Three-dimensional CFD simulation of the load rejection transient process considering regulating system

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Abstract: Both the flow patterns in hydraulic turbine and the governor regulating system of turbine-generator unit have great influence on the operation stability of units, but they were treated separately in previous studies. In this paper, the equation for regulating system is cooperated into the three-dimensional (3D) computational fluid dynamics (CFD) model of a hydraulic turbine, for attempting the simulation of the small disturbance transient processes in hydropower plants. The 15% load rejection transient process for a model pumped-storage system with a low specific-speed pump turbine is successfully simulated; reasonable torque, speed and other macro parameters are provided; the flow patterns inside the runner are analysed. The results show that the regulation characteristics of the small fluctuation are acceptable and accordant with practical experience, with the stability guaranteed and the attenuation of fluctuating parameters obvious. These verify the feasibility of using 3D CFD to simulate the load rejection transient process, and lay a foundation for further studying the starting stability of units.

1. Introduction

Pumped-storage power stations perform the functions of peaking, frequency regulation, valley filling, standby and others in the power grid, therefore frequent working condition changes and load regulation are required. In the transient processes, the regulation stability of pumped-storage units is very critical. Due to the S and hump shaped characteristics of pump-turbine, problems such as speed-no-load instability, synchronization difficulties, runaway oscillation, and slow convergence of small fluctuations often occur. The stability characteristics are closely related to the internal flow patterns of hydraulic turbine, while the reasonable regulation of speed control system is the main way to solve the problems, therefore both of them are important aspects to determine the operation stability of units. However, previous studies separately considered the effects of turbine flow patterns and speed regulating system on stability. In order to study speed-no-load stability, synchronization difficulties, and other small fluctuation problems, the flow patterns and speed governing system characteristics should be combined.

At present, computational fluid dynamics (CFD) method has already been used to study the 3D flow patterns and their dynamic influence on transient processes. Xia [1] investigated the original cause of the S-shaped characteristics and its correlation with the runaway instability by deriving a new turbine equation with flow losses and analyzing the flow characteristics of three model pump-turbines of different specific-speeds. Zhu [2] studied the impact of guide vane opening angle on the flow...
stability in a pump-turbine in pump mode, and he found that hydraulic losses were the reason for the unstable head variations and the poor flow regime was the source of the losses. Appropriate guide vane opening angles were needed to improve the flow regime and reduce the hydraulic losses. Zhang [3] captured the unstable behaviors of the pump-turbine, which showed that the runaway dynamic trajectories formed loops in the S-shaped region in the unit discharge and unit torque charts of the pump-turbine. Liu [4] carried out a 3D simulation of load rejection transient process of a prototype pump-turbine, and the laws of pressure pulsations and flow pattern evolutions during the linear closing process of wicket gates were analyzed. Chen [5] analyzed the dynamic instability at starting conditions of a prototype pump-turbine when giving the opening law by CFD simulation, and found that the amplitudes of pressure pulsations increase and the rotating stall zones disappear gradually as the wicket gates open to the no-load opening. Christian [6] studied the unstable Characteristics in turbine brake operation of pump-turbines, it was found that stationary vortex formation and rotating stall have finally the same physical cause and they all lead to an unstable characteristic. On the other hand, for the transient process analysis considering regulating system, the one-dimensional (1D) simulation methods were used [7], which can normally meet the practical requirements of project design. However, to solve the instability problems of many pumped-storage power stations [8], 3D CFD and 1D methods should be united, but there was seldom such works being reported.

In this paper, the 3D CFD method is first used to simulate the load regulation transient process of a model pumped-storage power station. Firstly, the models and procedures of introducing the regulating system into the 3D hydraulic turbine model are introduced; secondly, the 15% load rejection transient process is simulated, and the macro parameters and flow patterns are analyzed. There's one point that needs to be emphasized: at the initial time, the machine operated at the maximum load working condition, so a 15% load rejection means the load drops from maximum load to 85% of maximum load.

2. Simulation methods

2.1. 3D CFD setups

Computational domain: The object of this numerical simulation is a model pumped-storage power station, which includes the upstream water tank, penstock, pump-turbine unit, downstream conduit, and downstream water tank. The profile of pump-turbine is shown in figure 1. In order to analyze the changes of the pressure pulsations in different part during the transient process, 9 monitoring points and 3 monitoring surfaces were set, as shown in figure 2. The main parameters of the pump-turbine are shown in table 1.

![Figure 1. Profile of Pump-turbine](image1.png)

![Figure 2. Positions of monitoring points and surfaces](image2.png)
Table 1. Main parameters of the model pump-turbine

| Parameter | Value | Parameter | Value |
|-----------|-------|-----------|-------|
| $D_1$ (mm) | 280 | $\omega$ (rpm) | 1000 |
| $D_2$ (mm) | 146.3 | $z_0$ | 9 |
| $n_{ev}$ | 20 | $n_{ev}$ | 20 |

Turbulence model: The pump-turbine has complex shape and large surface curvature, the RNG k-epsilon turbulence model [9] was adopted to deal with the effect of large curvature.

Mesh generation: Tetrahedron mesh was adopted in the spiral casing, and prismatic mesh in the draft tube. In the stay vanes and the guide vanes region, in order to guarantee the orthogonality and density of the wall mesh, the hexahedral mesh was adopted on the surfaces of guide vanes. Due to the complicated flow patterns and mesh motions, it is possible to guarantee that the area has enough mesh number when the mesh is divided. The runner region was set up as a sliding region, and the blades of runner were discretized by the hexahedral mesh. The total number of grids is about 3.2 million.

Numerical scheme: During the simulation, the results of steady Reynolds Averaged Navier-Stokes were first used as the initial flow field. For transient simulation by the unsteady Reynolds Averaged Navier-Stokes the timestep was chosen as 0.0005s. The convergence criteria of residuals at each timestep were set to 1.0E-4, and the maximum number of iterations per timestep was set to 40. For both steady and unsteady simulations, the SIMPLEC algorithm was chosen to achieve the coupling solution for the velocity and pressure equations. The second order discretization in time and in space was used.

Boundary conditions: The upstream inlet and downstream outlet were set as the constant pressure boundary conditions, according to the upstream and downstream water levels 14.35m and 3.85m, respectively. The total pressure was defined at the upstream inlet, and the static pressure was defined at the downstream outlet. The rated speed of the runner is 1000r/min and the initial discharge is 0.044 m$^3$/s (corresponding to the guide vane opening 27 degrees), which is near the best performance working condition in turbine mode.

2.2. Regulating system equation and unit rotation equation

2.2.1. Regulating system equation. The regulating system of a hydropower station normally has three kind of regulating modes: opening regulation, frequency regulation, and power regulation [10], within which the frequency regulation is usually adopted when the load is accepted or rejected. In this study, the frequency regulation model was chosen, whose governing law is defined as Proportion Integration Differentiation (PID) [11]. The equation of the governor is shown in equation (1), and the discretization of it by using difference method is shown in equation (2). There’s one point that needs to be emphasized: the relationship between servomotor stroke and opening of guide vanes is considered as a linear relationship.

$$\frac{d^3 y}{dt^3} + A_1 \frac{d^2 y}{dt^2} + A_2 \frac{dy}{dt} + A_3 y = -(B_1 \frac{d^2 x}{dt^2} + B_2 \frac{dx}{dt} + x)$$  \hspace{1cm} (1)

where $A_i = b_p T_d T_n T_y$; $A_2 = b_p T_d T_n + b_p T_d T_y; A_3 = b_p T_d + b_p T_d + b_p T_y$;

$A_4 = b_p; B_1 = T_d T_n; B_2 = T_y$.

$$\frac{y^{(i+1)}}{y^{(i)}} = \left[ C_1 \Delta t^3 + (3 A_1 + 2 A_2 \Delta t + A_3 \Delta t^2) y^{(i)} - (3 A_1 + A_2 \Delta t) y^{(i-1)} + 3 A_1 y^{(i-2)} \right] / (A_1 + A_2 \Delta t + A_3 \Delta t^2 + A_4 \Delta t^3)$$

$$C_1 = -(B_1 + B_2 \Delta t + \Delta t^2) n^{(i+1)} - (2 B_1 + B_2 \Delta t) n^{(i)} + B_1 n^{(i-1)} / (n^{(0)} \Delta t^2) + 1$$ \hspace{1cm} (3)
2.2.2. **Governing equation of rotational speed.** When load changes, the runner speed will deviates from the specified value, and the regulating system will gradually adjust it to the prescribed value according to the regulation laws [12]. The deviated rotational speed can be expressed by equation (4)

\[
n^{(i+1)} = (n^{(i)} + 0.1875 \left( M_g^{(i)} + M_t^{(i)} - M_g^{(i+1)} - M_r^{(i)} + 2e_g M_t \right) \Delta t / GD^2 \) / (1 + e_g \Delta t / T_a)
\]

2.2.3. **Method of coupling regulating system with 3D pump-turbine.** In the process of governor adjustment, some parameters such as the dynamic moment, rotational speed, guide vane opening and others are calculated by user-defined function (UDF), then the required rotational speed and angular velocity of the guide vanes are taken as the boundary conditions to take part in the 3D calculation. Method of coupling regulating system with 3D pump-turbine is shown in figure 3. Take the calculation at (i+1)th timestep as an example, it can be seen from equation (1) and (2) that the rotational speed and opening of guide vanes are all related to the dynamic moment of the runner, so the dynamic moment calculated by Fluent is the first key step. Then the rotational speed and opening of guide vanes are calculated in turn according to equation (4) and (2). Thus, the angular velocity of the guide vane can be derived. Finally, the rotational speed and angular velocity of the guide vane are defined as boundary conditions by UDF to take part in the calculation of the 3D flow field. This is the calculation process of one timestep. In the same way, parameters at any time can be calculated using the same calculation steps.

From the above steps, it can be found that the 3D method is explicit and has no iterative process, but the one-dimensional method is implicit at present. Compared to the one-dimensional methods, 3D method can calculate dynamic moment accurately. That is to say, if the moment is correct, then the corresponding output of the rotational speed and opening of guide vanes should be correct. Secondly, there is no iterative process for the PID regulation in the actual power plant. So the steps to calculate the load rejection transient process by the 3D method are more consistent with the actual situation.

Figure 3. Steps to calculate the load rejection transient process by 3D CFD
3. Load rejection transient process of the model pump-turbine

3.1. Governor parameters
A 15% load rejection transient process was simulated. In order to obtain abundant characteristics of the load rejection transient process, the selected governor parameters should ensure multiple oscillations. Therefore, the parameters are selected as $b_l = 0.2$, $b_p = 0$, $T_d = 1.0$, $T_n = 1.2$, and $T_y = 0.02$. In addition, $e_g$ was set to zero to simplify the calculation.

3.2. Results of the transient process simulation

3.2.1. Macro parameters. The changes of the torque, discharge, guide vane opening, and rotational speed during the transient process are shown in figure 4, where the zero second is corresponding the beginning of the load rejection transient process. The results show that when the unit drops 15% load, the kinetic moment is greater than the resistance moment, causing the rotational speed increase. For the pump-turbine, as the initial speed is the rated speed, the centrifugal force of the internal flow in the runner is increased when the rotational speed begin to increase. It is because of this throttling effect that the head of the unit increases instantaneously and the moment of the runner increases suddenly. The action of the governor is based on the motion of the runner, decreasing the guide vane opening while rotational speed increasing. With the rotational speed increasing, the guide vanes continue to close, resulting in the decrease of the moment. Until when the resistance moment is equal to the kinetic moment, the rotational speed stops to rise and reaches a maximum. Then, the kinetic moment continues to decrease, and the speed begins to decrease. Since it haven’t returned to the rated value, the guide vanes continue to close, and opening reaches to the minimum value at 3.75s. In this process, the change of the flow-rate and moment has the same tendency with the guide vane opening.

The regulation is in the stable region of frequency regulation, and the fluctuation amplitude of the parameters such as speed, opening, torque and flow-rate are all attenuated obviously. Maximum speed deviation is 2.38%. The n11-Q11 characteristic curves of the load rejection transient process are shown in figure 5, showing the instability at the initial time and good convergence overall. In the initial stage of the transient process, the steady operating point A rushes to point B in a short time due to the sudden change of load and the water flow inertia. Then the characteristic curve is gradually convergent under the action of the governor, which indicates the validity of the governing equation. This shows that the regulating system introduced in the 3D pump turbine model plays an important role, and the 3D CFD method is feasible to calculate the load rejection transient process.

![Figure 4. Histories of the macro parameters during the transient process](image-url)
3.2.2. Characteristics of pressure fluctuations. During the load rejection process, the pressure changes violently with the obvious noise and vibration, and the safety of the unit will be seriously affected. Both the guide vane area and the spiral casing area are the upstream side of the runner, shown in figure 6 (a) and (b). At the initial time, the rotational speed increases and the flow rate decreases, resulting the positive water hammer causing the increasing of the pressure. But the rear of the runner is affected by the negative water hammer because of the decrease of flow-rate, and the value of pressure have a downward trend shown in figure 6 (c) and (d). With the action of the governor, there is a downward trend for the opening guide vanes, resulting in the decreasing of the rotational speed and gradually returning to the rated value. In this process, the water hammer effect is weakened, and the pressure before the runner is approaching decreasing, while the pressure behind the runner is gradually increasing, and the pressure fluctuation amplitude is attenuated by the rotational speed of the runner, which shows good convergence.
3.2.3. Flow patterns. Referring to the speed change curve, the critical time points such as at wave crest, wave trough and rated speed are selected to analyse the flow pattern, as shown in figure 7. The initial condition is the best condition, so there is no obvious import impact in the inlet of the runner, with the smooth streamline and the high efficiency. With the development of load rejection transient process, the rotational speed increases first, so the streamline is no longer smooth, and there is an impact on the runner blades. When the speed drops back to the rated value, the phenomenon is most obvious (t=5s). When the rotational speed continues to decrease, the flow field gradually changes to the optimum condition, and when the speed increases from the lowest point to the rated value, the flow field is the most smooth (t=10.7s).

Figure 6. Pressure fluctuation of nine monitoring points

Figure 7. Flow patterns in the runner during the transient process
4. Conclusions
In this paper, a three-dimensional CFD method considering the governor regulating system is proposed for simulating the load regulation transient process. The methods are described, and the macro parameters and flow and pressure fluctuation characteristics are analysed. It is shown that the regulating system has a proper feedback effect that gives reasonable macro parameter histories, and the runner flow fields have obvious influence on the converging characteristics of the trajectory of working point. This first attempt shows the possibility for studying more complex small fluctuation stability problems in pumped-storage systems by coupling 3D CFD with the PID regulating system.

Nomenclature
\( x \): Relative speed deviation of unit  
\( y \): Relative travel of servomotor (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  
\( T_s \): Servomotor response time constant  
\( y^{(i)} \): Relative travel of servomotor at (i)th timestep (Relative opening of guide vanes)  
\( y^{(i-1)} \): Relative travel of servomotor at (i-1)th timestep (Relative opening of guide vanes)  
\( y^{(i-2)} \): Relative travel of servomotor at (i-2)th timestep (Relative opening of guide vanes)  
\( n^{(0)} \): Initial rotational speed  
\( \Delta t \): time step  
\( n^{(i)} \): The rotational speed at (i)th timestep  
\( n^{(i-1)} \): The rotational speed at (i-1)th timestep  
\( M_r^{(i+1)} \): Driven moment at (i+1)th timestep  
\( M_r^{(i)} \): Driven moment at (i)th timestep  
\( M_g^{(i+1)} \): Resistance moment at (i+1)th timestep  
\( M_g^{(i)} \): Resistance moment at (i)th timestep  
\( M_r \): Rated torque  
\( e_g \): Load self-regulation coefficient of power grid  
\( GD^2 \): Moment of inertia  
\( T_a \): Acceleration time constant

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