CFD Analysis of the Runaway Stability of a Model Pump-Turbine

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Abstract: The relations between the runaway stability characteristics and the flow patterns inside the runner of pump-turbine are supposed to be close and should be studied. The runaway processes of a model pump-turbine at four guide-vane openings (GVOs) were simulated by the three-dimensional computational fluid dynamics. The results show that the runaway stability characteristics for the pump-turbine are different at different GVOs. For the small GVOs, the turbine characteristic trajectory undergoes damped oscillations; however, for large GVOs, the turbine characteristic trajectory settles into an un-damping oscillation. The evolution features of the reverse flow vortex structures (RFVS) at the runner inlet during the runaway oscillations have distinct patterns between the small and large GVOs. For small GVOs, the RFVSs only locate at the mid-span; however, for the large GVOs, the location of the RFVSs switches back and forth between the mid-span section and the hub side when the turbine passes in and out the turbine braking mode. The changes of RFVS at the runner inlet dominate the energy transfer among the hydraulic, mechanical and dissipation energies during the transient processes, and therefore affect the stability of hydraulic system.

1. Introduction

Pumped-storage plants are often acted as a means of fast accommodating the power demands and regulating the frequency of the power grid. To fulfill these requirements, the pump-turbines constantly undergo transitional processes between start, stop and other conditions. Under these circumstances, the S-shaped characteristics of pump-turbine may make the start-up procedure or runaway process after load rejection unstable [1-3]. Such instability may be caused by the excited oscillation in the hydraulic system [4, 5].

The criteria for predicting and assessing the runaway stability of pump-turbines based on the characteristic curves are indispensable for hydropower station operation. Martin [6, 7] has proven that the most important factors affecting the runaway stability are the slopes \( \left( \frac{dQ}{dn} \right)_{n=1} \) and \( \left( \frac{dT}{dn} \right)_{n=1} \) of the characteristic curves at the no-load point. However, the flow losses in the water conduits benefit the stability. Dörfler et al. [8] used artificial head loss (caused by partially closing the turbine inlet valve) to change the combined head versus flow characteristics of turbine and valve, and applied this method to a real hydropower plant for keeping stable operations despite unstable at the no-load point. This approach was widely
used in model tests for measuring the S-characteristics of pump-turbines\cite{9,10}. However, this method cannot be used for load rejection and does not eliminate the instability origin in the pump-turbine. Understanding the flow features that lead to the instable behavior is a prerequisite for the design of machines that can fulfill the growing requirements of operation stability and flexibility. Klemm\cite{11} introduced the misaligned guide-vane method to overcome the system instability. Essentially, the flow patterns within the flow passages were changed, and therefore the characteristic curves were also changed. Moreover, the misaligned guide-vanes may cause even higher pulsating pressure and shaft vibrations\cite{12}. Many publications focused on the flow patterns around no-load conditions. The reverse flow vortices at the runner inlet were thought to be the cause for the formation of S-shaped $n_{11}$-Q$_{11}$ characteristic curves\cite{13-18}. Xia\cite{19} proved that the flow losses caused by small setting angle of the blade inlet are the root cause of S-shaped formation of $n_{11}$-Q$_{11}$ characteristic curves by theory equation, and found the location of reverse vortices at the runner inlet impacts the slope of characteristic curves at no-load points at static condition. However, for the dynamic process, the impacts of flow patterns on the system stability are still not clear.

In this study, 3D-CFD simulations were conducted to investigate the runaway processes of a model pump-turbine at different guide-vane openings (GVOs), for revealing the causes of different dynamic stabilities at small and large GVOs.

2 Numerical simulation conditions
2.1 Computational domain and grid

The runaway stability of a pump-turbine is mainly affected by the slopes of the characteristic curves. In order to reduce the effect of the water conveyance system and enlarge the effects of runner and guide-vane on the dynamic characteristics, the computational domains only include flow passages from the spiral-case inlet to the draft-tube outlet (Fig.1 (a)). The draft-tube outlet is extended to a certain length for smoothing the swirl flow. The detailed specifications of the model pump-turbine are listed in Table 1, where $D_1$ and $D_2$ stand for the inlet and outlet diameters of the runner, respectively; $z_b$, $n_{sv}$ and $n_{gv}$ are the numbers of runner blades, stay vanes and guide-vanes, respectively; $n$ is the rotation speed of the runner; $\alpha$ is the degree of GVO. The flow in the model pump-turbine was simulated at 6°, 9°, 15°, and 24° GVOs.

| $n_s$ (m·Kw) | $D_1$ (m) | $D_2$ (m) | $z_b$ | $n_{sv}$ | $n_{gv}$ | $n$ (rpm) | $\alpha$ (°) |
|-------------|----------|----------|-------|----------|----------|----------|-----------|
| 29.17       | 0.280    | 0.1409   | 9     | 20       | 20       | 1000     | 6, 9, 15, 24 |

Fig.1 Computational domain and grid

Hybrid grids were generated for different domains by software ANSYS ICEM 14.0. The tetrahedral grids were used in the spiral-case; the wedge grids were used in the vane diffuser; the structured hexahedral grids were applied in the runner and draft-tube (Fig.1 (b)). Special refinements were applied in the runner and guide-vanes domains to match the requirement of
y+<10 at the best efficiency points. The grids for different GVOs only differ in the guide-vanes domain. Grid independence checks were conducted with several different mesh refinement levels for the four GVOs, respectively. When the number of grid elements is more than 4.0 million, the differences of macro-parameters, such as water head and hydraulic efficiency, are less than 0.3% between different levels, indicating the numerical simulation results are reasonable. Considering the numerical accuracy and time cost, the total element numbers were limited to no more than 9.0 million. The grid parameters used for the simulations are shown in Table 2. The total numbers of grids at the four GVOs are 7.86 million, 7.78 million, 8.22 million, and 7.75 million, respectively.

| Table 2 Number of grid elements (million) |
|------------------------------------------|
| Spiral-case Guide-vane (6°) | Guide-vane (9°) | Guide-vane (15°) | Guide-vane (24°) | Runner | Draft-tube with Extension |
| 1.01 | 2.06 | 1.98 | 2.42 | 1.96 | 2.45 | 2.34 |

2.2 Turbulence model and boundary conditions

The unsteady simulations were performed using ANSYS FLUENT 14.0. Based on a comparative analysis, the SAS-SST model was applied, which introduces the von Karman length-scale into the turbulence scale equation. The information provided by the von Karman length-scale allows the SAS-SST model to dynamically adjust to the resolved structures in the unsteady Reynolds Averaged Navier-Stokes (URANS) simulation, which results in a LES-like behavior in the unsteady flow regions. At the same time, the model provides standard RANS capabilities in stable flow regions [20, 21].

The boundary conditions were defined as follows: total pressure was defined at the spiral-case inlet, and static pressure was used at the outlet of the draft-tube. The remaining boundary conditions are imposed by the no-slip walls.

2.3 Time step and numerical scheme

During the numerical simulations, the results of steady RANS simulations were used as the initial flow field for the transient simulations. In the steady state simulations, the multiple reference frame approach was used for the runner zone; in the transient simulations, sliding mesh approach was used. The time-step was set to 0.0002 s, and the maximum number of iterations per time-step was set to 40 with the convergence criteria of residuals at each time-step were 1.0E-5. SIMPLEC algorithm was chosen to achieve the coupling solution for the velocity and pressure equations. Second order discretization schemes in time and in space were used.

3. Results and Analysis

3.1 Differences between dynamic trajectories and static characteristic curves

The runaway processes after load rejections at four different GVOs of a pump-turbine were simulated. Fig. 2 shows the dynamic trajectories and the static characteristic curves at the corresponding GVOs. With GVOs at 9°, 15° and 24°, the unit settles down into self-excited and cycle-limited oscillations. Whereas, the oscillation is damped for the GVO at 6°. The larger the GVO, the bigger range the cycle covers and the deeper the pump-turbine gets into the reverse pump mode. This change law means that the runner may become more unstable at the no-load condition when GVO increases.

The unit drops into the oscillations at the turning points (P1 to P4 in Fig. 2) where the slope of dynamic trajectories first begins to have a sharp change. At the primary stage of the runaway process, the dynamic trajectories of operating parameters are nearly consistent with the static characteristic curves until to the stagnation points (P’1 to P’4 in Fig. 2) where the dynamic trajectories and characteristic curves change their slopes. However, the dynamic trajectories will not follow the static characteristic curves below the stagnation points.

For the dynamic trajectories, the turbine water head $H$, defined for the operating parameters of $n_i = n D_i / \sqrt{H}$, $Q_i = Q / D_i^2 \sqrt{H}$ and $T_i = T / D_i^3 H$ in Fig. 2, is a given head between the inlet of the spiral-case and the outlet of the draft-tube. Therefore, $dT_{1i}/dn_{1i} = dH/dn = \ldots$
The aforementioned water head \( H \) during runaway process includes some dissipation energy and inertia energy in the draft-tube that will not be used by the runner. A modified dynamic water head \( H_d \) is defined between the spiral-case inlet and the runner outlet, as shown in Fig.3. The modified dynamic trajectories (Fig.4) become thinner for larger GVOs (9°, 15° and 24°) along the direction of \( n_{11} \) axis, while the range between the maximum and minimum values of \( Q_{11} \) has little change. At the extreme points of \( Q_{11} \), the change rate of \( Q \) over time is zero. Therefore, the effect of hydraulic inertia energy disappears. However, when the unit operates at the range with \( Q_{11} \) rapidly changing, the hydraulic inertia and flow losses in draft-tube increase significantly, which have obvious influence on the dynamics trajectories. The slopes at the no-load points turn to be positive, which meets the instability criterion. On the contrary, the dynamic trajectories for 6° GVO do not become thinner obviously, and the slopes at the no-load points turn to be negative, which means the dynamic process will become stable.

### 3.2 Changes of modified dynamic trajectories

The change law of dynamic trajectories using the modified dynamic water head is different between 6° GVO and the larger GVOs. Figs.5-7 show the change trends of dynamic trajectories and time histories of pressure and radial velocity fluctuations of monitoring probes (seeing Fig.3) for different GVOs. As it can be seen, the negative radial velocities occur during the runaway process for all GVOs. However, the change of velocities at runner inlet for 6°GVO is different from those at the larger GVOs. At 6° GVO, the reverse flow vortices only occur at the mid-span section, while at the large GVOs, the locations of reverse flow vortices have regular transition between the hub side and mid-span sections. The reverse flow vortices will block the flow passages and increase the total pressure at the guide-vane domain (e.g. \( P_{ro} \), in Fig.5 (b)-Fig.7 (b)), before the unit entering into the turbine brake mode in the forward direction, or after the unit coming out from the turbine brake mode in the backward direction. At the large GVOs, both the low frequency components and high frequency components of total pressure fluctuations \( P_{ro} \), at the guide-vane inlet are relevant to the transitions of reverse flow at the runner inlet (Fig.6 (b) and Fig.7 (b)). However, at 6° GVO, the low frequency component of total pressure fluctuations have apparent relations with the pressure oscillations at the runner outlet.

When the operating points move forward from the turbine mode to the reverse pump mode or move backward from the reverse pump mode to the turbine mode, the variations of total pressure \( P_{ro} \) at the runner outlet are different between the two directions. Moreover, the change law for 6° GVO is different with those for other larger GVOs. At 6° GVO, \( P_{ro} \) first increases in the forward direction, and then decreases in the backward direction (Fig.5 (a)). And also, both the total pressure values \( P_{ro} \) at the both no-load points (OP1 and OP2) are larger than the total pressure of the starting operating point (Fig.5 (b)), which makes the modified water head \( H_d \) smaller than the starting water head. Therefore, the two slopes at the no-load points (OP1 and OP2) turn to be negative and the operating parameters \( n_{11} \) at these points become larger than the original values (Fig.5 (a)).

On the contrary, at the larger GVOs, the total pressure \( P_{ro} \) first decreases in the forward direction, and then increases in the backward direction. Meanwhile, \( P_{ro} \) at the no-load point OP1 is smaller than the total pressure at the starting operating point, but \( P_{ro} \) at no-load point OP2 is larger than the initial total pressure (Fig.6 (b) and Fig.7 (b)). As a consequence, the slopes at the no-load points (OP1 and OP2) turn to be positive (Fig.6 (a) and Fig.7 (a)). Moreover, the operating parameters \( n_{11} \) at OP1 become smaller and at OP2 become larger than the values at original condition (Fig.6 (a) and Fig.7 (a)).
Fig. 2 Comparison between the dynamic trajectories and static characteristic curves

Fig. 3 Monitoring sections and probes

Fig. 4 Modified dynamic trajectories

Fig. 5 Dynamic trajectories and time histories of operating parameters at GVO 6°
3.4 Effects of hydraulic inertia and dynamic pressure on the dynamic trajectories

From above analysis, the stability at small GVOs is affected by the total pressure at the runner outlet. In order to identify the cause of the differences of total pressure fluctuations at different GVOs. The history curves of discharge, time rate of discharge, total pressure and static pressure at the runner outlet for 6° and 15° GVOs are presented in Fig.8. For the two GVOs, the change laws of the static pressure fluctuations are consistent with that of the time rate of discharge $dQ/dt$, which means the static pressure in conduits is dominated by hydraulic inertia. Generally, the total pressure at the runner outlet during oscillation process is also mainly controlled by the hydraulic inertia in the draft-tube and extension-tube because the proportion of dynamic pressure is small. However, the change law of the total pressure fluctuations at 6° GVO are significantly different from that of $dQ/dt$ (Fig.8(a)), while the one at 15° GVO are similar to that of $dQ/dt$ (Fig.8(b)). At the large GVO, the amplitude of time rate of discharge is much larger than that at the small GVO. Therefore, the hydraulic inertia at the large GVO has more significant impact on the total pressure at the runner outlet. Conversely, at 6° GVO, the hydraulic inertia has minor effect on the change of total pressure at the runner outlet.

Figure 9 shows the newly revised dynamic trajectories by excluding the hydraulic inertia head in the draft-tube at 6° GVO. The change law of the inertia-excluded dynamic trajectory is different from that of the modified trajectory at 6° GVO, while its change is similar to those of the modified dynamic trajectories at large GVOs (Fig.5 (a)-Fig.7 (b)). As a consequence, the hydraulic inertia has negative impact on the stability because it leads to positive slope at
the no-load point; meanwhile, the dynamic pressure has positive impact on the stability because it offsets the adverse effect of inertia and turns the sign of slopes to negative at the no-load points. The predominance of dynamic pressure at the runner outlet demonstrates that a large amount of dynamic energy is dissipated within the draft-tube at 6° GVO. Fig.12 shows the distributions of turbulence eddy frequency at different time for 6° GVO. During the process of the unit entering into the turbine bake mode, the turbulence eddy frequency increases around the runner inlet and outlet and near the side wall of draft-tube, in which larger amount energy was dissipated by the reverse flow vortices around the runner outlet leading edge. Therefore, the dynamic pressure can dominate variations of local water head.

Fig.8 Comparison of operating parameter histories between the small and large GVOs

Fig.9 Comparison of the original, modified and inertia-excluded dynamic trajectories

Fig.10 Distribution law of velocity profiles at the runner inlet at static conditions

Fig.11 Reverse flow vortices at different locations

(a) shroud side  (b) Hub side  (c) Mid span
3.5 Reverse vortices structures and energy transfer

Xia [19, 22] proved that the flow losses at the runner are the root cause of the S-shaped $n_{11}-Q_{11}$ characteristic curves by theory equation and the evolution laws of vortices at the runner inlet influence the slope of characteristic curves at the no-load points at static conditions. Fig.10 shows the distribution laws of velocity profiles at the runner inlet in the first quadrant of characteristic curves at static conditions and the corresponding typical flow patterns are shown in Fig.11. When the operation points get away from the best efficiency points to the zero discharge points, the distributions of the reverse flow vortex structures (RFVS) at the runner inlet have regular transitions. At large GVOs, the RFVS first incept at the shroud side in a narrow operating range, then turn to the hub side with smaller discharge, and finally translate from the hub to the mid-span near the runaway points. With the decrease of the GVO, the range of RFVS appearing at the shroud and hub sides decrease gradually. When the GVO is less than 9°, the RFVS only exist at the mid-span. The dynamic evolution laws of RFVS during the runaway process at different GVOs (Figs.5 (b)-7(b)) are similar to that at the static condition. For large GVOs, the no-load points are around the turning points where the locations of RFVS turn from the hub side to the mid-span section in the forward direction or from the mid-span section to the hub side in the backward direction during the oscillations (Fig.6 (b) and Fig.7 (b)). While for 6° GVO, the RFVS only occur at the mid-span section at the runner inlet (Fig.5 (b)), the no-load points have no relation with the location of the RFVS. Moreover, the evolutions of reverse flow at large GVOs (Fig.7 (b) and Fig. 8(b)) have much more effect on the variation of time rate of discharge $dQ/dt$, while the one at 6° GVO mainly affects the variation of discharge (Fig.5(b) and Fig. 8(a)).

![Fig.12 Distributions of turbulence eddy frequency at different times for 6° GVO](image)

Figure 13 show the histories of energy transfer during the runaway process for different

![Fig. 13 Histories of energy transfer during the runaway process](image)
GVOs. Apparently, for 6° GVO, the dissipation energy increase with the RFVS incepting and developing at the mid-span. When the unit undergoes oscillation, the total dissipation energy is larger than the total hydraulic energy over one oscillation cycle. Therefore, the runner obtains less mechanical energy over one cycle, which leads to a damped oscillation of mechanical energy. For larger GVOs, the RFVS occurs at the mid-span in the turbine bake and reverse pump modes, and more energy can be dissipated than when the RFVS is at the hub side in the turbine mode (Fig.13(b)-(d)). However, the runner can obtain enough mechanical energy at the turbine mode from hydraulic system because of lacking enough dissipation energy. As a consequence, the total hydraulic input energy is equal to the dissipation energy over one oscillation cycle. Hence, the total mechanical energy does not decrease over a cycle and the oscillation is un-damped.

4. Conclusions

To investigate the runaway instability and reveal the laws, 3D-CFD simulations of flow characteristics in a model pump-turbine at different GVOs were carried out. The conclusions are as follows:

1. The hydraulic inertia in the draft-tube has a negative impact on the stability because it leads to positive slope of the dynamic trajectories at the no-load points. The larger the GVO, the apparent the effect of the hydraulic inertia exerts on the instability.

2. The dynamic pressure, caused by a large amount of energy dissipation around the runner outlet to the cone section of the draft-tube, has positive impact on the stability. This effect offsets the adverse effect of inertia and turns the sign of the slopes of the dynamic trajectories to negative at the no-load points.

3. The evolution of RFVS at the runner inlet dominates the energy transfer among the hydraulic energy, mechanical energy and dissipation energy during the runaway transients, and then affects the stability of hydraulic system.

To sum up, the energy transfer in microcosmic scale caused by the evolutions of RFVSs dominates the change trend of characteristic curves at the no-load points. The superiority between inertia and dissipation energy will determine runaway stability.

Acknowledgments

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