Dynamic characteristics of a pump-turbine during hydraulic transients of a model pumped-storage system: 3D CFD simulation

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Abstract: The runaway process in a model pumped-storage system was simulated for analyzing the dynamic characteristics of a pump-turbine. The simulation was adopted by coupling 1D (One Dimensional) pipeline MOC (Method of Characteristics) equations with a 3D (Three Dimensional) pump-turbine CFD (Computational Fluid Dynamics) model, in which the water hammer wave in the 3D zone was defined by giving a pressure dependent density. We found from the results that the dynamic performances of the pump-turbine do not coincide with the static operating points, especially in the S-shaped characteristics region, where the dynamic trajectories follow ring-shaped curves. Specifically, the transient operating points with the same $Q_{11}$ and $M_{11}$ in different moving directions of the dynamic trajectories give different $n_{11}$. The main reason of this phenomenon is that the transient flow patterns inside the pump-turbine are influenced by the ones in the previous time step, which leads to different flow patterns between the points with the same $Q_{11}$ and $M_{11}$ in different moving directions of the dynamic trajectories.

1. Introduction

The dynamic characteristics of a hydro-turbine are the operating trajectories that it performs during the transients in the hydropower system. When analyzing the real-life situations, we always represent the turbine’s dynamic characteristics by the hill charts obtained from the scaled model tests in static operating conditions. This substitution ignores the influence of transient flow on the turbine performance and its validity should be demonstrated carefully.

The dynamic characteristics of the hydro-turbines have been investigated by several researchers. Yamabe[1] measured the characteristics of a pump-turbine by giving the transient discharge at a constant speed and found that there exists two different characteristics for the same discharge in the
different moving directions of the transient trajectories. Nielsen[2] studied the runaway processes of a Francis turbine by model test and pointed out that the dynamic trajectories appear curly around the runaway point, which differ from their static characteristics. Olimstad et al.[3] found that the transient trajectory does not match the static operation points when they measuring the hill chart of a pump-turbine. Yin[4] simulated the runaway process of a pump-turbine by the 3D (Three Dimensional) CFD (Computational Fluid Dynamics) and the results showed that the dynamic trajectory runs through a damped spiral ring in the region of S-shaped characteristics. Liu et al.[5] compared the dynamic trajectories of a pump-turbine with different rotational inertia; the results illustrated that smaller inertia could lead to larger rings in dynamic trajectories. Therefore, from the above reports we can conclude that the dynamic characteristics of a hydro-turbine always show curly feature and by no means follows its static operating points. Nielsen[2] suggested that this phenomenon is mainly due to the hydraulic inertia in the dynamic head of the turbine. This opinion just takes account of the outer performance and ignores the influence of transient flow on the inner flow patterns of the turbine. Dorfler et al.[6] believed that transient flow patterns in the pump-turbine are influenced by their previous conditions, but he did not give any flow patterns evidences.

The purpose of this study is to investigate the relationship between the dynamic performance and the inner flow patterns of a pump-turbine in a given rotational inertia. Firstly, the runaway process in a model pumped-storage system was simulated by coupling 1D (One Dimensional) pipeline MOC (Methods of Characteristics) equations with a 3D pump-turbine CFD model, in which the water hammer wave in the 3D zone was defined by giving a pressure dependent density. Then, the dynamic outer performance and inner flow patterns were given and analyzed.

2. Numerical methods
Two issues must be addressed so as to simulate the 3D hydraulic transients in a pump-turbine accurately: coupling the dynamic responses in 1D pressure pipes to the local flow patterns in the 3D pump-turbine and modeling the water hammer wave in the 3D zone.

2.1. 1D-3D coupling method
The key problem for 1D-3D coupling is to solve the flow variables on the interfacial boundary. At present, there are several coupling methods, among which the partly overlapped coupling (POC) method[7] is physically reliable and straightforward in programming, as described below.

Assume a simple system consisting of a 1D MOC domain upstream and 3D CFD domain downstream (see figure 1) where the 3D grid and 1D grid are overlapped within a single 1D grid length. We use a section of CFD grid (section 1-2) to model a MOC grid (section PM) in order to obtain the discharge and pressure on the boundaries of the two parts simultaneously. At time step $i$, the flow variables are known at all grid nodes, including the 1D and 3D parts. Therefore, for the 1D part, the $C^+$ equation, eq. (1), can be constructed between nodes $S$ and $P^{i+1}$, which gives a relationship between the discharge $Q_p^{i+1}$ and piezometric head $H_p^{i+1}$ at time step $i+1$

$$C^+ : H_p^{i+1} = H_s^{i} - B(Q_p^{i+1} - Q_s^{i})$$  \hspace{1cm} (1)

where $B = a / (gA)$, and the head loss is ignored here because the distance between the two nodes is very short.

Similarly, we can construct an auxiliary $C^-$ equation, eq. (2), which also specifies the relationship
between $H_p^{(i+1)}$ and $Q_p^{(i+1)}$

$$C^- : H_p^{(i+1)} = H_M^{(i)} + B(Q_p^{(i+1)} - Q_M^{(i)}). \tag{2}$$

From eq. (1) and (2), the head and discharge at node $P^{(i+1)}$ can be solved easily. The detailed coupling procedures are described as follows.

**Figure 1.** Schematic diagram of the partly overlapped 1D and 3D coupling

- Assume the flow variables in the 3D domain are known at the time step $i$.
- Calculate the average head $H_{2-2}^{(i)}$ and discharge $Q_{2-2}^{(i)}$ on section 2-2 of the CFD grid at the time step $i$.
- Update $H_M^{(i)}$ by $H_{2-2}^{(i)}$ and $Q_M^{(i)}$ by $Q_{2-2}^{(i)}$, and then solve the 1D flow variables using the MOC at the time step $i+1$.
- Construct the $C^+$ eq. (1) between the nodes S and $P^{(i+1)}$ and the $C^-$ eq. (2) between the nodes M and $P^{(i+1)}$, and then combine the two equations and solve for $Q_p^{(i+1)}$ and $H_p^{(i+1)}$.
- Use $Q_p^{(i+1)}$ and $H_p^{(i+1)}$ as boundary condition on the section 1-1 for CFD simulation and obtain the 3D flow variables at the time step $i+1$.

2.2. **3D water hammer model**

To model 3D water hammer, the relationship between density and pressure, namely the state equation, should be properly added to the continuity and momentum equations for a CFD simulation. In the previous study [7], we proposed two methods for modeling 3D water hammer wave within the software FLUENT, and the one by defining a pressure dependent density has clear physical meaning and has been proved to be stable and accurate. The final expression of the density is:

$$\rho = \rho_0 e^{\frac{p - p_0}{\rho_0 c_p}} \tag{3}$$

FLUENT provides a user-defined density function, which enables the user to customize the density as a function of other flow variables. Here we take eq. (3) as the density function and solve it explicitly after the pressure $p$ has been obtained at every step.

3. **Simulation and analysis**

We chose the runaway process which caused by sudden load rejection with failed servomotor as the object of study, since from the stationary viewpoint, this process would follow the constant guide vane opening curve in the hill chart.

3.1. **Computational conditions**

The model pumped storage system considered here included one penstock, one pump-turbine, and one tailrace tunnel, see figure 2. In this case, the pump-turbine was simulated by the 3D CFD (using the software FLUENT) while the piping systems were simplified to 1D pipelines and solved by the MOC.
Additionally, the 1D-3D POC method mentioned above was adopted to exchange data between the 1D and 3D zones. Besides, eq. (3) was applied to model the water hammer wave in the 3D zone. The wave speed in the whole system was defined as 1000m/s. During the unit acceleration and deceleration period, the moving mesh model was used in the rotational zone and the angular speed was defined according to eq. (4).

\[ n_{i+1} = \frac{9.55M}{J} \Delta t + n_i \]  

(4)

For 3D pump-turbine, the mesh sensitivity was investigated first and a mesh included about 4300,000 cells was finally chosen. As for turbulence simulation, the \( v^2-f \) model was adopted, for its advantage in simulating flow dominated by separations. The successful application of this model in simulating the characteristics of a pump-turbine can be found in reference [8].

Initially, the pump-turbine was operating in turbine mode with the rotational speed of 1000rpm and the discharge of 0.043m\(^3\)/s.

3.2. Results

When the electrical load was disconnected, the electromagnetic torque dropped to zero immediately and the hydraulic torque was only used to accelerate the runner. Then, the operating parameters began to oscillate, for the instability characteristics of the pump-turbine.

During the oscillations, the operating points of the pump-turbine switched back and forth between the turbine and reverse pump modes and expressed significant periodic persistent oscillations. Figure 3 shows the histories of the rotational speed, discharge, dynamic head, and torque within 10s. Although the evolutions of these four parameters are out-of-phase, they share the same period, which is about 5.5s.

![Figure 2. Schematic diagram of the whole computational domain](image)

![Figure 3. Histories of the operating parameters during the runaway process](image)
from the CFD simulation) in the same guide vane opening, especially in the region of S-shaped characteristics. Figure 4 compares the dimensionless operating parameters between the dynamic and static status. It is obvious that the dynamic trajectories deviate from the static points gradually as $Q_{11}$ or $M_{11}$ decrease and run follow the flat rings around the regions of the S-shaped characteristics. Specifically, during the decrease of $Q_{11}$ or $M_{11}$, the dynamic trajectories move along Path 1 while for the reverse process, they follow Path 2. The existence of these two different paths indicates the non-reversibility of the dynamic performance during the reversible operating process.

Figure 4. Comparisons between the dynamic trajectories and the static operating points

3.3. Analysis of the dynamic characteristics

As mentioned earlier, Nielsen[2] suggested that modifying the working head ($H_m$) by excluding hydraulic inertia from the dynamic head ($H_d$), see eq. (5), can make the dynamic trajectories follow the static curve all the way.

$$H_m = H_d - H_1 \frac{dQ}{dt}$$

Figure 5. Comparisons among the static points, the corrected and original dynamic trajectories

However, for the pump-turbine under study, modifying the working head cannot eliminate the ring-shaped dynamic characteristics. Figure 5 compares the data among the static operating points, the corrected and original dynamic trajectories. It is obvious that the corrected dynamic trajectories are just more flat and closer to the static operating points than the original one. Therefore, the hydraulic inertia is just one reason for the difference between the dynamic and static process; it influences the
shape of the dynamic characteristics, but it is not the primary reason of the ring-shaped curve.

The relationships between $M_{11}$ and $Q_{11}$ among the dynamic, corrected and static conditions are almost the same (see figure 6). Primarily, there is not obvious difference between the dynamic and corrected curves, which indicates that the influence of the hydraulic inertia on $M_{11}$ and $Q_{11}$ can be ignored. Then, although there is a discrepancy between Path 1 and 2, it is relatively small compares to the differences in the $n_{11}$-$Q_{11}$ and $n_{11}$-$M_{11}$ curves (see figure 5).

**Figure 6.** Comparisons of the $M_{11}$-$Q_{11}$ relationships

Therefore, it is easy to conclude that the different outer performances between the dynamic and static operating status of the pump-turbine is mainly reflected in $n_{11}$, for a given rotational inertia.

$$\eta = \frac{D_{M_{11}} n_{11}}{9.55 \rho g Q_{11}}$$

(6)

We can observe from eq. 6 that different $n_{11}$ denote different efficiencies of the turbine in the same $M_{11}$ and $Q_{11}$. Besides, the difference in efficiencies indicates difference in flow patterns. To be specific, larger $n_{11}$ denotes more smooth flow while smaller one denotes more disordered flow for the same $M_{11}$ and $Q_{11}$.

3.4. Flow patterns evidence

To prove the above argument, the instantaneous flow patterns inside the pump-turbine during the discharge decrease and increase processes were investigated. The first point to be focused is the starting point of the ring-shaped trajectory, TP1. We can observe that the flow in the vane channels are extraordinarily smooth (see figure 7 (a)) while in impeller channels, there are local flow separations near the inlet (see figure 8 (a)). With the rotational speed increase, the dynamic trajectory moves to the first runaway point, TP2. At this moment, the flow patterns in the vane channels are similar to those in TP1 (see figure 7 (b)). Besides, the flow separations in the impeller channels expand from the inlet to the middle part(see figure 8 (b)) and almost every channel is blocked partially, about half of the section, by flow separations (see figure 9 (a)). Then, with the decrease of the discharge, the trajectory comes to the first zero discharge point, TP3. It is obvious that large scale vortices generate in guide vane channels and block almost the whole channels (see figure 7 (c)). Meanwhile, there generates two obvious recirculation zones in impeller channels (figure 8 (c) and figure 9 (b)). Not long after that, the trajectory reaches the maximal negative discharge point TP4. We can find that the impeller channels are dominated by reverse flows (see figure 8 (d)) which leads to more intensive vortices in the vane channels (see figure 7 (d)). Additionally, transverse flow develops between the guide and stay vanes,
which disconnects the channels between guide vanes and their corresponding stay vanes. After this, the discharge begins to increase and leads the dynamic trajectory returns to the zero discharge point, TP5. Obviously, there are similarities in flow patterns in the vanes region between this point and the previous point, TP4 (see, figure 7 (e)). Meanwhile, three visible recirculation zones develop in each impeller channel (see figure 8 (e) and figure 9 (c)). Compare to the previous zero discharge point TP3, this point performs poorer in flow patterns. Subsequently, the trajectory enters into the second runaway point, TP6. We can see that separation vortices developed in the wake of the stay vanes move downstream and attach to the guide vanes (see figure 7 (f)). Moreover, in the impeller channels, about two thirds of the cross sections are blocked by flow separations (see figure 8 (f) and figure 9 (d)). Compare to the previous runaway point TP2, this point also show more disordered flow patterns. Finally, the discharge reaches its maximal data and the dynamic trajectory returns to the starting point of the ring. At this moment, the streamlines return to smoothness again in vanes (see figure 7 (g)) and impeller regions (see figure 8 (g)). Furthermore, the flow patterns are similar to the ones in TP1, even though there are a few more intensive flow separations in the pressure side of the blades.

From the above investigation, we found that the flow patterns in the pump-turbine perform differently in the same \( Q_{11} \) or \( M_{11} \) in the different moving directions of the dynamic trajectories. Specifically, the flow is more disordered in vanes and impeller regions during the dynamic trajectories moving from the reverse pump mode (more disordered flow patterns) to the turbine mode (more smooth flow patterns) than those in the reverse process. The reason would be attributed to that the transient flow patterns are influenced by not only the present flow parameters (pressure and discharge), but also the flow patterns in the previous time step, which is similar to the viewpoint of Dorfler et al.[6].

In summary, the transient flow patterns in the pump-turbine are influenced by the previous flow conditions, which is different from those in the static operating status. This phenomenon leads the dynamic trajectories moving along the ring-shaped curves and is the main cause of the difference between the dynamic and static characteristics.
Figure 7. Instantaneous velocity streamlines in the middle section of the stay and guide vanes channels.
Figure 8. Instantaneous relatively velocity streamlines in the middle section of the impeller channels

Figure 9. Instantaneous 3D streamlines in the impeller channels

4. Conclusions

In this study, we have simulated the runaway process of a pump-turbine by a 1D-3D coupling approach and analyzed the dynamic characteristics. The conclusions are as follows:

- The dynamic trajectories of the pump-turbine under study follow ring-shaped curves in the region of S-shaped characteristics. To be specific, the points with the same $Q_{11}$ and $M_{11}$ in different moving directions of the dynamic trajectories give different $n_{11}$.
- The hydraulic inertia influences the shapes of the dynamic trajectories, but it is not the primary reason of the ring-shaped curves.
- The transient flow patterns inside the pump-turbine are influenced by the previous flow
conditions, which leads the dynamic trajectories move along the ring-shaped curves and is the main cause of the difference between the dynamic and static characteristics.

The following work should be done in the future:

- More parameters should be studied to analysis the difference between dynamic and static conditions comprehensively.
- Flow visualizations should be done in model test to validate the conclusions in this study.

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Nomenclature

- \( a (a_0) \) (m/s): acoustic wave speed (in the initial state)
- \( A \) (m\(^2\)): cross-section area
- \( D_1 \) (m): runner diameter
- \( H \) (m): head
- \( H_m \) (m): corrected head
- \( I \) (s\(^2\)/m\(^2\)): geometric factor of inertia
- \( M \) (N\(\cdot\)m): hydraulic torque
- \( n \) (rpm): angular rotational speed
- \( p(0) \) (Pa): pressure (in the initial state)
- \( Q \) (m\(^3\)/s): unit discharge
- \( \rho \) (kg/m\(^3\)): water density (in the initial state)
- \( \rho(0) \) (kg/m\(^3\)): water density (in the initial state)
- \( \eta \) (efficiency)
- \( \Delta t \) (s): time step size

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