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Extreme water-hammer pressure during one-after-another load shedding in pumped-storage stations

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ABSTRACT

The intermittent and unpredictable wind and solar power leads to the frequent transient processing of pumped-storage stations, increasing the probability of load shedding. When one turbine sheds its load, the other turbines in the same hydraulic unit become overloaded and may shed their loads, which is referred to as a “one-after-another (OAA)” load-shedding process. An extremely high water-hammer pressure (WHP), namely, high spiral case pressure (SCP) or low draft tube pressure (DTP), may arise in this case, directly threatening the safety of the PSS. The objective of this study was to theoretically determine the hydraulic connections between the turbines and reveal the mechanism of the rapid rise in the WHP under the OAA load-shedding conditions. Theoretical derivations inferred that the drastic pressure changes in a trail shedding turbine (TST) are caused by the hydraulic connection with the lead shedding turbine (LST) in the S region. Furthermore, numerical simulations and model experiments were performed for the OAA load-shedding process, which confirmed the validity of the theoretical analysis. Finally, an analysis was conducted on the distribution of the water inertia in the upstream and downstream branch pipes, and engineering measures were proposed to guarantee the safe operation of PSS systems.

1. Introduction

Pumped-storage stations play a pivotal role in the frequency control and power regulation of electric power systems as the penetration of intermittent and unpredictable energies such as the wind and solar power increases [1]. However, to fulfill their regulatory function, pump-turbines constantly undergo transitional processes between starting and stopping, thus increasing the probability of load shedding.

To reduce project costs, an increasing number PSS installations have employed multi-turbines sharing the same main pipes. However, if one of the turbines sheds its load, the resultant water-hammer pressure (WHP) increases the water head of the other turbines sharing the same hydraulic unit through the branching pipes. The increased water head could cause these turbines to overload, triggering a load-shedding process and thus generating an extremely high WHP.

During the dynamic process after load shedding, the operating point of the pump-turbine often goes though the S region. In this region, complex hydraulic phenomena occur inside the runner, such as stationary vortices and rotating stalls [2,3], which could lead to the blockage of the runner channels and a rapid rise in the WHP due to the sudden drop in flow. It has been revealed through numerical calculations that the reverse bending of the S curve affects the maximum value of the WHP [4]. Moreover, Zeng et al. [5] and Olimstad et al. [6] reported the association between the S-curve slope and system stability. In the study by Zeng et al. [7], theoretical derivations were used to obtain the correlation between the S-curve slope and the WHP during the transition process. All these studies indicated that the S characteristics have a considerable effect on the transition process and even a more pronounced effect on the OAA load-shedding process.

The other influential factor for the WHP under OAA load-shedding process is the inter-turbine hydraulic connections. Chen et al. [8] developed a numerical model for the hydraulic interferences between PSS turbines and used numerical calculations to analyze the effects of the load-shedding process of one turbine on the other turbines that are operating normally. Their results provided the conditions for the occurrence of the OAA load-shedding process. As the inter-turbine hydraulic connections

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are determined by the pipe arrangement, Zhang et al. [9] investigated the effects of pipe layout on the OAA load-shedding process. For an annularly arranged PSS, Zhang [10] conducted a sensitivity analysis through numerical calculations and obtained the most dangerous shedding interval in the OAA load-shedding process. Furthermore, using numerical calculations, Fang and Jiri [11] reported the occurrence of an extremely high WHP during the OAA load-shedding process of a specific PSS and an extremely low DTP. The results of the numerical calculations also indicated that the WHP magnitude could be controlled by using a misaligned guide vane (MGV) [12], combining the optimization of the guide-vane closing patterns [7], and closing the inlet valves [13]. By numerical calculations, Zeng [14] concluded that the flow exchange between the turbines was the cause of the WHP increase in the TST during an OAA load-shedding process. Flow exchange was also observed under the runaway conditions of two parallel pump-turbines [15]. Although numerical calculations have revealed the important characteristics of the WHP rise during the OAA load-shedding process, they cannot explain the physical mechanism of this phenomenon. Therefore, there is a need to conduct further theoretical derivations to uncover the mechanism and devise more focused engineering measures. In addition, simulations of the extreme WHP during the OAA load-shedding process have never been confirmed by field tests or experiments, thus the verification of the occurrence and the mechanism of the extreme pressure using an experimental model is of significance.

2. Mathematical model

To ensure an adjustable design, a small value for the maximum SCP, and a large value for the minimum DTP are required. Therefore, the water head, which is approximately equal to the difference between the SCP and the DTP, was used as the control standard. Moreover, to simplify the theoretical derivation, a simplified layout without surge tanks was used as the basis for the case study. As discussed in the reference [14], the effect of the surge tank on the WHP is consistent in different OAA load-shedding conditions, so excluding the surge tank is acceptable. The arrangement of the pipes is shown in Fig. 1, and the parameters of the pipes and turbines are shown in Table 1.

In order to facilitate the theoretical derivation, the following non-dimensional relative parameters were defined by dividing the numerical values by the respective rated values at the working condition point:

$$n_{ed} = \frac{n_{ED}}{n_{ED,r}} = \frac{\sqrt{H_r}}{n_r}\frac{n}{\sqrt{H}} \quad Q_{ed} = \frac{Q_{ED}}{Q_{ED,r}} = \frac{\sqrt{H_r}}{Q_r}\frac{Q}{\sqrt{H}} \quad M_{ed} = \frac{M_{ED}}{M_{ED,r}} = \frac{H_r}{M_r}\frac{M}{H} \tag{1}$$

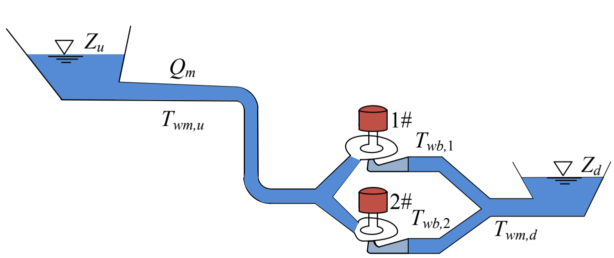

Fig. 1. Theoretical analysis of one-after-another (OAA) load-shedding process.

2.1. Basic equations

The rotational acceleration equation for a turbine array is

$$T_a\frac{dx_i}{dt} = m_i - m_g \tag{2}$$

where $T_a = \frac{GD^2n_r^2}{365P_r}$, $i = 1$ and $i = 2$ represent the LST and TST, respectively.

Without considering the friction loss in the pipe system and the water elasticity, the momentum equations for the upstream and downstream main pipes are

$$z_u - h_u = T_{wm,u}\frac{dq_m}{dt} \tag{3}$$

$$h_d - z_d = T_{wm,d}\frac{dq_m}{dt} \tag{4}$$

where $T_w = \frac{LQ_r}{gAH_r}$, $T_{wm,u}$ and $T_{wm,d}$ represent the time scale of water flow of the upstream main pipes and downstream main pipes, respectively. The following expression can be derived by combining Eqs. (3) and (4):

$$1 - (h_u - h_d) = T_{wm}\frac{dq_m}{dt} \tag{5}$$

Without considering the water-head loss, the initial relative value of water head for the power station is $z_u - z_d = 1$.

The momentum equations for the flows inside the branch pipes of the two turbines are given by

$$(h_u - h_d) - h_i = T_{wb,i}\frac{dq_i}{dt} \tag{6}$$

Combining Eqs. (5) and (6), the water heads of the turbine flows are obtained as

$$\Delta h_i = h_i - 1 = -T_{wb,i}\frac{dq_i}{dt} - T_{wm}\frac{dq_m}{dt} \tag{7}$$

To facilitate the numerical calculation, Eq. (7) is rewritten as

$$\frac{dq_1}{dt} = -\frac{T_{wb,2}}{T_{w,1}T_{w,2} - T_{wm}^2}\Delta h_1 - \frac{T_{wm}}{T_{w,1}T_{w,2} - T_{wm}^2}(\Delta h_1 - \Delta h_2) \tag{8}$$

$$\frac{dq_2}{dt} = -\frac{T_{wb,1}}{T_{w,1}T_{w,2} - T_{wm}^2}\Delta h_2 - \frac{T_{wm}}{T_{w,1}T_{w,2} - T_{wm}^2}(\Delta h_2 - \Delta h_1) \tag{9}$$

When taking the water elasticity into consideration [16,17], the water head of the flow is

$$\Delta h_i = -T_{wb,i}\tanh(T_e s)q_i - T_{wm}\tanh(T_e s)q_m \tag{10}$$

where $T_e = \frac{2L}{c}$, s is the Laplace operator. From Eq. (10), the variation in the turbine water head is determined by the T_w allocation among the main and branch pipes. The effects are completely consistent irrespective of whether the main pipe is located upstream or downstream.

2.2. Mathematical model for inter-turbine hydraulic connection

From Eqs. (7) and (10), it can be inferred that the rate of change of the LST flow, rather than the magnitude, affects the transition process of the TST. Therefore, it was assumed that the LST flow undergoes linear changes within a small interval, i.e., $q_1 = A_1 t$ and