obtained:

1) A transient method can be used to measure the S-shaped characteristic curve, especially the unstable region with a large opening. The experiments also confirm that the runaway stability is associated with the curve slopes at runaway points as well as the system inertia.
2) Pump turbine S-shaped characteristics are fairly relevant with the water-hammer pressure in the load rejection process.
3) Comparison of the transition processes under different guide vane closing schemes revealed that the three-phase closing scheme adaptable to the S-shaped characteristics can simultaneously control different transient parameters.
4) The extreme draft tube pressure under the OAA load rejection conditions was obtained on the model, and the experiments showed the root cause was the pump turbine S-shaped characteristics.

In addition to the experiments presented in this paper, the model can also be employed in other experimental studies of transition processes, such as power failure in the pump mode, startup in the turbine mode and power frequency regulation, etc. In particular the pressure pulsation during different transient scenarios can be tested and studied on such a model.

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