Analysis on the internal flow field in vaneless space and draft tube of one reversible pump turbine during load rejection under turbine mode

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Abstract. The development of pumped storage plants presents quicken tendency since the deregulation of energy market, for balancing the unstable grid caused by electricity generated by renewable energy. Whilst pumped storage plants are increasingly operating at off-design conditions in order to meet the requirement of electrical grid, therefore performance of unit at off-design conditions has become more important. In this work, three dimensional unsteady calculations were carried out for simulating pump turbine load rejection period based on the dynamic sliding mesh method and turbulent model of detached eddy simulation, also weakly compressible fluid was considered. The results was analyzed in both time domain and frequency domain to study the internal flow field characteristics, while flow field evolution in vaneless space between guide vane and runner was also investigated, reasons of vortex rope formation and characteristics of vortex rope in draft tube were studied.

1. Introduction
The pumped storage plants operate in a wide range of off-design conditions has become particularly frequent, and this is the main reason induce unit instability [1-2]. During the period of unit operating at off-design conditions, the unsteadiness is amplified and the parameters including pressure, torque and flow rate fluctuations may rise greatly due to the appearance of vortex rope, which will cause serious damage to the whole unit system and shorten the lifespan of unit components [3]. Consequently, the vortex rope study during transient processes is important for the feasibility of a pumped storage power station. Many studies have been done to mitigate or prevent the vortex rope phenomenon in recent years: Additional air is injected in the center of draft tube in Kirschner’s work [4], the experimental results show the reduction of pressure fluctuations in the draft tube; Wang et al. adopted a similar way of injecting an axial water jet in the center of draft tube through the runner hub based on numerical simulations, the results illustrate vortex rope in the draft tube can be positively influenced [5]; Liu et al. analyzed vortex rope in load rejection period, they obtained the runaway speed can reach 1.15 times compared to the initial rotational speed during the transient process and the vortex rope occurs before the pump turbine runs at zero moment point [6].

On the other hand, interaction between stator and rotor is another major reason that cause unit vibration and noise [7], some scholars have done some research on this aspect. Jia[8] worked on
influence of rotor-stator interaction with the cascades match of runner blades and guide vanes, however, during the period of off-design operating conditions, it is impossible to change the number of blades for solving the problem about interaction between rotor-stator, hence a study of this aspect in unit transient process is necessary. In Li’s work[9], a research on similarities of the flow in the rotor–stator interaction affected region was carried out for prototype and model pump turbine, his result illustrates that the relative fluctuation amplitudes in the model are slightly higher than those in the prototype, meanwhile, this research shows Reynolds number affection on the flow discrepancy and pressure fluctuation difference in the rotor-stator affected region will facilitate better estimations of pump turbine performance from model to prototype. Sun et.al simulated pump turbine steady performance with different radial gaps between runner tips and volute tongue, their results explain there is an optimal radial gap for one pump turbine to obtain the highest efficiency, in addition, pressure fields acquired from unsteady simulations indicate that rotor-stator interaction cause high frequency pressure pulsation in the volute and low frequency pressure pulsation in the runner [10]. Liu et.al analyzed pressure fluctuations in the vaneless space between runner and guide vane for different openings of pre-opened guide vanes, the results indicate that in vaneless space: the amplitude of pressure fluctuations increased with the pre-opening increase and the dominant frequency equals to the blade passing frequency, however, the second dominant frequency decreased with the pre-opening increase [11]. Rodriguez [12] proposed a way to interpret the vibration induced by interaction of rotor-stator as a consequence of a modulation in the amplitudes of the interactions, instead of the result of an excited diameter mode that has been usually considered. Specifically, Zuo et.al give a review of pressure fluctuations in the vaneless space of high-head pump turbine [13], however, little research was carried out on the interaction of rotor-stator during the transient period.

Unsteady numerical study of dynamic process for complex hydraulic machinery is still in its infancy [14], the reason for the units produce abnormal vibration and noise in transient operating process is not very clear. As a consequence, in order to ensure units’ safety and stability and make units own a long lifespan, it is necessary to study the units’ stability and reveal the cause of unit vibration, further work should be proposed to reduce or avoid these unstable phenomena.

With the development of computer technology in recent years, CFD (Computational fluid dynamics) has become the primary way to investigate the characteristics of hydraulic machinery, CFD methods have been used to acquire the change of pressure and other parameters in pump turbines due to its feasibility in engineering practice [13]. In this context, based on CFD techniques, three dimensional unsteady simulations are performed on the load rejection period of one pump turbine which belongs to one Chinese pumped storage plant. In order to realize the continuous guide vane closure, optimized methods of dynamic sliding mesh is adopted in this process which will be presented in another work; to ensure authenticity, weakly compressible fluid is considered due to significant change of pressure; turbulent model used DES (Detached eddy simulation) for capture fluid details well with less computing cost than LES (Large eddy simulation). The main results of parameters including torque, pressure and flow rate are analyzed in both time and frequency domains for studying the interaction of rotor-stator, also 3-D flow streamlines is used to analyze vortex rope evolution of internal flow field.

2. The numerical model

The geometric pump turbine model is shown in figure 1, which computational domain consists of different zones: one volute, 20 stay vanes, 20 adjustable guide vanes, one runner with 9 backward blades and one draft tube. The characteristic parameters at best efficient point of this pump turbine model include Q=297 kg/s, H=30 m, n=670 rpm.

The calculating domains are built in Siemens Solidedge [15] whereas their mesh are generated in ANSYS ICEM [16], ANSYS CFX 16 is adopted to achieve numerical calculations while the post processing analyses are carried out in Matlab and CFX-Post. All simulations run in a workstation in parallel mode using 32 cores (Intel(R) Xeon(R) CPU E5-2650 188 @ 2.00GHz) and 128 Gb of RAM, the computational time for the unsteady simulation of continuous load rejection process was about twenty days.
This work aims at investigating the unsteady flow field in pump turbine during the process of load rejection by closing guide vanes, thus the mesh generation requires a high quality definition due to guide vanes movement. In order to increase the accuracy of the numerical solution, hybrid grid has been generated in the whole computational domain: structured blocks are used to split guide vane, runner and draft tube domains whereas volute utilizes unstructured mesh. The runner domain consists of a hub, a shroud and seven backward blades, is defined by about 1.93 M cells; the guide vane zone with 22 vanes, contains 1.52 M grid cells; in addition, O-type grids have been applied for all blades in both runner and guide vane domains. In order to capture characteristics of flow in draft tube, structured mesh is also generated in this part with about 1.42M cells, while tetrahedral meshes are used for the volute and stay vane. Figure 2 shows some mesh details in different regions of this model.

![Figure 1](image1.png)

**Figure 1.** Scheme of the main calculating zones of the pump turbine model.

A mesh sensitivity analysis has been carried out by simulating five different grid sizes in CFX, figure 3 shows a mesh independent solution at point A (about 2.36 M elements), however, the local pressure pulsations appear to be correctly captured only with total mesh number greater than 3.63 M (Point B). For the purpose of ensuring the ability to capture local pressure pulsations well, the mesh size was further subdivided at Point C (6.66 M elements). As for further increase of mesh size (Point D), the results indicate that about only 0.25% divergence in the average value of pressure fluctuations. As a consequence, total mesh size of this model adopted 6.6 M elements was applied in all simulations.

![Figure 2](image2.png)

**Figure 2.** Details of the mesh in some regions: (a) In the guide vane and runner, (b) In the draft tube.
3. Results and Discussion

This model was validated as shown in [3] based on incompressible fluid, figure 4 illustrates the validating data based on weakly compressible fluid. The uncertainty of the model must be taken into account to correctly evaluate the results, the errors between simulations and experiments may be arising from the practical limitations in numerical model. However, the uncertainty of relevant indicators as M11 and Q11 in both pump and turbine modes were less than 2% for the conditions near the design point and up to 13% for extreme off-design conditions, near the guide vane final closure point. The results show that the model with compressible water obtained a little better agreement, but all the errors between the numerical and experimental data are in a reasonable range not only at BEP condition.

Based on validated model, 3-D simulations were performed by using DES turbulent model which has been applied in many industrial cases and provided with good results [17-19]. In addition, weakly compressible fluid was applied in all simulations by incorporating equation (1) of state for water [20].

\[ P - P_0 = a_0^2 (\rho - \rho_0) \]  

(1)

In this equation, \( P \) is the pressure, \( a \) is the sound speed which is assumed to be constant in this work, \( \rho \) is the density and the subscript 0 represents a reference quantity.

The main context of this work is under turbine condition, total pressure is set at the inlet which is combined with the opening condition at the outlet, due to the presence of the highly disturbed flow field in the draft tube. Scalable wall functions for solid walls are used in steady simulations whereas automatic near wall treatment is applied in unsteady simulations. Three couples of interfaces among different parts are connected with GGI (General grid interface), whilst standard transient sliding
interfaces are adopted between stator and rotor. Each time step equals to 1 degree of runner rotation, and the maximum number of iterations for each time step is defined as five for the second-order implicit time stepping. As for settings for convergence, the root mean square (RMS) values of the residuals including: u momentum 10-5, v momentum 10-5, w momentum 10-6, and turbulence kinetic energy 10-5.

Figure 5 provides locations of monitoring points which are important due to the interaction of rotor-stator vary significantly along passage, different monitoring points have been set to probe the variations of parameters (pressure, flow rate and torque) during the analysis.

![Monitoring Points](image)

**Figure 5.** Locations of some monitoring points, (a) Top view of the monitoring points’ distribution; (b) Locations of monitoring points on the runner blade; (c) locations of monitoring points on the guide vane; (d) Locations of monitoring points in the draft tube [3].

### 3.1. Water ring zones during load rejection

The coefficient of pressure fluctuation \( (C_p) \) is defined in equation (2) and the values of \( C_p \) obtained from some points in guide vane domain are shown in figure 6. \( P \) represents the pressure whereas \( \bar{P} \) represents the average pressure. The pressure fluctuation gradually increased along the passage, this phenomenon is more obviously after \( T_r=0.82 \) when the relative opening of guide vanes decreased to less than 16%. The developing trend of pressure acquired from monitoring points in guide vane domain can be divided into two categories: the points near to stay vane have a increased tendency (GR, V1) while pressure at other points in this domain have a downward trend (V2-V5, GR). As closer to runner inlet, the pressure has greater reduction (V5, GR).

\[
C_p = (P - \bar{P}) / \bar{P}
\]  
(2)
**Figure 6.** (a) Pressure coefficients of monitoring points in guide vane domain; (b) Flow field in water rings.

It should be noticed that a counter-trend of the pressure development of SG and GR which locate in the vaneless spaces, pressure at SG kept a stable increase while pressure at GR reduced in this load rejection process, furthermore, pressure at GR has larger fluctuations after $T_f=0.82$ due to it was also significantly affected by fluid in runner.

The vaneless spaces between rotor and stator increased during the shutdown phase, which induced the flow water was stopped before guide vanes and formed a water ring between stay vane and guide vane, where is the region with higher pressure. Meanwhile, most of the high pressure flow energy in this region was dissipated rather than converted into mechanical energy which means hydrodynamic moment decreased in runner (figure 7 (a)). Figure 7 (a) illustrates torque evolution of runner blades in the load rejection period, the values of torque have a downward tendency; it worth noticing that two red circles in this figure which locate at the beginning and the end, torque fluctuations are higher than the other region, the reason for the left circle is the sudden closure of guide vanes induced internal flow field change from a stable BEP condition, whereas the right circle cause is unstable flow field in runner passages which have been almost blocked (figure 8), however, all torque fluctuations are in a small range which is beneficial to contrast the occurrence fatigue phenomena.

Furthermore, the water ring between the guide vane and runner was thicker due to the high values of the pressure gradient in these regions, centrifugal forces appeared in the guide vane domain were generated also with the contribution from runner rotation. It should be noticed that flow in water ring has a similar rotating direction of runner, the maximum amplitudes of blade passing frequency can also be observed at the guide vane trailing edge whereas points located on the other side of guide vane showed lower amplitude (figure 6(a)). Distinctive frequencies are shown in figure 7 (b) at different monitoring points, the spectral pressure frequency at the head edge of vane (V1) is almost 15 times of the value at the trailing edge of vane (V5), which can be observed with the highest amplitudes by black circles in figure 7, while other points (V2, V3, V4) locate on the middle passage of vane are relatively stable compared with values of these two points: V1 and V5.
Figure 7. (a) Trend of torque on runner blades. (b) Normalized power-spectra of the pressure signals acquired from monitoring points locate on guide vane from St=0.02 to St=0.5.

Figure 8 reports development of pressure coefficients in the vaneless space between guide vane and runner, where V5 and R22 locate at the tips of vane and blade. It can be seen that the change of pressure increased as closer to runner, pressure at R22 has the largest reducing gradient, which explains rotor-stator interaction of pressure fluctuations in vaneless space enhanced as radius decrease. The major reason of this phenomenon is: wake of guide vane interfere with wake of runner, and flow in this region influenced by the fluid viscous and blade external flow which has larger turbulent pulsation.

Figure 8. Pressure coefficients acquired from monitoring points V5, GR and R22 (figure 5).

At the beginning of the process of load rejection, fluid in the whole passage is stable due to the pump turbine worked at the design point condition, the growth of the water ring thickness and the inflow angle at the runner blade leading edge changed far away from design point was associated with reduction of the space between guide vanes, this phenomena induced larger centrifugal force on runner. Furthermore, the formation of water ring is the major reason that prevented water flowing into the runner domain, and gradually less water flow in runner passage induced blocks which accompanied with lower pressure due to departure from the design point (figure 9). Figure 9 shows streamlines in the runner passage and pressure distribution on the runner blades at the end of the process of guide vanes closure, a small amount of flow near to runner inlet region formed reflux and flowed back. It can be seen that flow field near to runner outlet consisted with water from draft tube which has the opposite rotating direction compared to runner rotating direction. Moreover, basically pressure on two sides of runner blade at corresponding location was basically equal due to the runner domain was approximately empty.
3.2. Analysis of the flow field in the draft tube

On the other hand, the flow rate decrease at the runner outlet is the main reason that caused vortex rope in the draft tube (figure 9), figure 10 shows some details of flow at the inlet of draft tube under some moments, specifically, figure 10(a) presents 3D streamlines while figure 10(b), (c) and (d) illustrates 2D fluid in this region, vortex rope in the draft tube appeared at $T_f=0.076$ when the closing process has been finished about 5%, and the vortex rope became stronger as the flow rate kept decreasing. Gradually two opposite flow appeared in the draft tube, one was similar to the previous stage through this domain with a larger rotating velocity $V_D$ ($T_f=0.506$), the other one was formed by reflux and rotated in the opposite direction which flowed back, and the yellow circles illustrate this phenomenon has a regional expansion.

When the vanes reached near to final positions, vortex rope formed by reflux can be seen in the draft tube clearly. The vortex rope rotating direction was consistent with runner rotating direction ($T_f=0.855$), this phenomenon was the main reason that led to severe blockage to the channels and it was the primary cause that huge pressure fluctuations appeared during this operating period (figure 11).
Figure 10. Details of flow at the inlet of draft tube.

Figure 11. Pressure coefficients of monitoring point RP2 from Tf=0.8 to Tf=1.

The flow axial velocity played a key role in the instability of unit, which can be identified in conjunction with vortex intensity that caused changes of vortex patterns in the draft tube cone, starting from vortex rope appeared at Tf=0.076, the averaged axial symmetric swirling flow was unstable from the inlet section of the cone and this phenomenon was getting obvious.

Figure 12(a) reports pressure distribution at the inlet of the draft tube, the nine different pressure regions at the wall of the draft tube cone were caused by the rotation of runner, this phenomenon related to nine passages between each two runner blades and the evolution of their wake profile. Figure 12(b) shows normalized power-spectral of pressure at monitoring points locate at draft tube inlet (figure 5(d)), the analysis of the pressure fluctuations at point RP2 detected the runner rotating frequency (St = 0.6763) which corresponds to the formation of vortices in the draft tube domain.

Figure 12. (a) Pressure distributions at the inlet of draft tube; (b) Normalized power-spectra of the pressure signals acquired from monitoring points RP1, RP2 and RP3.
4. Conclusions

3-D numerical simulations performed on a validated model based on DES turbulent model with weakly compressible fluid, according to the above analyses, it can be summarized that:

Rotor-stator interaction of pressure fluctuations in vaneless space (between guide vane and runner) enhanced as radius decrease;

The increased inflow angle is the main reason that caused unstable flow field in runner passage while rotating flow in water ring region is another major cause that prevented water flowing into runner domain;

The flow field in draft tube was affected significantly since guide vanes had been closed approximately 3.5%, where the vortex rope appeared and the initial vortex rotating direction was consistent with the runner rotating direction; then reflux appeared and rotated around draft tube hub in opposite direction compared with flow on the draft tube wall (vanes have been closed about 42.8%), and these two flow parts finally formed two separate rotating water paths, the two separate water paths is one significantly important reason causing unit vibration; after the guide vanes have been closed about 85.6%, only the second vortex pattern existed with a small amount of flow.

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NOMENCLATURE

| Symbol | Definition |
|--------|------------|
| BEP    | Best efficiency point |
| DES    | Detached eddy simulation |
| H(m)   | Head |
| L(m)   | Hydraulic diameter |
| n(r/min) | Runner rotating speed |
| PSD    | Power spectral density |
| P(Pa)  | Pressure |
| Q(kg/s) | Mass flow |
| St=fL/V | Strouhal number |
| t(s)   | Time |
| T=t/tMax | Time factor |
| V(m/s) | Velocity |

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