Three-Dimensional CFD Simulations of Start-Up Processes of a Pump-Turbine Considering Governor Regulation

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Abstract: The pumped-storage power station is an efficient stability regulator of the power grid. However, due to the instability of the pump-turbine in the S-shaped characteristic region, rotational speed fluctuation is easy to occur in the speed no-load condition, making synchronization with and connection to the grid difficult. To investigate the key factors of these difficult grid connections, the start-up processes of a practical pump-turbine under the lowest head condition were simulated by using the three-dimensional CFD method, in which the governor regulating equations with different regulating parameters were integrated successfully. The results show that the working points oscillate with the fluctuations of rotational speed, discharge, and torque, and different regulating parameters have a significant influence on the dynamic histories. In addition, the internal flow patterns, especially the backflows at the runner inlet, keep apparent values at the middle span (0.5 span) but have regular transitions near the shroud side (0.7–0.8 span). The faster the guide vanes adjust, the faster the backflows change, and the larger the macro parameters fluctuate. Overall, the instability of the start-up is the result of the periodical evolutions of backflows at the runner inlet, because the trend and period of the radial velocities at different inlet span locations are consistent with those of the discharge.

Keywords: pump-turbine; start-up; regulating system; no-load condition; s-shaped characteristic curve

1. Introduction

With the rapid growth of renewable energies [1], keeping the stability of the flexible power grid becomes a focused issue [2]. As a large piece of energy storage equipment, the pumped-storage power station (PSPS) plays an important role in peak shaving [3], valley filling [4], and frequency regulation [5]. However, in actual operations, due to the influence of the S-shaped and hump-shaped characteristics of pump-turbines [6,7], violent pressure pulsations and unit vibrations are the troubled problems for many PSPSs when the working points deviate from the optimal ones. During the start-up process, the fluctuations of rotational speed and torque may become obvious when the working point is approaching the speed no-load condition, where the S-shaped characteristics exist, making the synchronization and grid connection difficult [8]. Many cases were reported; for example, Tiantang PSPS failed to start up many times [9], Baoquan PSPS experienced power oscillations [10], and Xianyou PSPS encountered rotational speed fluctuations [11].

Previous research on start-up instability showed that the positive slope at the runaway point in the no-load opening curve on the $n_{11}$–$Q_{11}$ plane may be the key reason, because in this condition, the energy transferred to the water body can be an excitation of the
The positive slope is due to the S-shaped characteristic curve, for which the cause was found to be the flow loss at the runner inlet caused by the small blade inlet angle, and the S-induced instability may originate from the extremely complicated flow patterns inside the runner channels [6]. To stabilize the start-up process, optimizing Proportion–Integration–Differentiation (PID) regulating parameters should be first tried, because it is convenient and economic. When PID regulation is ineffective, setting misaligned guide vanes (MGV) can be tried [13,14]. Why these two approaches are effective or ineffective needs to be analyzed in depth. However, most of the relevant researches applied one-dimensional (1D) simulations [15–17] and experiments [18,19], which can reflect the variation of macro parameters but cannot investigate the internal flow mechanism.

Therefore, to understand the internal flow mechanism of start-up instability, several three-dimensional computational fluid dynamics (3D CFD) simulations have been conducted. Through investigating the transitions of flow patterns and pressure pulsations in the pump-turbine, the sources of instability at the no-load working point or during the start-up transient process can be found, and solutions can also be proposed. Li [20] conducted the 3D simulations of a pump-turbine under a turbine braking condition with an MGV device and deduced that the vibration reason of the pump-turbine unit during the start-up process is the secondary flow generated in the vaneless space and the attached vortex falling off from guide vanes. Similarly, Thomas [21] carried out the unsteady simulations of the pump-turbine at the no-load working point with a small opening, concluding that the local vortices that form and evolve in the runner passages near the leading edge were the source of no-load instability. Gentner [12] investigated the turbine behavior at runaway with numerical and experimental methods and concluded that the instability source is located in the runner passages or vaneless space and the abundant vortex structures may be the cause of instability. Liu [22] adopted numerical simulation to investigate the characteristics of the internal flow with different guide vane openings and operating conditions and found that the strong spatial heterogeneity of backflows occurs at the runner inlet and pointed that the guide vane opening is also a key factor affecting the backflow characteristics. These works were for the instability mechanism of fixed working conditions.

As for the transient start-up process, 1D methods are normally used for optimizing the start-up procedures and governing rules, and 3D simulations are recently introduced for investigating flow mechanism. Unterlugauer [23] adopted different guide vane action rules in 3D CFD simulations of start-up processes of a conventional Francis turbine. Without S-shaped characteristics, the unit can be connected to the grid smoothly, but a counter rotating draft tube vortex that influences runner stability was discovered. Li [24] simulated a start-up process of a pump-turbine under the high water head and found that the grid connection was smooth because of no obvious S-shaped characteristics in the no-load opening, even if a large number of unstable vortex structures were inside the unit. Casartelli [25] simulated the transient start-up process based on 1D simulation data and found that the continuously non-uniform variation of inflow and backflow at the runner inlet was the main factor leading to no-load oscillation during the start-up process.

To sum up, although the research and cognition on the start-up stability have made great progress, including the finding of unstable flow patterns and the influences of start-up procedures and governor parameters, how the internal flows affect the start-up stability is still ambiguous. Therefore, in this paper, considering the governor regulating equations with different regulating parameters, the start-up processes of a practical pump-turbine were simulated successfully by using the 3D CFD method. The instabilities of the start-up processes, including the flow mechanism, pressure pulsations, and effects of regulating parameters, were revealed. The main contents are as follows: (1) the start-up process and regulating system are introduced in Section 2; (2) the numerical setup is presented in Section 3; (3) the results including macro parameters and internal flow patterns are analyzed in Section 4; (4) the conclusions are provided in Section 5.
2. Start-Up Process and Regulating System

2.1. Stages in Start-Up Process

The “open loop–closed loop” control is adopted during the start-up process [26]. The procedures are as follows: after the unit receives the start-up command, the guide vanes will open with prescribed laws to the given opening and keep stalling to wait for the rotational speed rising to the given speed (here 90% of the rated rotational speed); after that, the governor regulation will be put into operation automatically to push the rotational speed reaching 100% of the rated speed; after that, the governor regulation will be put into operation automatically to push the rotational speed reaching 100% of the rated speed; after that, the governor regulation will be put into operation automatically to push the rotational speed reaching 100% of the rated speed; after that, the governor regulation will be put into operation automatically to push the rotational speed reaching 100% of the rated speed.

2.2. Governor Regulation Model: Proportion Integration Differentiation (PID)

When the rotational speed rises to the given value, the governor regulation is required to participate in adjusting the rotational speed to approach the (1 ± 0.2%) of the rated speed; then, the unit can connect to the power grid stably. Nowadays, the regulating models include frequency regulation (PID), power regulation (PI), and opening regulation [27], while frequency regulation is often used in the start-up process [16, 17, 28]. In the PID model, the governor adjusts the guide vane opening according to the rotational speed; then, the rotational speed should finally stabilize at the rated value, and the final opening is the no-load opening under the current head. According to the existing research, the equation of coupling the guide vane opening and rotational speed in PID regulation is shown in Formula (1) [29]

\[
A_1 \frac{d^3 y}{dt^3} + A_2 \frac{d^2 y}{dt^2} + A_3 \frac{dy}{dt} + A_4 y = -(B_1 \frac{d^2 x}{dt^2} + B_2 \frac{dx}{dt} + x)
\]

where \(A_1 = b_p T_d T_\tau T_y, A_2 = b_p T_d T_\tau + b_p T_d T_y + b_p T_d T_y, A_3 = b_p T_d + b_p T_d + b_p T_y, A_4 = b_p, B_1 = T_y T_\tau, \) and \(B_2 = T_d.\) The differential transformation can be obtained in Formulas (2) and (3), which reflect the coupling of the guide vane opening and rotational speed [29].

\[
y(t) = \left[ C_1 \Delta t^3 + (3A_1 + 2A_2 \Delta t + A_3 \Delta t^2) y(\Delta t) - (3A_1 + A_2 \Delta t) y(\Delta t - 2\Delta t) + 3A_1 y(\Delta t - 3\Delta t) \right] / (A_1 + A_2 \Delta t + A_3 \Delta t^2 + A_4 \Delta t^3)
\]

\[
C_1 = \frac{-(B_1 + B_2 \Delta t + \Delta t^2) \omega(t) - (2B_1 + B_2 \Delta t) \omega(t - \Delta t) + B_1 \omega(t - 2\Delta t)}{/(\omega^0 \Delta t^5) + 1}
\]
in which $x$: relative speed deviation of unit, with $x = (\omega - \omega_0)/\omega_0$; $y$: relative travel of servomotor (related to the relative opening of guide vanes); $b_p$: permanent droop; $b_t$: temporary droop; $T_d$: time constant of damping device; $T_n$: differential time constant of frequency measurement; and $T_Y$: servomotor response time constant.

3. Numerical Methods

3.1. Computational Domain and Main Parameters of the Pump-Turbine

The object of the simulations is a prototype pumped-storage power station, which adopts the layout called “one diversion tunnel for two units". Only one unit and its hydraulic system were considered here. The computational domain shown in Figure 2 includes the upper reservoir, diversion tunnel, surge tank, penstock, pump-turbine unit, tailrace tunnel, and lower reservoir. The basic parameters of the pump-turbine are shown in Table 1.

Figure 2. Schematic diagram of numerical computational domain.

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

| Parameter                          | Value  | Parameter                          | Value  |
|------------------------------------|--------|------------------------------------|--------|
| The runner inlet diameter $D_1$ (m) | 5.26   | The rated speed $\omega_0$ (rpm)   | 250    |
| The runner out diameter $D_2$ (m)  | 3.57   | The inertia of rotating parts $J$ (kg m²) | 4,747,287 |
| The rated head $H_r$ (m)           | 195    | The number of runner blades $z_b$  | 9      |
| The rated output $P_r$ (MW)        | 306    | The number of guide vanes $n_{gv}$ | 20     |
| The weighted average efficiency (%) | 92.382 | The number of stay vanes $n_{sv}$  | 20     |

In the simulations, 3D CFD was used for the pump turbine, while the 1D method of characteristics (MOC) was used for other components, and the partly overlap coupling method was used for the 1D and 3D coupling interface [30–32]. Figure 3 shows the brief schematic diagram of overlapped meshes between 1D and 3D domains, and the detailed information is based on these three references.

Figure 3. Schematic diagram of overlapped meshes between 1D and 3D domains.
The User-Defined Function (UDF) in Fluent can obtain the piezometer head \((H)\) and discharge \((Q)\) at the current timestep; then, the MOC method solves two governing equations (the \(C^+\) and \(C^-\) equations in Formula (4)), which are transformed from the 1D continuity and momentum equations, and the piezometer head \((H)\) and discharge \((Q)\) at the next timestep can be obtained. Finally, the head at the next timestep calculated by the MOC method can be set as the 3D boundary condition to conduct the 3D simulation.

\[
\begin{align*}
C^+ : H_{i}^{t+1} & = C_{p} - BQ_{i}^{t+1} \\
C^- : H_{i}^{t+1} & = C_{M} + BQ_{i}^{t+1}
\end{align*}
\] (4)

where \(C_{p} = H_{i-1}^{t} + BQ_{i-1}^{t} - RQ_{i-1}^{t} \bigg| Q_{i-1}^{t} \bigg|, C_{M} = H_{i+1}^{t} - BQ_{i+1}^{t} + RQ_{i+1}^{t} \bigg| Q_{i+1}^{t} \bigg|, B = a_{0}/(gA),\) respectively. The superscript \(t\) and subscript \((i, N)\) indicate the timestep and node index, respectively. \(a_{0}, g, A,\) and \(R\) are the wave speed of the water, gravity acceleration, conduit cross-sectional area, and friction resistance coefficient of the conduit, respectively.

3.2. 3D CFD Setup

Mesh generation: Three-dimensional (3D) CFD simulations were carried out by using commercial software ANSYS FLUENT, and the mesh was generated by using ICEM-CFD. The upstream extension pipe, the spiral casing, the runner, the draft tube, and the downstream extension pipe were discretized by hexahedral structure grids, and local grid refinement was used in the vane region (Figure 4a). To achieve dynamic grid simulation for the guide vane region, the initial guide vane opening was set at 0.1° (Figure 4b), and the initial runner rotational speed was set at 0.1 rad/s. Since the final guide vane opening during the start-up process is near 10°, we conducted the mesh dependence considering the numerical torque normalized by the rated one at a runaway point under 10°, and we finally chose the mesh with the grid number 5.47 million based on the time-consuming factor, though it is coarse for the whole computational domain and the \(y+\) values are large in this prototype pump-turbine unit (Figure 4c).

![Figure 4. Schematic of mesh. (a) Grids for different parts; (b) Grids for guide vane region in the initial opening; (c) Mesh dependence.](image-url)

Numerical scheme: In these simulations, double precision was adopted because the variation of the guide vane opening at each timestep was very small. Thus, the CFD simulation requires a lot of computing resources due to the double precision and dynamic
grid simulation. Finally, to reduce the simulation time and keep accuracy at the same time, the timestep was selected as 0.002 s. The SAS-SST turbulence model [6,23] and SIMPLEC algorithm were adopted.

Boundary conditions: The upper and lower reservoirs were set as pressure boundaries, with the total pressure and static pressure specified at the inlet and outlet of the water conveyance system, respectively. The 1D–3D coupling method was used for exchanging discharge and pressure data between the 1D pipelines and the 3D pump-turbine. For the rotor–stator interface configuration, the multiple reference frame approach was used for the runner zone in steady-state simulations, while the sliding mesh approach was used in transient simulations.

3.3. Selection of Start-Up Condition

In the low head condition, obvious oscillations of torque and guide vane opening occurred many times for the concerned pumped-storage power station. This no-load instability was attributed to the large guide vane opening at the no-load condition and the obvious S-shaped characteristic curve along with improper regulating parameters. The recent research shows that the unstable factors of start-up are mainly affected by the head, the given opening, the given speed, \( k_p, k_i, k_d \), and other factors [33]. These factors will directly affect the variation of the rotational speed and the guide vane action because they have significant influences on the working point trajectory, which further affect the time needed for unit grid connection. However, to investigate the unstable internal flow mechanism during the start-up, especially when the working point enters the S-shaped region, the regulating parameters (including \( k_p, k_i \) and \( k_d \)) are more important to be considered. In addition, recent research pointed out that the given opening at the initial stage and the regulating parameters have a great effect on the stability and grid connection time. Therefore, in this study, a relatively small given opening with two sets of regulating parameters (Case1 and Case2 in Table 2) was selected, and the variation laws of macro parameters, mechanism of turbine flow evolution, and reasons for grid connection difficulty were analyzed and explained.

Table 2. Regulating parameters for the start-up process.

|       | \( b_t \) | \( b_p \) | \( T_n \) | \( T_d \) | \( T_y \) | \( e_g \) | \( k_p \) | \( k_i \) | \( k_d \) |
|-------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|-----------|
| Case1 | 1         | 0         | 1         | 25        | 0.02      | 0         | 1.04      | 0.04      | 1         |
| Case2 | 0.5       | 0.7       | 0         | 0.7       | 0.02      | 0         | 2.2       | 0.285     | 1.4       |

Where \( k_p = (T_n + T_d)/(b_t T_d) \); \( k_i = 1/(b_t T_d) \); \( k_d = T_n/b_t \).

4. Results of Start-Up Processes

4.1. Variation of Macro Parameters

The dynamic trajectory curves of Case1 and Case2 are shown in Figure 5, and the unit parameter is defined as \( n_{11} = n D_1 / \sqrt{H} \) and \( Q_{11} = Q / (D_1^2 \sqrt{H}) \), where \( H = E_1 - E_2 \), with \( E_1 \) and \( E_2 \) being the total energy values at the spiral casing inlet and runner outlet, respectively. The working points start from the zero point quickly and go along the 12.0° curve; then, they move right to the S-shaped region. After the rotational speed reaches 90% \( n_{r_c} \), the two dynamic trajectory curves show different paths and convergence features due to the different governor parameters. When the working point in Case1 reaches the S region (Figure 5a), the dynamic loop converges and slowly approaches the no-load point, while a divergent oscillation occurs in Case2 (Figure 5b), and the results have a similar trend as in Ref [25].

The variations of macro parameters in Case1 and Case2 can be divided into three stages: guide vane opening stage (\( t = 0–12 \) s), guide vane stalling stage (\( t = 12–20.75 \) s), and guide vane regulating stage (\( t > 20.75 \) s).
(1) $t = 0.0-20.75$ s:

After receiving the start-up command, the governor opens the guide vanes within 12 s from $0^\circ$ to $12^\circ$ according to the given rules in the early stage ($t = 0-12$ s), during which the discharge and runner torque increase. Then, the guide vanes keep constant opening $12^\circ$ ($t = 12-20.75$ s), during which the discharge and torque begin to decrease after a short increase, which is mainly affected by the runner acceleration.

(2) $t > 20.75$ s:

When the rotational speed rises to $90\%n_r$ at $t = 20.75$ s, the governor activates PID regulation, and the simulated results correspond to a faster rotational speed increase. At this time, the guide vane opening begins to decrease, causing the rapid decreases of discharge and torque, which further leads to a slower increase in the rotational speed. During this period, the selection of regulating parameters has a significant impact on the rotational speed and even determines whether the pump-turbine unit can be successfully connected to the grid. The reason is that the no-load working point is located in the extremely unstable S-shaped region, but the regulating system affects the change of discharge by adjusting the guide vane opening, which leads to an obvious fluctuation of working point and then affects the success of the grid connection.

![Figure 5](image-url)

**Figure 5.** Dynamic trajectories of working point and histories of macro parameters. (a) Case1; (b) Case2; where, $y_{ed}$, $n_{ed}$, $Q_{ed}$, and $T_{ed}$ are the dynamic values normalized by the rated ones.

**Case1** (Figure 5a): At $t = 25.1$ s, the rotational speed reaches the $(1\pm0.2\%)n_r$ threshold for the first time, and the working point is near the no-load condition. However, because the torque is still large, the rotational speed continues to increase, and the guide vane opening continues to decrease. At $t = 30.0$ s, the guide vane opening decreases to its minimum, and the rotational speed increases to its maximum when the torque reaches $0.0$ N·m. Then, the rotational speed decreases, but under the action of the regulating system, the guide vane opening will increase again to stabilize the rotational speed. As a result of the low head in this start-up process, the characteristic curve near the no-load point is nearly vertical, and instability may occur due to the positive slope. In Case1, the torque and discharge still fluctuate in a small range but show a convergent trend.
Case2 (Figure 5b): The rotational speed reaches its maximum firstly at \( t = 26.2 \) s with a smaller value than in Case1, and it cannot reach the \((1 \pm 0.2\%)n_r\) threshold. However, it can be seen that after the regulating system is put into operation, the guide vane opening decreases more rapidly and the speed increases more slowly than those in Case1, but then, the guide vanes open and the speed increases again faster, which also exceeds the upper limit of \((1 + 0.2\%)n_r\), and the fluctuations of all macro parameters gradually become larger and larger. Therefore, compared with the results in Case1, the fluctuation ranges of rotational speed, discharge, torque and guide vane opening in Case2 are much larger, which makes grid connection more difficult.

4.2. Radial Velocity Characteristics at Runner Inlet

The instability of start-up is related to the special flow patterns, and their evolutions are also the sources of severe pressure pulsations and vibrations of the pump-turbine unit. The present study found that the start-up instability originates from the transitional flow patterns in the vaneless space and runner passages; therefore, the mechanisms of working point oscillations in the S-shaped region affected by the backflow at the runner inlet can be enlightening. In Casartelli’s paper [25], he pointed out that the backflow and inflow region per channel passage cancel each other out, resulting in a near-zero average flow rate. That means that the inward and outward flowing portions in the vaneless space are close to an equilibrium, and this state changes continuously and irregularly. In addition, according to Thomas’s paper [21], he analyzed the fluctuations of \( Q_{in} \) (inflow) and \( Q_{out} \) (backflow) over three revolutions of the runner and found that these two kinds of flows have to be in equilibrium. Overall, the discharge at the runner inlet can be simplified in Figure 6 (\( Q_{in} > 0 \), the normal inflow to runner passage; \( Q_{out} < 0 \), the backflow to the vaneless space).

\[
\begin{align*}
\text{Vaneless space} & \quad \text{Runner inlet} \quad \text{Runner passage} \\
Q_{in} & \quad Q_{in} \\
\text{Smooth inflow} & \\
\hline
\text{Normal operation} & \quad \text{Start-up condition} \\
Q_{in} & \quad Q_{out} \\
\text{Backflow vortices} & \quad Q_{in}
\end{align*}
\]

Figure 6. Schematic of discharge distributions in normal and start-up conditions.

Therefore, in order to analyze the evolution of flow patterns and pressure pulsations in the pump-turbine during the whole start-up process, monitoring points at different locations including the spiral casing (SC), the stay vane region (SV), the vaneless space (VS), and the draft tube (DT) were selected (Figure 7). For analyzing the inlet flow transitions in detail, the measuring points in the vaneless space were spaced with 0.1 span (SP).

Yang [34] studied the runaway characteristics of different pump-turbines with large openings and showed that the formation and evolution of backflows at the runner inlet at different heights may lead to the instability of the unit. In this study, the same method was used to study the evolution of the radial velocity (backflows) at different inlet heights.
Case1: The original data of velocities at different positions in Case1 are presented in Figure 8a, and it can be seen that the violent pulsations occur at the initial stage and PID-regulated process, namely the unstable phenomena at the runner inlet. Then, the low-pass filtered values are shown to reflect the development trend of flow patterns at the runner inlet. When $t = 0.0–13.6 \text{ s}$, with the increase in discharge and rotational speed, the inflow and backflow alternate, and the amplitudes of velocity pulsations increase firstly and then decrease. The discharge reaches the maximum at $t = 13.6 \text{ s}$ (Figure 8b), but the working point is not located at the optimal one, because the rotational speed is still small; therefore, the velocity pulsations are still large. When $t = 13.6–20.75 \text{ s}$ (Figure 8b), the guide vane opening remains unchanged, and the rotational speed continues to increase; then, with the increase in centrifugal force and decrease in discharge, the velocity pulsations decrease. When $t > 20.75 \text{ s}$, the regulating system is activated, and the unit will enter the no-load condition gradually; then, the radial velocities at the runner inlet fluctuate violently, which is accompanied by strong rotor–stator interaction and unstable flow patterns.

Xia [6,35], Yang [34], and Fu [36] pointed out that the backflows at the runner inlet will move due to the change of working point, and the locations of backflows are also affected by the runner shape. In small discharge condition, the backflows are mainly caused by centrifugal force and largely concentrated in the middle of the runner inlet. In addition, in the transient process, due to the variation of macro parameters such as discharge, the backflows have transitions. In order to analyze the variation of radial velocity at the runner inlet in detail, the low-pass filtered values are shown in Figure 8b to reflect the development trend of flow patterns. Overall, it can be found that the radial velocity transitions are mainly in the middle region, and especially the velocity values at 0.8 SP show alternating features between positive and negative. In addition, the fluctuation period and trend at 0.8 SP are consistent with those of discharge.
To explain the fluctuations of radial velocity at different monitoring points more clearly, the distributions of radial velocities at the runner inlet corresponding to the low-pass filtered velocity values are shown in Figure 9. With the increase in guide vane opening, the inflow will enter the runner passages, leading to the radial velocities gradually increasing (Figure 9b). In addition, although the rotational speed continues to increase (Figure 9c), the radial velocity values at each location are still different because the working point is in the off-design condition all the time.

**Figure 9.** Distribution profiles of radial velocity at typical times. (a) $t = 0$ s; (b) $t = 7$ s; (c) $t = 20$ s.

When the operating point enters the S-shaped region, the selected typical times are when the discharge is at the minimum and maximum. Obviously, the backflows are mainly generated in the middle region (0.3 SP–0.8 SP), and their periodic occurrence and disappearance are found to be at 0.7 SP and 0.8 SP. For example, at $t = 31$ s, the backflow velocities at 0.8 SP increase, but the inflow velocities at 0.1 SP and 0.9 SP decrease, indicating that the backflow region diffuses from the middle span to the hub and shroud sides as the discharge decreases (Figure 10a). When $t = 39$ s, the radial velocities at 0.7 SP and 0.8 SP are reversed; namely, the backflows are transformed into inflows, and also, the inflow velocities at 0.1 SP, 0.2 SP, and 0.9 SP increase, indicating that the backflow region shrinks to the middle span, which is accompanied by the increase in discharge (Figure 10b). Therefore, the variation of discharge comes from the contraction and expansion of the backflow region at the runner inlet. When the backflow region expands from the middle span to other two sides, the backflows at the runner inlet seriously block the flow into the runner, resulting in the rapid discharge decreasing (Figure 10c,e). However, the backflow region shrinks from the hub and shroud sides to the middle; then, the upstream inflow can smoothly enter the runner passages from the hub and shroud sides as the discharge increases (Figure 10d,f). As the macro parameters gradually tend to be stable, the variation of radial velocity distributions also becomes smaller.

**Figure 10.** Radial velocity distributions at typical times in Case1. (a) $t = 31$ s; (b) $t = 39$ s; (c) $t = 47$ s; (d) $t = 54$ s; (e) $t = 63$ s; (f) $t = 71$ s.
Case 2: The backflow characteristics at the runner inlet in Case 2 show different characteristics from those in Case 1 after the working points enter the S region. Generally speaking, as the fluctuation amplitudes of macro parameters increase gradually, the fluctuation characteristics of radial velocity show the same variation trend. However, the backflow region is also mainly concentrated in the middle part (0.3 SP–0.8 SP), which may be affected by the S-shaped characteristics in the small discharge condition, and it can also be found that the velocity variation amplitudes at 0.7 SP and 0.8 SP are larger than those in Case 1, which is also consistent with the variation of discharge (Figure 11).

Similarly, the simple diagram of radial velocity is presented to analyze the backflow characteristics in Case 2 (Figure 12) as well as the typical times when the discharge is the minimum and maximum value. Overall, because the occupation range and generation time of backflows are affected by the different closing laws of guide vanes, the shapes of velocity distribution are obviously different from those in Case 1, and especially at the minimum discharge time. The backflow region is very large, and the inflow region is very small. In addition, the shapes of these backflow regions vary in a large range, which means that there is a large fluctuation of discharge.

Figure 11. Backflow features at the runner inlet during start-up in Case 2. (a) Original data; (b) The low-pass filtered data.

Figure 12. Radial velocity distributions at typical times in Case 2. (a) $t = 42$ s; (b) $t = 48$ s; (c) $t = 55$ s; (d) $t = 61$ s; (e) $t = 68$ s; (f) $t = 75$ s.
4.3. Evolutions of Flow Patterns in Pump-Turbine

In order to analyze how the regular evolution of backflows and blockages at the runner inlet affect the start-up instability, the streamlines at typical times on vertical sections including the vane region and runner passages are selected.

Firstly, in the initial stage, it can be seen from Figure 13 that the radial velocity variations at different heights are consistent during the small discharge period, and the variation amplitude is large; therefore, flow pattern evolutions at typical times at 0.5 SP are selected (Figure 14). The rotations of Blade1 and Blade2 show that the sudden opening of the guide vanes and the serious deviation of the inlet velocity triangle from the design condition cause the upstream water to attack the pressure surface. Then, part of the water enters the runner blade passages, and the others return back to the vaneless space and enter the previous channel. At the fore-ends of runner blade passages, large stall vortices can be seen attaching at the suction surfaces, and water only enters the runner along a narrow space at the pressure surface. When \( t = 2.2 \) s, the monitoring point is near the Blade1, locating at the backflow region, and the radial velocities are negative; with the clockwise rotation of runner, the monitoring point is far away from the blade and is in the inflow region between Blade1 and Blade2; namely, the inlet velocity is positive \( (t = 2.6 \) s); then, at \( t = 3.2 \) s, Blade2 rotates to the monitoring point, and the radial velocities are negative again. With the increases of rotational speed and discharge, the variation periods of radial velocity pulsation become shorter, the amplitudes become smaller, and the backflows disappear gradually.

Secondly, the development trend and range of backflows in Case1 and Case2 are different after the governor regulation is activated. As shown in Figures 10 and 12, the backflows keep apparent values at 0.5 SP, but they have transitions near the shroud and hub sides, especially at 0.8 SP. Therefore, the evolutions of streamlines colored by velocity...
in the radial direction at the vertical section plane and runner passages are selected to analyze the instability sources after the working point enters the S-shaped region.

Figure 15 shows the flow pattern evolutions in Case1. Due to the gradual variation but convergence of various macro parameters, the range of backflows at the runner inlet tends to be stable, and there is only an obvious mutation at the blade leading edge. For example, at $t = 31$ s (small discharge condition), inflow enters the runner passages mainly across 0.1 SP and 0.9 SP. The middle position is disturbed by runner blades and the upstream flow, forming strong shear action and vortex structures, which seriously blocks the water flow into the blade channels. At $t = 39$ s, the streamlines on the shroud side are relatively smooth (0.8 SP and 0.9 SP), but on the hub side, the vortex core gradually generates, which indicates that the backflow region moves from the shroud side to the hub side. At $t = 47$ s, the discharge reaches its minimum value again, which indicates that the backflow region is larger than that at $t = 39$ s and moves to the shroud side. The above phenomenon indicates that the discharge variation comes from the backflow transitions at the runner inlet after entering the S-shaped region, and the range and shape of backflows change all the time since they move up and down regularly. At the runner outlet, the water flow attacks the runner blades, which is caused by the upward lift force due to the small discharge and large rotational speed.

![Flow pattern evolutions on a vertical section during the regulating period in Case1.](image)

**Figure 15.** Flow pattern evolutions on a vertical section during the regulating period in Case1. (a) $t = 31$ s; (b) $t = 39$ s; (c) $t = 47$ s; (d) $t = 54$ s; (e) $t = 63$ s; (f) $t = 71$ s.

In Case2 (Figure 16), after the working point enters the S-shaped region, the large variation of guide vane opening leads to the large variations of discharge and the backflow region. To show the flow pattern transitions, the same vertical sections at typical times when discharge is at its minimum and maximum are selected. When the discharge is the minimum, the backflow range is large (Figure 16a,c,e), which almost occupies the whole inlet; therefore, the backflow region moves slightly. However, when the discharge is the maximum, the backflow range becomes small (Figure 16b,d,f), and the main position of backflows moves at these three times. After that, with the large variations of macro parameters, the moving range of backflows is much larger than that in Case1.

Figures 15 and 16 only show the flow patterns at a specific position and at a specific time, and they do not represent those in all flow channels. In addition, the flow pattern evolutions from the hub to the shroud side in vertical sections will be also affected by blade passage distortion, causing the non-uniform stall vortices in every runner passages. Since the radial velocities at 0.8 SP vary violently and the backflows are always at 0.5 SP, the streamlines on 0.8 SP are selected for the following analysis.
Case 1: Since the variations of macro parameters in Case 1 gradually stabilize after $t = 47$ s, the flow patterns in the runner passages will be also similar at different times (Figure 17). Overall, almost every channel generates blockages and velocity is relative large at the runner inlet, and the impact action is serious, causing the strong backflows. However, the distribution along the passages is uneven and also several channels can accept the normal inflow.

Case 2: When the discharge is the minimum value, the development trend of flow patterns is consistent with that in Case 1. However, because the discharge in Case 2 is smaller, the flow pattern disorder at suction surfaces and the runner outlet is more severe ($t = 55$ s and $t = 68$ s), and backflows occur almost at each runner inlet (Figure 18). At the time of maximum discharge ($t = 61$ s and $t = 75$ s), the velocity at 0.8 SP increases obviously, indicating the less severe inlet impact and that the flow blockage is small.

In general, the two cases in this paper are presented with small opening and small discharge. After entering the S-shaped region, due to the large rotational speed and centrifugal force, the upstream water cannot completely enter the runner passages, forming backflows and causing stall vortices and instability. However, because of the different regulating laws of guide vanes, the dynamic trajectories of working points are different, and the flow pattern evolutions at the runner inlet are also different.

Figure 16. Flow pattern evolutions on a vertical section during the regulating period in Case 2. (a) $t = 42$ s; (b) $t = 48$ s; (c) $t = 55$ s; (d) $t = 61$ s; (e) $t = 68$ s; and (f) $t = 75$ s.

Figure 17. Streamlines in the runner passages at typical times in Case 1. (a) $t = 47$ s; (b) $t = 54$ s; (c) $t = 63$ s; (d) $t = 71$ s.
4.4. Pressure Pulsation Characteristics

(1) \( t = 0.0–20.75 \) s:
At \( t = 0.0 \) s, the low-pass filtered pressure at the spiral casing inlet and vanes region is affected by the upstream water level, while those in the vaneless space and draft tube are affected by the downstream water level. When the guide vanes are opened, the pressure before the guide vanes is affected by the negative water hammer, causing the pressure decrease in spiral casing and vane regions, while the pressure in the vaneless space, which is behind the guide vanes, increases continuously. However, the pressure mainly decreases in DT, because the runner acceleration has a greater influence on the pressure reduction than the water hammer effect. In addition, it can be clearly seen that the amplitudes of pressure fluctuations at the monitoring points in the vaneless region are significantly higher than those in other locations, which is due to the more violent rotor–stator interaction caused by the acceleration of the runner. When \( t = 12 \) s, the guide vanes stop opening; then, the discharge decreases, the pressure before the runner is subject to positive water hammer, and the pressure rises. Moreover, with the increase in the rotational speed, the working point moves toward the better operating region, and the high amplitude pulsations in the vaneless space decreases gradually.

(2) \( t > 20.75 \) s:
Once the regulating system is put into operation, the guide vane opening decreases rapidly, causing the positive water hammer upstream from the guide vanes and the negative water hammer in the draft tube. In addition, due to the unstable characteristics in the S-shaped region and violent rotor–stator interactions, the amplitudes of pressure pulsations are both very high in these two cases (Figures 19a and 20a). For the low-pass filtered data (Figures 19b and 20b), the variations in the vaneless space are relatively smaller due to the obstruction by guide vanes in both two cases. However, by comparison, due to the variations of the rotational speeds, the histories of pressure in the spiral casing and draft tube show the characteristics of convergence in Case1 but divergence in Case2.
5. Conclusions

To investigate the instability during the start-up transient process, 3D CFD simulations considering the PID regulating system were conducted under the low head condition, and the stability mechanism focusing on the unstable flow patterns and pressure fluctuations has been investigated. Due to the large final no-load opening and obvious S-shaped characteristics, the internal flow patterns, affected by the regulating parameters, show different evolution development, causing the different variations of the rotational speed, discharge, torque, and guide vane opening. The conclusions are as follows:

1. The main reason for the oscillations of parameters such as the rotational speed and discharge is the evolutions of backflows at the runner inlet. The fluctuation period and trend of radial velocities at the runner inlet, which represent the evolutions of backflows, are consistent with those of the discharge in the PID-regulated process, and they reveal the reasons of the equilibrium and irregular between inflow and backflow.

2. The backflows at the vertical sections and runner passages are all show uneven and time variant distributions, and the pressure pulsations in the vaneless space are the most violent. The low-pass filtered trend of pressure pulsations is determined by the fluctuation of rotational speed, especially before the guide vanes.

3. The regulating parameters \(k_p, k_i,\) and \(k_d\) directly determine the angular speed of guide vanes and have a large influence on the evolutions of inflow and backflow at the runner inlet, leading to the different working oscillations under different regulating parameters.

4. Only one pump-turbine runner has been selected in this study to analyze the effects regulating parameters on the start-up process, but different runners have different S-shaped characteristics, which may affect the start instability. In addition, different starting rules, including a different given speed, given opening, and operating head, can also determine the success or failure of the grid connection. Therefore, in future works, the above-mentioned aspects should be comprehensive consideration, and the runner with a more obvious S-shaped characteristic can be chosen to investigate how the macro parameters and backflows evolve during the start-up process. In addition, the more accurate simulation model including the number of grids, the turbulence model, and the precision should be considered comprehensively to obtain the more reliable results, because it may have effects on the accuracy of macro parameters and runner inlet backflow transitions, and increasing the number of grids to capture the more accurate flow patterns will be necessary in the future.

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**Abbreviations**

- $\Delta t$: timestep (s)
- $\omega_t$: rotational speeds of runner at the current timesteps (rad/s)
- $\omega_{t-\Delta t}$: rotational speeds of runner at the previous timesteps (rad/s)
- $T$: torque (N·m)
- $k$: opening/closing speed of guide vanes (rad/s)
- $a$: an invariant constant (rad/s)
- $y(t)$: relative guide vane openings at the current timesteps (rad)
- $y(t-\Delta t)$: relative guide vane openings at the current timesteps (rad)
- $\omega_0$: rated rotational speed (rad/s)
- $I$: inertia of rotating parts (kg·m$^2$)
- $x$: relative speed deviation of unit, with $x = (\omega - \omega_0)/\omega_0$ (-)
- $y$: relative travel of servomotor (related to the relative opening of guide vanes)
- $b_p$: permanent droop (-)
- $b_t$: temporary droop (-)
- $T_d$: time constant of damping device (-)
- $T_n$: differential time constant of frequency measurement (-)
- $T_y$: servomotor response time constant (-)
- $y_{ed}$: the dynamic values normalized by the rated guide vane opening (-)
- $n_{ed}$: the dynamic values normalized by the rated rotational speed (-)
- $Q_{ed}$: the dynamic values normalized by the rated discharge (-)
- $T_{ed}$: the dynamic values normalized by the rated torque (-)
- $D_1$: the runner inlet diameter (m)
- $D_2$: the runner out diameter (m)
- $H_r$: the rated head (m)
- $P_r$: the rated output (MW)
- $z_b$: the number of runner blades (-)
- $n_{gv}$: the number of guide vanes (-)
- $t$: the superscript timestep (-)
- $i$: the subscript node index (-)
- $N$: the subscript node index (-)
- $a_0$: the wave speed of water (-)
- $g$: the gravity acceleration (m/s$^2$)
- $A$: the conduit cross-sectional area (m$^2$)
- $R$: the friction resistance coefficient of the conduit (-)

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