Pressure pulsation in small opening operating mode of pump turbine’s turbine mode

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Abstract. The units of pumped-storage power station always have a serious instability when the pump turbine working at transition process of turbine mode. Both the steady and unsteady numerical simulation were carried out to simulate the inner flow in pump-turbine. When considering and not considering the weak compressibility of water respectively, the internal flow field of two kinds of turbine operating mode and pressure fluctuation of a small opening condition of turbine operating mode are analyzed. The results show that the "water seals", a plurality of annular flow area which appeared in vaneless zone, under the small opening during turbine mode and those water seals blocked inflow from the guide vane domain. A pressure pulsation, which dominant frequency is 7fn, has emerged in the vaneless space under the small opening condition during the turbine mode. The amplitude of the pressure pulsation of the case which considered the compressibility is slightly larger than the others. Additionally, a pressure pulsation, dominant frequency is rotating frequency, has occurred in draft tube. And this pressure fluctuation is related to the structure of spiral vortex. Furthermore, when the compressibility is considered, there is a pressure pulsation occurred at the main frequency of 10fn in the draft tube. Meanwhile, the spiral vortex zone becomes heavier.

1. Introduction
The pump turbine will show strong instability at small flow rate[1]. Due to the inherent characteristics of the pump turbine, the blade inlet impingement and the blade outlet have vortex bands, which makes the draft pipe pressure pulsation larger. In the small flow area, the high pressure side flow's separation and dead water zone are formed. When these conditions are running, the pressure pulsation amplitude before the runner is increased sharply. The changes of internal flow field and the pressure pulsation, caused by rotor-stator interaction, of vaneless and draft tube under small flow rate condition is directly related to the stability of the unit. A series of problems caused by the pump turbine in small flow rate condition in the industry has aroused great concern of pumped storage power station host equipment stability and vibration problems.

The Swiss Nicolet.C[2] analyzed the influence of static phase resonance at the junction of the amplitude of pressure fluctuation in the calculation, and to capture the standing wave generated at the scroll. Vlad Hasmatuchi[3] has found that from the turbine to runaway conditions, the static pressure pulsation of guide leaves has an obvious increasing trend by numerical simulation. R. Blanc-Coquand[4] et al study the reversible turbine operating conditions and pump working condition by unsteady CFD calculation, analysis the influence of the rotating parts of the number of leaves, the size of the flow and the vaneless area on rotor-stator interaction. Gunnar Backman[5] uses 3D unsteady flow
calculation to study the rotor-stator interaction phenomena by analyzing the flow field, pressure fluctuation amplitude and frequency.

Xu Yu,[6] et al obtained the internal flow of the pump in the reversible water turbine. Liao Weili, [7] et al made a thorough analysis of the pressure pulsation of the hydraulic turbine. The large eddy simulation (LES) of single channel flow in reversible pump turbine runner has been successfully realized by Xu LAN[8] and the calculation results are in good agreement with the existing experiments. Low specific speed of pump turbine small flow rate of turbine condition of pump turbine was calculated by Ji Xingying[9], obtained the "S" curve, and compared with the experimental results, the "S" characteristics in turbine is low head start working condition have a direct relationship with the emergence of annular flow.

Most of the numerical simulation of pump turbine without considering the water weak compressibility. For high head pump turbines, the density of water in high pressure zones can not be reduced to a constant, especially considering cavitation factor, the weak compressible type of water can not be neglected.

Chen Shifan[10] through the establishment of the weakly compressible fluid model of three-dimensional simulation, the pumped storage power station load rejection, the results show that considering the fluid compressibility of the case, the numerical results is closer to the experimental data than incompressible fluid model. Yan[11] et al use the compressible model to simulate the rotor-stator interaction between the runner and the guide vane of the pump turbine. It is found that the results of the weakly compressible model are closer to the experimental ones.

2. Mathematical model and computational details

**Mathematical model**

The continuity equation and momentum equation are adopted in simulation.

**Continuity equation**

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \]  \hspace{1cm} (1)

**Momentum equation**

\[ \rho \frac{D \mathbf{u}}{Dt} = \rho f - \nabla p + \mu_{eff} \nabla^2 \mathbf{u} + (\mu_{eff} + \lambda) \nabla (\nabla \cdot \mathbf{u}) \]  \hspace{1cm} (2)

**Weakly compressible equation**

In the calculation process, the density of water is determined according to the acoustic pressure equation:

\[ \rho_1 = \rho_0 + (P - P_0)/a^2 \]  \hspace{1cm} (3)

Where: \( \rho_1 \) the actual density of water; \( \rho_0 \) the reference density for water; 998kg/m3; \( P_0 \) for reference pressure; 101325 Pa; \( a \) for sound speed; 1400 m/s.

3. Computational model and boundary conditions

**Computational model**

A full flow geometry model is established for a high head pumped storage power station’s pump turbine in china. The calculation range of the whole flow passage of the model pump turbine consists of 5 parts, namely, spiral case, stay vane, guide vane, runner and draft tube. As shown in Figure 1:
Grid partition
Mesh model by using ICEM software, meshing process, all components are used hexahedral structured grid, grid refinement in all parts of the interface and near the wall, make the boundary layer grid, which can guarantee the high quality mesh, but also reduces the time of numerical simulation. The number of grid units in the whole model is 5 million 800 thousand, and the computational domain grid is shown in figure 2.

Figure 1. Hydraulic model of computational domain

Figure 2. full flow passage grid and Stay vane - guide vane - runner mesh

calculate boundary conditions and operating point parameters
The medium density is iteratively calculated by the weakly compressible equation. And the Reynolds averaged Navier-Stokes equation are used to simulate the flow of the pump turbine, The SST k-Omega turbulence model is used to seal the equations. Given that the inlet boundary condition of the spiral case is mass flow inlet, the outlet of the draft tube is the pressure outlet, and the no slip boundary condition is adopted at the wall. In order to make the interpolated velocity component and turbulent flow as well as the integral pressure and flow flux, the interface of the adjacent computational domain is set to “interface”. the physical time step of steady numerical simulation is $\Delta t = \frac{1}{\omega}$ ( $\omega$ is the wheel rotation speed), The unsteady calculation is performed under the initial conditions of the convergence of the steady computation the physical time step is $\Delta t = \frac{1}{360T}$ (T is a rotation period at the corresponding operating point) That is, the time required for each wheel to rotate at 1 degrees is taken as a time step, and each step is iterated 10 times, The transient calculation, the rotation region and the static region interface mode are selected and solved by Transient Rotor Stator model.

Select 2 different gateopening turbine for steady numerical simulation, The specific working condition parameter is: The Working Condition 1: the guide vane angle $\theta = 10^\circ$, unit speed $n_{11}=37r/min$, Unit flow $Q_{11}=0.320$ m$^3$/s; The Working Condition 2: the guide vane angle, $\theta = 18^\circ$, unit speed $n_{11}=38.33r/min$, Unit flow $Q_{11}=0.61$ m$^3$/s. The unsteady numerical simulation of The
Working Condition 1 and the steady numerical simulation of The Working Condition 2 is taken into account the weak compressibility is considered and not considered separately.

4. Calculation result analysis

based on unsteady numerical simulation results of 2 different gateopening turbine working condition, The flow field is analyzed, and the streamlined diagram of the middle section of the guide vane and runner is taken as shown in figure 3, When the Working Condition 1 is running, the irregular annular area is formed in the vaneless zone between the guide vane and the runner, and a water sealing ring is formed, When considering the weak compressibility of water, the sealing water area is slightly larger than without consideration, and the flow regime in the channel is also worse; In the Working Condition 2, the streamline is smooth and the flow regime is good. When considering the weak compressibility of water, the flow regime has no obvious change.

Flow field analysis

Figure 3. the streamline distribution in the middle section when not considering (a) and considering (b) the weak compressibility of water
The point tracking method is adopted to extract the streamline of pump turbine, that is 21 measuring points are uniformly arranged at the same height between the guide vane and the runner, the height direction is divided into 3 layers, which are near the upper crown section, the middle section and the lower ring section, a total of 63 measuring points. The inflow and outflow of each measuring point under two working conditions were tracked. In view of the simplicity of the diagram, the 7 most representative streamlines (The number of leaves is 7) are retained, as shown in figure 4. The Working Condition 1, the streamlines moves back and forth to form a seal water area. Compared with The Working Condition 1, The Working Condition 2 flows smoothly.

Figure 5 is the streamline distribution of the draft tube under different working conditions. In the Working Condition 2, the streamline is smooth, the flow regime is good, and the streamline distribution of the Working Condition 1 becomes worse, and the spiral vortex belt is formed. In the two conditions, when considering the weak compressibility of the water, the vortex zone of the draft tube is more obvious than that without consideration. In Working Condition 2, when considering the
Compressibility of the water, a slight columnar vortex zone appears when considering the compressibility of water, without consideration, it does not appear.

To sum up, the Working Condition 1 in the vaneless area, the runner channel and the draft tube flow pattern are very poor, and the water retaining ring is formed in the vaneless area, and a strong spiral vortex band is formed in the draft tube. The working condition 2 is the optimum working condition, and the flow regime is good. Considering the weak compressibility of water, the flow regime will become worse, especially when the Work Condition 1.

Pressure fluctuation analysis
The pressure pulsation monitoring point is located in: between the guide vanes, the suction surface and tail of the guide vane and the vaneless section between the guide vane and the runner; the straight taper section and the bent elbow section of the draft tube. Seven monitoring points are arranged at the same height around the monitoring points, In the height direction, it is divided into three layers, respectively, near the upper crown section, the middle section and near the lower ring section, as shown in table 1 and figure 6.

| Monitoring point name | Location of monitoring points                        |
|-----------------------|------------------------------------------------------|
| gvm                   | Between guide vanes                                  |
| gvout                 | Guide vane outlet                                    |
| gvs                   | Suction surface of guide vane                        |
| gr                    | Vaneless zone                                        |

Figure 6. layout of monitoring points for pressure pulsation

4.2.1. Analysis of pressure fluctuation in guide vane and vaneless region. The data of the point of 24 (2-4) placed on the guide vane, the guide vane exit, the suction surface of the guide vane and the middle section of the vaneless section were analyzed. Pressure fluctuation time diagram and frequency diagram of pressure fluctuation are as follows:
Figure 7. pressure fluctuation time diagram of measuring point between guide vanes

Figure 8. frequency diagram of pressure fluctuation at the middle section of guide vanes

Figure 9. pressure fluctuation time diagram of measuring point of guide vane outlet

Figure 10. frequency diagram of pressure fluctuation at the measuring point of the middle section of tail of guide vane

Figure 11. pressure fluctuation time diagram of suction surface of guide vane

Figure 12. frequency diagram of pressure fluctuation at the measuring point of middle section of suction surface of guide vane

Figure 13. pressure fluctuation time diagram of suction surface point of vaneless zone

Figure 14. pressure fluctuation time diagram of suction surface point of vaneless zone

Figure 7, figure 9, figure 11 and figure 13 are the time domain diagrams of the pressure fluctuation obtained by the data of the measurement point of (point name24) placed between the guide vanes, the outlet of guide vane, the suction surface of guide vane and the vaneless zone. These can be seen that the variation trend of the pressure amplitude of measuring point is basically the same, showing obvious periodic variation law. The closer to the rotation axis is, the more obvious the periodicity is. Considering the weak compressibility of water, the periodicity become poor, but still very obvious. This moment, the maximum amplitude of the pressure pulsation between the guide vanes and the vane outlet is greater than when the compressibility is not considered, the maximum value of the pressure pulsation amplitude at the suction surface of guide vane outlet is smaller. Considering and not considering compressibility, the minimum value is relatively close.
The data of the measuring point of (point name24) is taken from the between guide vanes, guide vane outlet, guide vane suction surface and middle section of vaneless area. The Fourier transform is used to obtain the middle section pressure pulsation frequency-figure 8, figure 10, figure 12 and figure 14.

Without considering the weak compressibility of water, the measurement points of the between guide vanes, guide vane outlet have five frequencies, respectively 3fn, 7fn, 9fn, 14fn, 21fn. The main frequency is 7fn called the leaf times frequency. Considering the weak compressibility of water, there are five frequencies, respectively 3fn, 7fn, 10fn, 14fn, 21fn. The main frequency is 7fn. The composition of the pressure pulsation frequency in the middle section of the guide vane suction surface has a 28fn that have not between the guide vanes and the outlet of guide vane. The main frequency is 7fn. That is to say, whether considering the weak compressibility of water, the leaf times frequency has not affect on the main frequency on the pressure pulsation frequency points. They are all the leaf times frequency. For the pulsating amplitude, regardless of whether or not to consider the compressibility of water, the main frequency amplitude gap is not large. In the guide vane outlet and suction surface, the main frequency amplitude of several points of not considering the weak compressibility of water is slightly higher than considering the weak compressibility of water.

It can be seen that the frequency composition becomes complicated, Without considering the weak compressibility of water, the measurement points of vaneless area have six frequencies, respectively 3fn, 7fn, 14fn, 21fn, 28fn, 63fn. the main frequency is 7fn, Considering the weak compressibility of water, there are six frequencies, respectively 3fn, 7fn, 10fn, 14fn, 21fn, 28fn. the main frequency is 7fn. For the pulsating amplitude, regardless of whether or not to consider the compressibility of water, the main frequency amplitude gap is not large.

4.2.2. Analysis of pressure fluctuation in draft tube.

According to the measured data of the draft tube, the time fluctuation diagram of pressure fluctuation at the point of draft tube figure 15 is obtained. Without considering the weak compressibility of water, there is no regularity in the variation of pressure fluctuation. Considering the weak compressibility of water, the variation of pressure fluctuation is regular, the maximum value of pressure amplitude is bigger and the minimum value is smaller.
According to the measured data, Fourier transform is used to obtain the frequency diagram of pressure pulsation in the draft tube (figure 16). Without considering the weak compressibility of water, there is a frequency of $f_n$, that is turn frequency. Considering the weak compressibility of water, there are three frequency points in the straight cone section, $f_n$, $10f_n$ and $21f_n$ respectively, and the main frequency is $10f_n$. There are three frequencies in the elbow section, $3f_n$, $10f_n$ and $13f_n$, respectively. The main frequency is $10f_n$. For the pulsating amplitude, when considering the weak compressibility of water, the main frequency is several times as much as it is incompressible.

There are some picture(figure 18, 19) to show the vortex in the draft tube under two simulating methods, since the pressure fluctuation results are very different. As shown in figure 17 plane 1 and plane 2 cross the measure point of draft:

![Figure 17. Plane position diagram](image)

![Figure 18. pressure contour of plane 1(a)](image)

![Figure 19. pressure contour of plane 1(b)](image)

![Figure 20. pressure contour of plane 2(a)](image)

![Figure 21. pressure contour of plane 1(b)](image)

As shown in figure 18~21, a single periodic (T) pressure contour change has obtained through the plane of the measured point of the draft tube. Plane 1 without considering the water weak compressible (a), the overall pressure has seldom changed, the maximum difference of measuring points pressure is 8000 Pa. When considering compressibility (b), 1/6 T and 4/6T appeared high pressure distribution, the maximum difference of measuring points pressure is 15000 Pa. Plane 2 has a similar change with plane 1. The maximum difference of measuring points pressure is 6000Pa (a) and
18000Pa (b). It can be seen that when considering compressibility, the pressure pulsation near the measuring point of the draft tube is much stronger than that without considering compressibility.

5. Conclusion
In conclusion, the following conclusions can be drawn:

The small opening condition of turbine operating mode (Working Condition 1) have appeared - "water seal" in Vaneless zone namely in the zone of behind guide vane and before runner appear a plurality of annular flow area, that is mean a obstruction to flow.

Without considering the weak compressibility of water, when the small opening condition of turbine operating mode, the measurement points of vaneless area have a main frequency is 7fn. When the water compressibility is weak, the main frequency near the tongue is 20fn. The remaining main frequency is 7fn. When considering compressibility, the pulse amplitude is slightly higher than that without considering the compressibility. The frequency of the measured points near the vaneless and near the guide vane is 7fn, In the guide vane outlet and suction surface, the main frequency amplitude of several points of not considering the weak compressibility of water is slightly higher than considering the weak compressibility of water. 

Without considering the weak compressibility of water, there is a pressure pulsation in the draft tube, which dominant frequency is the rotating frequency, and there is a spiral vortex band structure. When considering the weak compressibility of water, the main frequency of pressure pulsation in the draft tube is 10fn. At this point, the spiral vortex zone increases and the amplitude of pressure fluctuation increases.

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References
[1] Xiang Gaoming. INVESTIGATION OF PRESSURE FLUCTUATION WITH FLUID-STRUCTURE INTERACTION COSIDERED OF A PUMP TURBINE AT PUMP MODE[D] Harbin Institute of Technology,2015. ( in Chinese)
[2] Nicolet. C, Ruchonner.N. Hydroacoustic simulation of rotor-stator interaction in resonance conditions in Francis pump-turbine[A]. 25th IAHR Symposium on Hydraulic Machinery and Systems[C]. Lausanne, IAHR Symposium, 2010: 9-20.
[3] Vlad Hasmatuchi, Hydrodynamics of a pump-turbine at off-design operation conditions: numerical simulation[C]. Proceedings of ASME-JSME-KSME Joint Fluids Engineering Conference 2011.
[4] R.Blanc-Coquand, S.Lavigne, JL.Deniau. Experimental and Numerical Study of Pressure Fluctuation in High Head Pump-Turbine[C].Hydraulic Machinery and Systems 20th IAHR Symposium.
[5] Backman G. CFD Validation of Pressure Fluctuations in a Pump Turbine[D]. Lulea: Lulea University of technology, 2008.
[6] Xu Yu Tang Xuelin Wu Yulin. Three-dimensional Turbulent Flow Analysis Through a Pump-turbine Runner at Pump Modes [J]. JOURNAL OF HYDROELECTRIC ENGINEERING. 2000(3): 75-83. ( in Chinese).
[7] LIAO Weili JI Jinting LU Peng LUO Xingqi. Unsteady Flow Analysis of Francis Turbine [J]. JOURNAL OF MECHANICAL ENGINEERING. 2009, 45(6): 134-140. ( in Chinese).
[8] XU Lan , CUI Guixiang , XU Chunxiao , ZHANG Zhaoshun , CHEN Naixiang. Large eddy simulation of turbulent flows passage of reversible runner [J]. JOURNAL OF HYDROELECTRIC ENGINEERING. 2007, 26 (4). ( in Chinese).
[9] JI Xing-ying1,2, LAI Xu. Numerical simulation of the S-shaped characteristics of the pump-
turbine [J]. CHINESE JOURNAL OF HYDRODYNAMICS Series A. 2011(03): 318-326. (in Chinese).

[10] Chen Shifan, Zhou Daqing, Chen Yang. Numerical simulation of flow in penstock of pumped storage power station based on weakly compressible flow model [J]. Journal of Drainage and Irrigation Machinery Engineering (JDIME), 2014, 32(9): 765－770. (in Chinese).

[11] Yan J, Koutnik J, Seidel U, et al. Compressible simula-tion of rotor-stator interaction in pump-turbines [C] // Proceedings of 25th IAHR Symposium on Hydraulic Machinery and Systems. Timisoara, Romania: [s. n. ], 2010: 1－8.