Numerical analysis of rotor-stator interaction in a high-head pump-turbine at turbine mode

Hui Ruan¹*, Weili Liao, Like Wang, Yaping Zhao, Xingqi Luo

Abstract
Rotor stator interaction (RSI) between runner blades and guide vanes in pump-turbines may cause unstable vibrations and serious damage to hydraulic system. In order to investigate this problem, five turbine operating points with different guide vane openings were selected for three dimensional calculations of steady flow and unsteady flow, and the water blocking phenomenon caused by flow separations on the pressure sides of runner blades and formed by RSI has been analyzed. The obtained results show that under the same unit speed conditions, the smaller the guide vane opening, the more obvious the flow separations on the pressure sides of the runner blades, the larger range of water blockage was formed by RSI. Flow separations on the pressure sides of the runner blades in certain negative incidence range would not formed water blockage under the effect of RSI, and the range of negative incidence was closely related to the guide vane opening. The forming of water blockage had obvious periodicity, and rotated with the runner. At a runner blade passage, it presented on a water blocking area disappear, the next water blocking area grow, the collapse of water blockage caused impact to the guide vanes, and led to the vibration amplitude of guide vane increased greatly. The primary cause of instability appeared in high-head pump turbine at no-load turbine condition was also obtained.

Keywords
Pump turbine — Rotor-stator interaction — Water blockage

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INTRODUCTION
In practical application, the majority of pump turbines used the same speed to operate under pump and turbine condition. In order to meet the requirement of pump condition, turbine must operate far away from optimum points. Figure 1 shows the synthetic characteristic curves of a high-head pump turbine runner. The unit speed interval of turbine operation was 37.02r/min–39.26r/min, and they were higher than the optimum point with the unit speed 34.01r/min. Therefore the inflow condition of runner inlet was not the unimpact entrance in the turbine mode. In order to investigate this problem, we got the maximum efficiency point (OP1) and three other operation points (OP2, OP3, OP4) from figure1.

Figure 2 analyzes the runner inlet velocity triangles when the conditions are changed. The guide vane opening of OP1 is the same with guide vane opening of OP2, but the unit speed of OP2 is higher than unit speed of OP1. The unit speed of OP2 is the same with unit speed of OP3, but the guide vane opening of OP3 is smaller than guide vane opening of OP2. The unit discharge of OP1 is the same with unit discharge of OP4, but the guide vane opening of OP4 is bigger than guide vane opening of OP1. It is supposed that the inlet flow angle of runner at BEP (OP1) is equal to the blade angle of runner. Therefore the inlet flow angles of runner at OP2, OP3 and OP4 are smaller than the blade angle of runner, they are so-called negative incidences. At this moment, water impact to the suction sides of blades, which lead to flow separations on the pressure sides of runner blades. So, under part load conditions (especially small output conditions) turbine operates in the region that flow separations appeared on pressure sides of blades, and because of the small radical distances between runner inlet and guide vanes outlet, the high rotate speed and the big flow velocity, and the strong interactions, the flow separations on pressure sides of high-head pump turbine runner blades become more complicated under the influence of RSI, and they will have great effect on the stability of hydraulic system.

Figure1 Synthetic characteristic curves of a high-head pump turbine scale model
Due to the unsteady characteristics of rotor-stator interference are bound up with the conversions between mechanical energy and work, how to accurately simulate and predict the rotor-stator interference phenomenon becomes the research hotspot in the field of turbomachinery. Concerning this issue, many numerical simulation methods were presented. For example, R.K.Fisher et al[1] started with fluid excitation, presented and discussed the general equations which characterize RSI pressures and revealed the time variant pressure mode shapes which characterize their dynamic pressure fields. C Nicolet et al[2-3] presented the hydroacoustic modeling[4], simulation and analysis of RSI of a scale model of a Francis pump-turbine and related test rig using a one-dimensional approach. J Yan[5] investigated the influence of water compressibility on pressure pulsations induced by RSI in hydraulic machinery. Christof GENTNER et al[6] performed a numerical study-comprising steady state and unsteady calculations in a pump turbine, and found that a likely cause for runner failure was an impact on every runner blade at nose vane passage which eventually led to fatigue damage in blade-band and blade-crown intersection. Andrej LIPEJ[7] made transient numerical analysis of reversible pump-turbine in pump mode, presented detailed analysis of the unsteady torque on guide vane pivots, and found that the vibrations caused by rotor-stator interaction depend on the discharge and the inlet angles of the guide vanes in the pump mode. L. He et al[8] proposed a space-time gradient method for efficient and accurate unsteady flow analysis of blade row interactions. REN Yuxin[9] proposed an efficient implicit time marching scheme in terms of dual time-stepping technique for the simulation of three dimensional viscous rotor-stator interactions in turbomachinery. XIAO Huimin[10] applied sliding mesh to investigate the rotor-stator coupling of a Francis turbine, and analyzed the computational results for part load point of operation. It is feasible that numerical simulation methods were applied to investigate the rotor-stator interference.

1. MODELING AND MESHING
A scale model of high-head pump turbine was simulated, and the major parameters are shown as follows: the diameter of runner at pump inlet is 454.33mm, the diameter of runner at turbine inlet is 1000 mm, the height of guide vane is 69.78mm and the scale between prototype and model is 4.27:1.

This paper presented the modeling, simulation and analysis of rotor-stator interaction of a high head Francis pump-turbine based on commercial software CFX. The computational modeling of the high head pump-turbine took into account the spiral casing, the 20 stay vanes, the 20 guide vanes, the 7 rotating runner vanes and the draft tube, as shown in figure 3.

A hexahedral grid was generated for the model pump-turbine by decomposing the structured blocks. The pressure value of a monitoring point located between guide vane and runner was the monitoring object, a grid consistency research with five different densities was performed at the best efficiency point (BEP), as shown in figure 4. The structured hexahedral mesh with approximately 7.93 million elements meets the accuracy requirement. Figure 5 shows the mesh distribution of each part, the mesh element is 0.74 million, 3.07 million, 3.55 million and 0.57 million for spiral casing, guide vanes, runner and draft tube, respectively.

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**Figure 2** Velocity triangle of runner inlet

**Figure 3** Computational domain

**Figure 4** Grid independence test

**Figure 5** Mesh distribution
2. NUMERICAL CALCULATION METHOD

Numerical calculations of the internal flow field of pump-turbine were based on the continuity equation and momentum conservation equation for the isothermal compressible flows of a Newtonian fluid. By the Reynolds decomposition and time averaging of the Navier-Stokes equations, we got the RANS equations. To close the equations, the Reynolds stress tensor was approximated by the eddy viscosity model, and the shear stress transport (SST) turbulence model was used to solve the eddy viscosity coefficient. Finite volume methods were applied to discrete the governing equations, the convective terms were approximated by the first-order upwind differencing scheme, the diffusive terms were approximated by the central differencing scheme, the temporal domain was discretized by the time step, corresponding mathematical equations and experience coefficient can be found in reference11. The boundary conditions were set as follows: at the inlet plane, the mass flow rate with normal flow direction was specified, and at the outlet plane, pressure condition was applied. All surfaces of pump-turbine components which were in contact to the water passage were defined as no-slip walls. Between rotating and stationary components, frozen rotor interfaces were used for steady state calculations and transient rotor-stator interfaces for unsteady simulations. In our case all unsteady state calculations had been started from the solution obtained by steady state calculation. The time step size was selected such that one runner revolution was divided into 360 time steps corresponding to an angular change of 1.0°. For all computations, 6 revolutions were carried out. The last revolution was used to plot time histories of pressure fluctuations. The MAX residual target was set to 10⁻³ for velocity components and pressure to ensure converged solutions.

In order to confirm the accuracy of numerical simulation method, figure 6 compares the experimental and numerical hydraulic efficiencies at three operating points. The maximum difference between the experimental and numerical efficiencies was observed at a low discharge (guide vane opening α=15mm, unit speed n₁=39.26 r/min, unit discharge Q₁₁=0.291 m³/s) operating condition. The numerical hydraulic efficiency was 1.43% higher than the experimental efficiency. The lowest difference between the experimental and numeric results was 0.19% at the BEP (α=22.5mm, n₁=34.01 r/min, Q₁₁=0.492 m³/s). The difference between the experimental and numerical efficiencies at the high discharge operating point (α=27mm, n₁=38.33 r/min, Q₁₁=0.543 m³/s) was 0.7%. So the numerical simulation results had good consistency with the experimental results, and the feasibility of the numerical simulation method was verified.

![Figure 6](image)

To investigate the flow separations on the pressure sides of runner blades caused by negative incidence under the influence of RSI, this paper got 6 turbine operating points to simulated. The unit speeds were 37.02r/min, the guide vane openings were 9mm, 12mm, 15mm, 18mm, 21mm and 24mm, and the corresponding guide vane angles were 6°, 8°, 10°, 12°, 14° and 16° respectively.

3. RESULTS AND DISCUSSION

Streamline distributions at the inlet of runner were observed, it was found that there were obvious flow separations on the pressure sides of runner blades at 6°, 8° and 10° guide vane opening conditions. In order to further analyze the flow characteristics of the flow separations, monitoring points were arranged, by tracing the inflow and outflow of every monitor points, the streamlines of monitoring points under different guide vane opening conditions were obtained, as shown in figure 7. At 6°, 8° and 10° guide vane opening conditions, water flow through the spiral casing, stay vanes and guide vanes did not smoothly flow into the runner, but disturbed back and forth between guide vanes and runner blades, and obstructed other water flow into the runner, causing seven water blockages in the inlet of runner. With increasing of guide vane angle, at 12° guide vane opening condition, there were two water blockages, and at 14° guide vane opening condition, there was only one water blockage. But at 16° guide vane opening condition, the back and forth disturbance phenomenon disappeared, all water could flow into the runner smoothly, such as shown in figure 7f. Therefore flow separations on the pressure sides of runner blades at 12° and 14° were existed, and had effect on stabilities of turbine at part load conditions.

![Figure 7](image)
Water blockages were caused by flow separations on the pressure sides of runner blades and formed by RSI, so the water blockages possessed unsteady characteristics like RSI. Unsteady simulation at 10° guide vane opening (\(n_1=37.02\, \text{r/min}, \,Q_1=0.311\, \text{m}^3/\text{s}\)) has been achieved. Because it was difficult to use monitoring point tracking streamline method to analyze the results of unsteady calculation, it was expected to find a pressure variable to describe water blocking phenomenon. Through calculation and comparison, it is concluded that a certain value of total pressure (640000 Pa) equivalent distribution fit well with water blockage, as shown in figure 8.

The formation of water blockage is connected with flow separations on the pressure sides of runner blades and rotation of runner, so the investigation of unsteady properties of water blockage can’t in a position of guide vane for the observation point. If observed water blockage in position of a guide vane, the obtained changing law was how the water blockage pass through guide vane, rather than how the water blockage form and develop. Therefore, one blade of runner set as target object, and water blockage changing law was observed when the target blade rotated. It is found that the formation and development of water blockage has periodicity, and the periodic time is 0.01981268 s, in a cycle, the target blade rotates 75°.

Figure 9 shows the changing process of water blockage in a period. At initial time, there was a mature water blockage (denoted as block 1) in the target blade passage, and near the pressure side of target blade, another water blockage (denoted as block 2) began to primary. With the rotation of runner, block 2 grew, at \(t=t_b+6\,\text{T/25 time}\), block 1 started hitting guide vane, at \(t=t_b+12\,\text{T/25 time}\), rotation made block 1 out of guide vane, kept a certain time, at \(t=t_b+12\,\text{T/25 time}\), block 1 hit the next guide vane. When block 1 gradually collapsed due to the hitting, at the same time, block 2 continued to grow. At \(t=t_b+21\,\text{T/25 time}\), block 1 collapsed completely and block 2 was maturely formed, block 2 kept 4T/25 time, next block began to primary. Overall, there was a reciprocal relationship between block 1 and block 2, when the previous block broke

### Table 1 Parameters of highest efficiency points and water blockage formed critical points

<table>
<thead>
<tr>
<th>Guide vane angle(°)</th>
<th>Highest efficiency points</th>
<th>Critical points</th>
</tr>
</thead>
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<td></td>
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<td>Q11</td>
</tr>
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<td>0.500</td>
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</table>

Water blockage formed critical points at 8°, 10°, 12° and 14° guide vane openings have been found, and the unit speeds were higher than unit speeds of the highest efficiency points, the difference was 1.27r/min, 1.37 r/min,1.59r/min and 1.52r/min for 8°, 10°, 12° and 14°, respectively. There was no water blockage at 16° guide vane opening. It was implied that flow separation on the pressure side of the runner blade in certain negative incidence range would not formed water blockage under the effect of RSI. According to the velocity triangles, the diameter of runner at turbine inlet was 1000 mm, and compared with the inlet flow angles of runner at highest efficiency points, there was -2.58°, -2.95°, -3.11° and -3.17° allowance for 8°, 10°, 12° and 14°, respectively.
down, the after block formed. The collapse of previous block could cause impact to multiple guide vanes because of runner rotation.

The collapse of water blockage caused impact to the guide vanes, just because of the rotation of runner, the guide vane and the position of impact had been changing. So, in order to analyze the effect of impact on the vibration characteristics of guide vane, according to the impact position shown in figure 9, three monitoring points were selected, MPT1 was set up in the tailing edge of guide vane, MPT2 was set up in the middle between two guide vanes, MPT3 was set up in the suction side near the tailing edge of adjacent guide vane, the plane layouts of the measuring points is shown in figure 10, and the height of the plane is 0.5 times the height of guide vane.

Because the vibration of high-head pump turbine is complex, in order to distinguish the impact frequency of water blockage to guide vane from other vibrational frequency, the operating point without water blockage phenomenon was selected as reference and denoted as operating 2, the operating point with water blockage was denoted as operating 1. Figure 11 shows the time-domain contrast diagram of pressure fluctuations about MPT1, MPT2 and MPT3, The pressure fluctuations of monitoring points had obvious periodicity, and the amplitude of pressure fluctuation in operating 1 was higher than that in operating 2. Frequency-domain contrast diagrams of pressure fluctuation were obtained by Fourier transformation, as shown in figure 11d, figure 11e and figure 11f about MPT1, MPT2 and MPT3 respectively. The main frequencies were blade passing frequency \(7f_n\), \(7\) is the blade number, and \(f_n\) is the rotating frequency, but the vibration amplitudes of monitoring points in...
operating 1 were higher, the vibration amplitude of MPT1 was 4.41 times to vibration amplitude in operating 2, 4.4 times of MPT2, and 3.77 times of MPT3. It is illustrated that the water blockage phenomenon has strong effect on the vibration of guide vane.

At the same time, it can be seen that under operating 1 condition, there were $f_n$ frequency pressure pulsations of monitoring points, which did not appear under operating 2 condition. The $f_n$ frequency was associated with the forming cycle of water blockage. In a forming cycle, the target blade rotated 75 ° but in a blade passing frequency cycle, the blade rotated 51.43 °. In the perspective of a certain guide vane, the above phenomenon indicated that the impact law of the target blade rotate through the guide vane was different from the impact law of the next blade rotate through the guide vane. If the impact laws of 7 blades of runner were not the same, the $f_n$ frequency pressure pulsation would be formed. Under operating 2, there was no such problem that whether the water blockages impact to the guide vanes matched each other or not, so there was no $f_n$ frequency pressure pulsation.

The above analysis of $f_n$ frequency pressure pulsation was from the positive viewpoint, the following analysis of the vibration phenomenon is from the opposite side. In the perspective of a certain guide vane, if in a forming cycle of water blockage, the target blade exactly rotates 51.43 ° , when the next blade rotates through the guide vane, the impact law and occasion of water blockage is the same with that of target blade. If all the blades keep in step with the target blade, the water blockages will be up to the maximum, and even make the water into pump operating mode, and cause strong high frequency pressure pulsation and mechanical vibration. This inference well explains the primary cause of instability appeared in high-head pump turbine at no-load turbine condition.

4. CONCLUSION

1) From the steady simulations and analysis, it can be seen that under the same unit speed conditions, the smaller the guide vane opening, the more obvious the flow separation on the pressure side of the runner blade, with the effect of RSI, the larger range water blockage is formed.

2) A water blockage appearing as the judgment standard, the critical operating points of water blockage formed at different guide vane opening were obtained. The results show that the flow separation on the pressure side of the runner blade in certain negative incidence range will not formed water blockage by RSI, and the range of negative incidence is closely related to the guide vane opening.

3) From the unsteady simulations and analysis, it is obtained that the forming of water blockage has obvious periodicity, and rotates with the runner. At a runner blade passage, it presents on a water block area disappear, the next water block area grow, and the collapse of water blockage causes impact to the guide vanes. Through the contrast analysis of the pressure pulsations of monitoring points on the guide vanes between the condition with water blockage and the condition without water blockage, the water blocking phenomenon causes the vibration amplitude of guide vane increased greatly. Analysis of the vibration frequency from the opposite side, the primary cause of instability appeared in high-head pump turbine at no-load turbine condition is obtained.

ACKNOWLEDGMENTS

So long and thanks for all the supports from the National Natural Science Foundation of China (51179152,51379174), Fund for the Key Disciplines Construction of Higher Education Institutions in Shaanxi Province and the “13115” Engineering Technology Research Center of Shaanxi Province.

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