CFD simulation of the pump trip runaway transient process of a pumped-storage power plant with head 700 m

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Abstract. Simulating transient processes of pumped-storage power plants is essential for the design and optimization of pump-turbine and water conveyance system. Because pressure pulsations and runner forces cannot be obtained from the traditional 1D transient simulation methods, 3D CFD simulations are necessary. In this study, the fast runaway transient process, occurring after the pump trip of a pumped-storage plant with a head of 700 m, was simulated by using the CFD method that couples the 1D water conveyance system and 3D turbine. The phenomena, composed of water hammer fluctuations, pressure pulsation transitions, and flow pattern evolutions, were described and analyzed. We found that the flow patterns and pressure pulsations change violently as the working point goes through different regions. Especially, the pressure pulsations and unbalanced radial runner forces are the most severe in the S-shaped characteristic region. The strong impact on blades, large scale recirculating vortices in blade passages, obvious rotating stalls in vane and blade regions, transitional backflows at runner inlet and outlet, and strong circumferential velocity in vaneless space are the main sources. The dynamic working trajectory does not follow the measured static characteristic curve and demonstrates an undamped loop in the S-shaped region. This paper provides some evidence on the relations between flow structure and pressure pulsation features, although the mechanism of dynamic characteristics needs further investigations.

Key Words: Pumped-storage power plant, Pump trip runaway transients, CFD Simulation, Water hammer, Pressure pulsations, Flow patterns

1. Introduction

The runaway transient process after pump trip is dangerous, because huge water hammer, severe pressure pulsations and violent flow transitions happen when the working point slides rapidly through the five modes in the characteristic planes [1]. To prevent serious consequence, this transient phenomenon should be simulated in the design phase. The normally used transient simulation methods can be classified to several types: the one-dimensional (1D) unsteady pressured pipe flow model solved by the method of characteristics (MOC) [2,3], the electric circuit analogic model solved by finite difference schemes [4], and the wave function propagation methods [5,6]. All these methods [2-
6] can simulate water hammer phenomenon but cannot consider the influence of pressure pulsations and flow patterns. To overcome this problem, the 1D water conveyance system and three-dimensional (3D) turbine coupled simulation approaches were proposed in recent years [7-10]. The successful applications of the 1D-3D coupled CFD method in simulating runaway and load rejection transient processes of pumped-storage plants [11,12] opened a door of studying the dynamic characteristics of turbine inner flow in fast transient processes. Zhang [11,12] first simulated the runaway transient process from a turbine working point, and found that the runaway dynamic trajectory forms a loop in the S-shaped region in the four-quadrant characteristic charts of pump-turbine, not following the corresponding static characteristic curve. The mechanism of the loop phenomenon was attributed to the successive features of transient flow patterns, apart from the water inertia in the runner. Xia [13] studied the similar runaway transient process and concluded that the locally distributed backflow vortices at the runner inlet could enhance the local rotor-stator interaction (RSI), and the evolving rotating-stalls (RSs) could induce asymmetrical pressure distribution on runner blades. These two studies are about a model pump-turbine installed in a pumped-storage model test system [14], with corresponding turbine rated head 510 m for the prototype. Yang [15] chose a prototype pumped-storage with turbine rated head 106m as the subject and simulated the runaway process after a pump-trip accident. The evolutions of flow patterns and pressure fluctuations during the working point fast sliding across the four modes were vividly visualized, and the influence mechanism of flow pattern transitions on pressure fluctuation changes was discussed.

In this study, in order to further validate the 1D-3D coupled CFD method and to investigate the evolution laws of flow patterns and pressure pulsations in high-headed pump-turbines, we simulated the runaway transient process after the pump trip of a prototype pumped-storage system with the maximum static pump head 701m. The dynamic processes of water hammer fluctuation in the pipe system, the pressure pulsation transition and the flow pattern evolution in the pump-turbine are demonstrated and analyzed. These may enhance our understating of the phenomenon and give key parameters for the design of the project.

2. Simulation Subject and Methods

2.1 The Pump-Storage System

A high-headed pumped-storage power plant with two 350MW units in one water conveyance system was considered. It was in the design phase, in which layout optimization and turbine selection were conducted. The system includes an upstream reservoir (water level 1384.00m), a headrace tunnel ($L = 914$ m, $D = 6.2$ m), a headrace surge tank ($D = 10$ m), a main penstock ($L = 2018$ m, $D = 5.6$ m), two branched penstocks ($L = 90$ m, $D = 2.7$ m), two pump-turbine units, two tailrace conduits ($L = 170$ m, $D = 4.6$ m), a tailrace surge tank ($D = 10$ m), a main tailrace tunnel ($L = 1315$ m, $D = 6.2$ m), and a downstream reservoir (water level 709.00m), as shown in Figure 1. The setting elevation of the units is 596.00m, and the static head is 656m to 701m. The main parameters of the pump-turbine are listed in Table 1, in which $D_1$ is the turbine inlet diameter, $D_2$ is the turbine outlet diameter, $GD^2$ is the rotational moment of turbine-generator unit, $Z_b$ is the turbine bade number, $n_{gv}$ is the guide-vane number, $n_{sv}$ is the stay-vane number, and $n_t$ is the rated rotational speed.

![Figure 1. Schematic of the computational domain and monitoring points](image)
3.2 The 1D-3D CFD Simulation Method

The pumping condition with two units in operation was set as the initial condition. The runaway transients after two units suddenly tripped were simulated, keeping a constant guide vane opening value. To reduce simulation expense, we only considered one unit, by taking advantage of the symmetric manipulation.

The 1D-3D coupled method proposed by [7] was adopted, in which the diversion tunnel, penstock and downstream conduit were simulated by 1D MOC and the pump-turbine was modelled by commercial CFD software ANSYS FLUENT 15.0. The rotation of the runner was simulated by the sliding mesh model, and the variation of the rotational speed was calculated by integrating the equation of unit moment of momentum [8]. The propagation of pressure waves (water-hammer speed) in the pump-turbine was modelled by defining the water density as a function of pressure [8]. The flow variables between the 1D and 3D sub-domains were exchanged by the partly overlapped coupling method [7,8].

The time-step size in both sub-domains was set as $1 \times 10^{-3}$ s, corresponding to 3 degrees of the runner rotating at the rated speed. Also, the residuals at each time-step were below a typical criterion of $1.0 \times 10^{-4}$. A turbine mesh with 6.0 million grids was finally chosen after a grid dependence analysis, in which fine boundary layers are set up in the regions of stay-vanes, guide-vanes, and runner. The wedge grids were used in the vane diffuser, while the structured hexahedral grids were applied in the other parts, and the number of grid elements was shown in the Table 2. The SAS-SST turbulence model, the SIMPLEC algorithm, and the second order discretization were chosen.

### Table 2. Number of grid elements (million)

|                  | Spiral casing with extended tube | Guide- and stay-vane channels | Runner | Draft-tube with extended tube | Total |
|------------------|----------------------------------|--------------------------------|--------|-------------------------------|-------|
| **Value**        | 1.1                              | 1.5                            | 1.7    | 1.7                           | 6.0   |

### Table 1. Main parameters of the prototype pump-turbine

| Parameter       | Value | Parameter       | Value |
|-----------------|-------|-----------------|-------|
| $D_1$ (m)       | 4.3   | $Z_b$           | 9     |
| $D_2$ (m)       | 1.95  | $n_{gv}$        | 22    |
| $GD^2$ (10^6 kg·m^2) | 4.5   | $n_{sv}$        | 22    |
| $n_s$ (m·Kw)    | 90.2  | $n_t$ (rpm)     | 500   |

3. Results

3.1. Macro variable fluctuations

Figure 2 shows history curves of the main macro variables during the pump-trip runaway process. We know that the overall varying trends of variables are similar to those from 1D MOC simulations, namely the discharge, torque and speed change significantly when the work point goes across different modes. The large scale oscillations in the S-shaped region (composed of the high-speed turbine, turbine-braking and reverse pump modes) show an undamped feature, and the periods of macro variable fluctuations are all about 20 s. These oscillations of working point and variables are common for high-headed pump-turbines, and related to the resistance of water conveyance system and the slope of characteristic curve at the zero torque point [16]. The results of [12,13] also have similar oscillations because their heads are about 510m, however, the results of [15] show fast damping because its head is about 106m. The theoretical analysis has provided the criterion for runaway instability [16], but why the magnitudes or ranges of the undamped oscillations are different is still an issue should be answered.
As the carrier waves in the history curves, the high-frequency pulsations are obvious in the torque, runner force curves, especially when the operating point goes into the pump-braking, high-speed turbine, and turbine-braking modes. These high-frequency pulsations are caused by the turbine flow patterns, as stated by [17]. The unbalanced radial forces are very large in the S-shaped region, with the maximum peak-peak value around $6 \times 10^6$N, indicating very strong asymmetry in flow structures. These unbalanced radial forces will become the source of vibration and should be paid attention to in the design phase.

3.2. Dynamic working point trajectory

The dynamic trajectories of working point are drawn in the four quadrant characteristic charts (Figure 3), in which the unit parameters are defined as $n_{11} = nD_1 / \sqrt{H}$, $Q_{11} = Q / (D_1^2 \sqrt{H})$ and $M_{11} = M / (D_1^3 \sqrt{H})$ (H=E1-E2, where E1 and E2 are the total energy values of spiral-casing inlet and runner outlet, respectively). Connecting Figure 2 with Figure 3, one can know the transitions of turbine working conditions. It is clear that the working point starts from a pumping point in the pump region, then slides into the pump-braking region by cutting across the $Q_{11} = 0$ coordinate, into the turbine region by cutting across the $n_{11} = 0$ coordinate, into the turbine-braking region by cutting across the $M_{11} = 0$ line, into the reverse pump region by cutting across the $Q_{11} = 0$ coordinate, and finally oscillates in between the S-shaped region.

(a) Histories of torque, discharge and speed
(b) Histories of runner forces

Figure 2. Histories of macro variables during the pump-trip runaway process.
In general trend, the dynamic trajectory curves agree with the static characteristic curves of the same guide-vane opening. However, in the saddle-shaped region (low discharge region between the pump and pump-braking region), the S-shaped region, and turbine region, the deviations are obvious. In the saddle-shaped region, the dynamic curve in the \( n_{11} - Q_{11} \) plane is on the right of the static one, and the dynamic curve in the \( n_{11} - M_{11} \) plane is above the static one, indicating the simulated parameters are smaller in speed, larger in head and larger in torque. In the S-shaped region, the dynamic curve in the \( n_{11} - Q_{11} \) plane is on the left of the static one, and the dynamic curve in the \( n_{11} - M_{11} \) plane has the same characteristic, indicating the simulated parameters are smaller in speed and larger in head. In the pump braking and turbine regions, the dynamic curves in both planes are above the static ones, especially the simulated unit torque value is higher than the measured one, indicating the energy dissipation in simulation is smaller than that in measurement. These inconsistences cannot be simply attributed to simulation errors, because many simulations for static working points using the similar simulation models obtained very good agreement between the simulated and measured curves [11,17,18] and the simulated dynamic curve for a low-headed case fits the measured static curve well [15]. The reasons may be classified as follows: (1) the influence of the water inertia in the turbine between the two sections for calculating head [12,23]; (2) the influence of the successive feature of flow patterns [11]; (3) the influence of dynamic hysteresis [19,20]; (4) the errors of simulation model, grids, and transient scheme [21]. The former three may be synthesized and called dynamic effect to express the physical differences of flow characteristics between dynamic and static working conditions. From our previous observations, the dynamic effect is obvious for high-headed pump-turbines [22] but weak for low-headed pump-turbines [15].

In the S-shaped region, the oscillating loops are evident, when the low-pass filtered curves are plotted in the close-up of Figure 3. The loops are on the left of the static curve and also show S shapes. The up going and down going paths are clearly separated. Even if the paths of different cycles do not follow strictly each other, the undamped feature can be seen. These phenomena are related to the instability and dynamic effect in the S-shaped region.

3.3. Piezometric head pulsation transitions

The piezometric head fluctuation histories at the monitoring points in Figure 4 tell us that the pulsation intensity changes with the operating point transition from one mode to the other. The piezometric head pulsations are the high-frequency components that superimposed on the slow fluctuating components, with the former generated by turbine flow patterns and the latter by pipe flow inertia and water hammer.

![Figure 3. Dynamic trajectory curves in the four quadrant characteristic charts](image-url)
The low-pass filtered piezometric head histories show the water hammer characteristics, which agree with the normal results of 1D MOC. After the pump is tripped, the piezometric head at P1 drops fast from 1363 m at $t = 0$ s and to 1097 m at $t = 3.8$ s, before the working point goes into the pump-braking region. After that, the piezometric head begins increase, and reaches its first peak 1653 m at $t = 26.9$ s, when the working point just leaves from the turbine-braking region to the reverse pump region. The around 550m piezometric head fluctuation at P1 before the spiral case and around 200m fluctuation at P4 in the draft-tube are very huge, and should be considered in the design of water conveyance system.

The most violent piezometric head pulsations happen in the S-shaped region, and then the pump-braking mode, in which the flow patterns are more complex and violent. The piezometric head pulsations in the vaneless space (P3) have very large magnitudes, with the maximum peak-peak value in the S-shaped region around 1200m. These huge pulsations are caused by the strong rotor-stator interaction, which is related to [24] vaneless gap size, guide vane opening, guide vane height, runner blade parameters, etc. This strange RSI should be paid attention to in optimizing the turbine. The pulsating magnitudes at the downstream point P4 and upstream points P2 and P1 are around 100m, 200m, and 80m, respectively. These different pulsating magnitudes in the flow stream related locations indicate the transmission and damping features of piezometric head pulsations in the turbine. Also, it should be noted that the simulated pressure at P4 is lower than the cavitation pressure after the working point enters the turbine mode (about from $t = 14.0$ s to $t = 17.0$ s). The multiphase model should be used to more accurately simulate the runaway process after pump-trip in further studies.

3.4 Flow pattern evolutions
3.4.1 In pump mode
In the beginning of the runaway ($t = 0$ s), the pump is working near the optimal pumping condition, therefore, the flows in the draft-tube, runner, and guide and stay-vanes are smooth. When it comes to $t = 2.0$ s, the rotating stalls [25] appear in the vane region (Figure 5(a)) due to the decreased rotational speed and discharge. With the decrease of discharge, the rotating stalls in vanes become very obvious, and blade passage separations and draft-tube backflows become visible. At $t = 5.5$ s when the pump is working in the zero-discharge point, these unstable vortices become strong and dominate the whole flow passages.
3.4.2 In pump-braking mode
When the working point goes into the pump-braking mode, the water flows in the turbine direction and the runner rotates in the pump direction. The water jets from guide-vanes meet the fast coming blades, and forms strong impacts on the pressure sides, which dissipate a large part of energy. The impact makes some water turn left and return to the vaneless space, some water jump over and impact on the next blade, and some water turn right and enter the blade passages, which form large separation vortices around the suction sides of blades (Figure 6(a), \(t = 10.0\) s). Even if the overall discharge is downstream, there are strong backflows in center of the draft-tube (Figure 6(b), \(t = 10.0\) s).

![Figure 6 Flow patterns in the pump-braking mode (\(t = 10.0\) s)](image)

3.4.3 In turbine mode
When the working point goes into the turbine mode, the water flow and runner rotation are all in the turbine directions. Because of relative larger discharge and smaller speed, the flow patterns are similar to those in Figure 6, however, the flow impacts on the blade pressure sides and the separations in blade passages become weaker and the downstream flow near the draft-tube wall becomes stronger.

When the working point is near the optimal turbine condition (about \(t = 20.7\) s), the flow patterns in the whole flow passages are smooth, without obvious vortices. With the increase of speed, the working point comes into the high speed turbine region. Owing to the high speed and the decreased discharge, the separations in the blade passages appear again, with the high velocity gathered near the blade suction sides (Figure 7(a), \(t = 21.0\) s). In the draft-tube inlet, the high velocity is near the wall, while an overload vortex rotates in the center (Figure 7(b)).

![Figure 7 Flow patterns in the turbine mode (\(t = 21.0\) s)](image)

3.4.4 In turbine-braking mode
With the further increase of speed, discharge decreases sharply, leading to the working point sliding into the turbine-braking region. The water energy is dissipated by the strong impacts on runner blades, violent separation vortices in blade passages, and fast rotating vortex attached to the draft-tube wall (Figure 8, \(t = 26.0\) s). These are owing to the larger centrifugal force at the runner inlet, stronger circumferential velocity at the runner outlet, and smaller discharge in the flow passages.

![Figure 8 Flow patterns in the turbine-braking mode (\(t = 26.0\) s)](image)
3.4.5 In reverse pump mode

As the discharge further decreases, the working point slides into the reverse pump region. In this mode, the runner rotates in the turbine direction with high speed, causing the strong centrifugal force propelling the water upstream. In the runner, the water flows upstream with the main stream attached to the blade suction surfaces (Figure 9(a)). In the vaneless space, the circumferential velocity is very strong. In the draft-tube, with a swirl the main flow is in the center (Figure 9(b)). The low velocity vortices are abundant in the blade and vane passages, and the strong circular flow is apparent in the vaneless space.

3.4.6 Runner inlet flow transition in the S-shape region

Within the S-shaped region, the most mentionable flow phenomenon is the backflow transition [17] at the runner inlet. Xia [18] presented a chart that depicts the backflow features and their corresponding occurring locations in the S-shaped region. Like the turbine (head 510m) in [18], the present turbine (head 700m) also has similar rule: Backflows are on the hub side in the high speed turbine mode (Figure 10(a)), first transiting to the center when working point comes into the turbine braking mode (Figure 10(b)), and keeping staying in the center in the reverse pump mode (Figure 10(c)).

3.5 Pressure spectrum analysis

Figure 11 shows the time-frequency spectra of the transient pressures at the monitoring points, which were obtained by using the short time Fourier transform (STFT) [26]. At the beginning of the runaway process, the dominant frequency in the vaneless space (P3) is the blade passing frequency (BPF) and its integer multiples. Due to the decreases of rotational speed and discharge, the uneven vortex structures in the vane and runner regions cause the generation of relatively low frequency signals with relatively higher amplitudes (0.0 s < t < 5.5 s). In the transferring process from the pump mode to the pump-braking mode, it is obvious that, due to the separation vortices and the impact on runner blades,
higher frequency signals and low frequency signals mix up and display no clear boundary, and the changing trends of frequencies are in accordance with the change of rotational speed ($5.5 \text{ s} < t < 13.1 \text{ s}$). When the unit enters the turbine mode, the frequency distinction becomes clear, and the amplitudes decrease greatly near the better operation region ($13.1 \text{ s} < t < 21.5 \text{ s}$). These are because the vortices in the vane region become weak or disappear. Once the working point leaves the optimal and goes into the S-shaped region, the amplitudes increase gradually because of the violent rotor-stator interaction induced by backflows ($21.5 \text{ s} < t < 28.0 \text{ s}$). After $t = 28 \text{ s}$, the working point oscillates in the S-shaped region, and all the low frequency and high frequency signals generate and disappear periodically. These should be due to the various vortices in the vane region, vaneless space, runner, and draft-tube. The frequency spectra in the draft tube (P3) are featured with abundant low-frequency components, in which the amplitudes become stronger in the S-shaped region and the frequencies become larger when the speed becomes higher.

![Figure 11 Frequency spectra for pressures at the monitoring points](image)

### 4 Conclusions

In this study, the fast runaway transient process after the pump trip of a 700m head pumped-storage plant was simulated by using the 1D pipe-3D turbine coupled CFD method. The phenomena composed of water hammer fluctuation, pressure pulsation transition, and flow pattern evolution were described and analyzed. The main findings are summarized as follows.

1. The parameter histories from the 1D-3D coupled simulation are reasonable. The detailed pressure pulsations and clear corresponding flow patterns are important for understanding the phenomena and providing more realistic design values.

2. The pressure pulsations in the pump-braking mode and in the S-shaped region are severe, and the undamped large scale oscillations along the trajectory loops make pressure pulsations more obvious. The rotor-stator interaction, rotating stalls in vane regions and blade passages, and vortices in draft-tube are the main sources of pressure pulsations, with the first one of high-frequency and the latter two of low-frequency.

3. The flow patterns fast evolve during the runaway process and have different features in different modes. The featured patterns in the pump-braking mode are the strong impact on the blade pressure sides at runner inlet and the backflows in the draft-tube center; in the turbine-braking mode are the strong circumferential flow in the vaneless space and the backflows at the runner inlet; in the reverse-pump mode are the strong impact on the blade suction sides at the runner outlet and the upstream flowing jet along blade suction sides.

4. The undamped oscillation of working point, the deviations between the dynamic trajectory and the corresponding static curve, and the transitions of runner inlet backflow were found. The mechanism of these phenomena should be discussed in depth in the future studies.

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