Prediction and Analysis of the Axial Force of Pump-Turbine during Load-Rejection Process

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Abstract. Load-rejection is an important process of pump-turbine unit. In this process, hydraulic characteristics would change greatly. The axial force of runner and of the entire shaft system will be impacted. This study focuses on a prototype pump-turbine unit. The axial force of this unit during load-rejection is predicted and analyzed. The hydraulic transient analysis can give a more accurate boundary condition of pump-turbine. The computational fluid dynamics simulation considering the full leakage in runner crown and shroud can have a comprehensive prediction of the axial force. Results show that the rotation speed, head and flow rate change during load-rejection process. The axial force is also influenced and changes. The law of axial force variation has a strong relationship with the flow rate variation law. The internal flow analysis found that the pressure difference between crown leakage and internal-runner near outlet is the key factor in inducing axial force. This pressure difference is strongly influenced by runner internal flow regime. During load-rejection process, flow rate strongly varies and causes the huge change of internal flow regime. This is the main reason that runner axial force changes positive and negative alternatively.

1. Introduction

Pumped storage power station is important in modern power grid because of its flexibility in peak-regulation, phase-regulation and emergency power back-up. As the key component in pumped storage power station, reversible pump-turbine has complex unsteady internal flow regime especially in transient process [1]. Understand the unsteady internal flow will be helpful to enhance the security. As a commonly-seen transient process, load rejection has the extremely varying internal flow with very strong pulsations in head, flow rate, shaft power and local pressure field [2]. Therefore, the load rejection process of pump-turbines and other hydro-turbines have received a lot of experimental and numerical investigations.

Experimental investigations are difficult to conduct especially on prototype turbine because the load rejection process is destructive. Researchers have done some tests on the load rejection topic in Francis turbine [3]. The pulsations of flow field and structural field were indicated. However, it is very difficult to capture the internal flow regime and the overall variation of parameter by conducting experiment [4]. Thus, researchers conducted numerical simulations on the load rejection process and other transient
processes based on experimental verification. Currently, the main numerical works are based on the 1D hydraulic transient simulation [5-7]. The 1D simulation can get relatively accurate results on the flow characters in pipeline system. However, it simplifies the pump-turbine internal flow and ignored the turbulence characters during the load rejection process. With the development of computer, the 3D turbulence flow simulation becomes popular in studying the flow in pump-turbines [8-10]. The combination of 1D-3D simulations becomes also possible [11]. There are already many solved works about the 3D flow during load rejection process [12-14]. Different methodologies with advantages and disadvantages have provided us the 3D internal flow field.

In this study, the 1D-3D coupled simulation was conducted to treat the reservoir, pipeline, valves and the pump-turbine as an entire system. This multi-field coupling method will provide a comprehensive analysis on the prototype pump-turbine in the load rejection process.

2. Numerical Method

2.1. Method of 1D Hydraulic Transient Flow Simulation

The 1D flow simulation is based on the continuity equation and momentum equation of transient flow in pressurized pipeline:

\[
\frac{\partial Q}{\partial t} + gA \frac{\partial H}{\partial x} + \frac{f}{2DA} Q|Q| = 0
\]

(1)

\[
a \frac{\partial O}{\partial t} + gA \frac{\partial H}{\partial x} = 0
\]

(2)

where \(Q\) is flow rate, \(H\) is head, \(A\) is passing area, \(D\) is pipeline diameter, \(f\) is Darcy-Weisbach friction factor, \(a\) is wave speed, \(t\) is time and \(x\) is coordinate.

By introducing factor \(\lambda\), the continuity-momentum hyperbolic partial differential equations can be transferred into a total differential equation:

\[
\left( \frac{\partial Q}{\partial t} + \lambda a^2 \frac{\partial Q}{\partial x} \right) + \lambda gA \left( \frac{\partial H}{\partial t} + \frac{1}{\lambda} \frac{\partial H}{\partial x} \right) + \frac{f}{2DA} Q|Q| = 0
\]

(3)

Defining \(1/\lambda = dV/dt = \lambda a^2\) and considering that the flow velocity is much smaller than the wave speed, the total differential equation for transient flow in pressurized pipeline can be written as positive and negative characteristic equations [15]:

\[
\begin{align*}
\frac{dQ}{dt} + \frac{gA \, dH}{dt} + \frac{f}{2DA} Q|Q| &= 0 \\
\frac{dx}{dt} &= a
\end{align*}
\]

and

\[
\begin{align*}
\frac{dQ}{dt} \cdot \frac{gA \, dH}{dt} + \frac{f}{2DA} Q|Q| &= 0 \\
\frac{dx}{dt} &= -a
\end{align*}
\]

(4)

By modeling the pump-turbine unit, valve, reservoir and other elements, the boundary conditions can be determined and make the governing equations closed for solution. In this study, the initial condition of 1D simulation was determined by water level and other parameters of hydraulic system. The water level of upper reservoir and lower reservoir were treated as unchanged because of the large storage capacity. The pump-turbine characteristics were predicted by 3D simulation instead of using the static characteristic curves. The time step was set as 0.005 s in the entire load rejection process of 40 s.

2.2. Method of 3D Turbulent Flow Simulation

The 3D simulation in this study is based on the Reynolds averaged Navier-Stokes (RANS) equations. It is also combined with the continuity equation and momentum equation with consideration of turbulence effect. The averaged component and pulsating component are respectively modeled. The time-averaged continuity equation and momentum equation can be written as:

\[
\frac{\partial u_i}{\partial x_i} = 0
\]

(5)
\[
\rho \frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p + \nu \nabla^2 \mathbf{u} - \rho \mathbf{u} \mathbf{u} \cdot \nabla \mathbf{u} \quad (6)
\]

where \( u \) is velocity, \( t \) is time, \( \rho \) is density, \( x \) is coordinate component, \( \delta_{ij} \) is the Kroneker delta, \( \mu \) is the dynamic viscosity. \( \mathbf{u} \) and \( \phi' \) are the averaged and fluctuating component. \( \bar{S}_{ij} \) is the mean rate of strain tensor. Term \(-\rho \mathbf{u} \cdot \nabla \mathbf{u} \) is called the Reynolds stress.

In this case, the SST \( k-\omega \) model which is an eddy viscosity model was applied to close the RANS equation [16]. Based on the turbulence isotropy assumption, the eddy viscosity \( \mu_e \) is introduced to build the relationship between Reynolds stress and the mean rate of strain tensor as:

\[
-\rho \mathbf{u} \cdot \nabla \mathbf{u} = 2\mu_e \bar{S}_{ij} - \frac{2}{3}k \delta_{ij} \quad (7)
\]

where \( k \) is the turbulence kinetic energy. The SST \( k-\omega \) model is widely used turbulence model in engineering simulations. It can accurately simulate both the shear flow and adverse pressure gradient by zonally blending the \( k-\omega \) mode and \( k-\varepsilon \) mode. The turbulence kinetic energy \( k \) equation and specific dissipation rate \( \omega \) equation can be written as:

\[
\frac{\partial (\rho k)}{\partial t} + \nabla \cdot (\rho \mathbf{u} k) = P - \rho k^{3/2} l_{k-\omega} + \frac{\partial}{\partial x_i} \left( \mu_k + \sigma_k \mu_t \right) \frac{\partial k}{\partial x_i} \quad (8)
\]

\[
\frac{\partial (\rho \omega)}{\partial t} + \nabla \cdot (\rho \mathbf{u} \omega) = C_{\omega} P - \beta_k \rho \omega^3 + \frac{\partial}{\partial x_i} \left( \mu_t + \sigma_\omega \mu_t \right) \frac{\partial \omega}{\partial x_i} + 2(1-F_1) \rho \sigma_{\omega^2} \frac{\partial k}{\partial x_i} \frac{\partial k}{\partial x_i} \quad (9)
\]

where \( l_{k-\omega} \) is the turbulence scale that \( l_{k-\omega} = k^{1/2} \beta_k \rho \omega \). Term \( P \) is the production term, \( C_{\omega} \) is the coefficient of the production term, \( F_1 \) is the blending function, \( \sigma_k \), \( \sigma_\omega \) and \( \beta_k \) are model constants.

3. Studied Object

3.1. Basic Parameters

To predict the hydraulic transient, the hydraulic system is modeled based on the Fig. 1 which is the schematic map of the system. Table 1 and 2 shows the parameters of reservoir and pump-turbine unit.

![Figure 1. The schematic map of the hydraulic system](image)

**Table 1.** The parameters of reservoir.

| Water level(m)         | Checked flood level | Design flood level | Normal pool level | Static pool level |
|------------------------|--------------------|--------------------|-------------------|-------------------|
| Upper reservoir        | 744.46 (\( p=0.1\% \)) | 743.79 (\( p=0.5\% \)) | 741.00            | 715.00            |
| Lower reservoir        | 297.49 (\( p=0.1\% \)) | 296.96 (\( p=0.5\% \)) | 294.00            | 266.00            |
Table 2. The parameters of pump-turbine unit.

| Parameter                                    | Unit | Number |
|----------------------------------------------|------|--------|
| Number of unit                               | -    | 2      |
| Rated rotation speed $n_0$                   | rpm  | 430    |
| Flywheel torque of generator $GD_2$          | t-m$^2$ | 6000  |
| Flywheel torque of pump-turbine $GD_2$       | t-m$^2$ | 257   |
| Runner diameter $D_1$                        | m    | 4.16   |
| Rated head $H_r$                             | m    | 430    |
| Rated flow rate $Q_r$                        | m$^3$/s | 79.16 |
| Rated power of turbine mode $N_{Tr}$         | MW   | 306.1  |

3.2. 3D Model and Mesh
To conduct the 3D CFD simulation, the flow domain of pump-turbine is modeled as shown in Fig. 2. It consists of the volute, stay vane, guide vane, runner, draft tube, leakages, seals and local chambers. Based on the leakage modeling, the pressure distribution of runner can be simulated and get the value of axial force. Based on the above three-dimensional model of the flow field, the flow simulation mesh is divided. Both tetrahedral unstructured mesh elements and hexahedral structured mesh element are adopted. The number of components and total grids is shown in Table 3.

Table 3. The parameters of mesh.

| Part            | Mesh Number |
|-----------------|-------------|
| Volute          | 150796      |
| Stay vane       | 417420      |
| Guide vane      | 672040      |
| Runner          | 2185219     |
| Draft tube      | 174411      |
| Other parts     | 3210154     |
| Total           | 6810040     |

Figure 2. Schematic map of pump-turbine flow domain

3.3. CFD Setup
In this study, the initial condition of 3D simulation was also determined by water level and other parameters of hydraulic system. Data from 1D simulation were transferred onto the volute and draft tube boundaries. All the solid walls were set as no-slip type. General grid interfaces were set to connect and transfer data between each two subdomains. The time step was set as 0.001 s which is one fifth of the
time step in 1D flow simulation. 200 steps were taken for each runner revolution and the maximum iteration step was set as 60. The convergence criterion was set that the RMS residual of continuity equation and momentum equation should be less than $1 \times 10^{-4}$.

4. Results and analysis

4.1. Load-Rejection Process

In this study, two units rejecting together are simulated within 40 s. The hydraulic transient process of all the transient parameters are shown in Fig. 3. All the parameters are in relative value that $Q^* = \frac{Q}{Q_d}$, $H^* = \frac{H}{H_r}$, $n^* = \frac{n}{n_0}$, $A^* = \frac{A}{A_{opt}}$. $Q_d$ is the rated flow rate of 79.16 m$^3$/s, $H_r$ is the rated head of 430 m, $n_0$ is the rated rotation speed of 430 r/min, $A_{opt}$ is the optimal guide vane opening angle of 30 degrees.

As shown in Fig. 3, the guide vane opening angle is unchanged within 0~10 s, closes within 10~33 s and keep closed within 33~40 s. In this process, the relative rotation speed $n^*$ increases in 0~8 s from 1.0 to 1.4. Then, $n^*$ pulses in 8~26 s and decreases to 0.8 until $t=40$ s. The relative flow rate $Q^*$ goes into the first decrease-increase process in 0~21 s and into the second decrease-increase process in 21~22.5 s. After that, $Q^*$ keeps 0 until $t=40$ s. The relative head $H^*$ complicately pulses and its pulsation frequency accelerate with time. The amplitude of $H^*$ pulsation is about 0.7~1.4. It is about -30%~+40% against the rated head value. Based on the variation law above, the hydraulic parameters strongly vary and may cause instability of the pump-turbine unit. The axial force will be affected by the hydraulic transient process.

4.2. Load-Rejection Process

Based on the 3D simulation, different time steps are selected to analyze the axial force of pump-turbine during load-rejection process. Figure 4 shows the variation of relative axial force $F_z^* = \frac{F_z}{\rho g m}$ where $F_z$ is the axial force, $m_t$ is the weight of the entire shaft system. The negative value represents the upward force. It can be seen that the axial force pulses negative-positive in the load-rejection process. In 0~12 s, $F_z^*$ varies from -0.49 to 0.23. In 12~22 s, $F_z^*$ varies from 0.23 to -0.21. In 22~40 s, $F_z^*$ decreases gradually to 0 and become stable.

After comparing the axial force variation with the hydraulic parameter variation, a strong relationship between axial force and flow rate can be obviously found. When flow rate decreases, the upward axial force becomes downward. When flow rate increases, the downward axial force becomes upward. When flow rate becomes 0, the axial force diminishes. Thus, the flow rate change can induce the change of internal flow regime. At this time, the pressure in crown and shroud leakages changes and may cause the variation of axial force. In this case, $F_z^*$ is always less than 1. It means that the total axial force of shaft system is downward. The pulsating downward force will generate over-load on thrust bearing and upper bracket. There will be security problems.

![Figure 3. Hydraulic variation in load-rejection](image1)

![Figure 4. Axial force variation in load-rejection](image2)
4.3. Internal Pressure
Choosing three typical time steps that \( t = 0, 12 \) and \( 22 s \) for a better analysis, the pressure contour is shown in Fig. 5. The definition of \( \Delta p \) is the pressure difference between crown leakage chamber and internal-runner near outlet. In this area, pressure is usually unbalanced and forms axial force on runner.

![Figure 5. Internal pressure distribution in runner and leakages](image)

Seen from Fig. 5, the \( \Delta p \) is negative when \( t = 0 \) s. The axial force at this time is upward (negative \( F_z \)). At \( t = 12 \) s, the \( \Delta p \) becomes positive and the axial force is downward. At \( t = 22 \) s, the \( \Delta p \) becomes negative again and the axial force becomes upward again.

5. Conclusion
In the load-rejection process, the hydraulic parameters are transient and unstable in pump-turbine. Flow rate and head will vary with the rotation speed and guide vane opening angle. The axial force changes with flow rate and the internal pressure distribution. The pressure difference between crown leakage chamber and internal-runner near outlet pulses and becomes main reason of axial force variation. This study will be helpful for pump-turbine units and other water turbine units in reducing the large axial force and enhance the operation stability.

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