Prediction of pressure fluctuation of a hydraulic turbine at no-load condition

T J Chen¹, X J Wu², J T Liu³ and Y L Wu¹
¹ State Key Laboratory of Hydroscience and Engineering, Tsinghua University, Beijing 100084
² Hunan M&W Energy Saving Technology LTD, Hunan, China, 410000
³ Beijing Institute of Control Engineering, Beijing 100190

Email: ctj@mail.tsinghua.edu.cn

Abstract. In order to study characteristics of pressure fluctuation of a turbine during the starting period, a turbine with guide vanes device at no-load condition was investigated using RNG k-ε turbulence model. The inner flow distribution and pressure fluctuation characteristics were analyzed. Results show that the pressure fluctuations in the region between the runner and guide vanes are different around the runner inlet. The dominant frequency of pressure fluctuation in the vaneless space close to the casing outlet is the blade passing frequency, while the dominant frequency at the rest region is the twice of the blade passing frequency. The increase of amplitude of pressure fluctuation close to the casing outlet can be attribute to the large scale stall at suction side of the runner inlet.

1. Introduction
Hydraulic turbine is commonly used as a residual pressure energy conversion device in petrochemical industry. It plays an important role in decreasing costs and improving economic efficiency. Hydraulic turbine will run through runaway operating point during start-up and shutdown process. When a hydraulic turbine runs at partial conditions, there will be a complex internal separation vortex and reverse flow, which may cause hydraulic turbine unstable and even cause the entire system to a larger pressure pulsation. The research on pressure fluctuation characteristics at runaway point of a hydraulic turbine has an important significance for understanding the stability of a hydraulic turbine.

At present, the selection and design of hydraulic turbine was done empirically by analyzing the operating characteristics of reverse pump. Most of hydraulic turbines in engineering application were designed by reverse pump [1-3] and optimizing the pump impeller to change the performance of a hydraulic turbine [4-6]. Studies on the stability of the hydraulic turbine are still needed. Yang Junhu et al [9] analyze effects on the external characteristics of the hydraulic turbine based on numerical simulation by changing the blade shape. Yang Sun Saint et al [10] investigated external characteristics of a hydraulic turbine by changing the length of the blade using experiment and numerical simulation method. Fernandez [11] and Yang Junhu [12] studied the influence of vanes on the performance of a hydraulic turbine, and they found that there is a certain intrinsic link between efficiency and flow rate coefficient of hydraulic turbine. The constant hydraulic turbine characteristic is more obvious and the pressure will not change with the changes in flow rate by the use of guide vanes. Inhama [7] analyzed the force on runner and the runaway speed of a hydraulic turbine which design from a reverse pump.
Shahram [8-9] analyzed detailed external characteristics of hydraulic turbine under different speed by experiment and numerical simulations, and it proposed a reasonable range of rotating speed of a reverse pump when it was used as a hydraulic turbine.

In this paper, the external characteristics and internal flow field of a hydraulic turbine at runaway operating point are analyzed by numerical simulation. The internal flow characteristics as well as rotor-stator interaction of the hydraulic turbine are investigated. The stability of the hydraulic turbine at runaway operating point is also studied.

2. Model of hydraulic turbine
A hydraulic turbine with guide vanes is chosen and modeled and the parameters of the hydraulic turbine are shown in table 1. Three-dimensional hydraulic model of the hydraulic turbine is shown in figure 1. $D$ is the diameter of the runner. $B$ is the number of the blades. $V$ is the number of the guide vanes. $n$ is the rotating speed. The hydraulic turbine is mainly composed of guide vanes, runner and double suction outlet.

ICEM is used to mesh the four components. The mesh on blades and vanes is used 12 layers for more accurate description near the wall region. Each hydraulic turbine components are divided by hexahedral mesh. The mesh of the runner is shown in figure 2. The total number of mesh of the entire hydraulic turbine is 6.5 million.

### Table 1. Parameters of pump-turbine.

|   |   |   |   |
|---|---|---|---|
| $D$ (m) | $B$ | $V$ | $n$ (r/min) |
| 0.45 | 9  | 20 | 3000 |

![Figure 1. Profile of the turbine.](image1)

![Figure 2. Mesh of the runner.](image2)

In order to analyze the stability of the hydraulic turbine during start-up and shutdown process at the run runaway point, unsteady calculations are performed.

3. Turbulence model and boundary conditions
When a hydraulic turbine runs at partial conditions, the internal flow is complex and flow separation will appear. In order to capture the complex flow in hydraulic turbine and obtain more accurate results of pressure fluctuation, RNG $k-$s model is used to calculate the flow in the hydraulic turbine. Equations of RNG $k-$s turbulence model are shown as follows:

$$
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho u_i k) = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + P_k - \rho e
$$

(1)
\[
\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_i}(\rho \epsilon u_i) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma} \right) \frac{\partial \epsilon}{\partial x_j} \right] + \frac{C_{1e}}{k} \epsilon P_k - C_{2e} \rho \frac{\epsilon^2}{k}
\]  

(2)

where \( P_k \) is the production of turbulence kinetic energy; \( C_{1e} \) is a constant; \( \sigma_k \) and \( \sigma_\epsilon \) is the Prandtl number of \( k \) equation and \( \omega \) equations, respectively.

Total pressure inlet at the casing inlet and pressure outlet at double suction outlet of the hydraulic turbine is set. Interface is used for the data switching between turbine runner inlet and guide vanes as well as runner outlet and double suction. The runner rotates for 1° is used as the time step. SIMPLEC algorithm is used to couple pressure and velocity. The second-order upwind scheme is used to discrete equations. No-slip boundary conditions are used on the walls. Computing time sustains the runner rotating five cycles and the residual of 0.0001 is used as the convergence conditions within each time step.

4. Results and conclusions

3.1 Time domains of pressure fluctuations

For the hydraulic turbine, the rated head is 230 m and the rated flow is 0.8 m$^3$/h. Pressure fluctuation measuring points are shown in figure 1. Measuring points between the vanes and the runner which are A, B, C and D has an equally distribution along the circumferential direction. Time domain of pressure fluctuations on measuring points A, B, C and D are shown in figure 3. The relative amplitude of the pressure fluctuation is shown in table 2.

Pressure fluctuation at point A has maximum relative amplitude. Point C has minimum amplitude of pressure fluctuation. The amplitude of pressure fluctuation circumferentially increases after a small decrease at clockwise direction from point A. Point A which is near the volute outlet. There have different sections on the left and right sides of the volute, causing different flow rate, resulting in the increase of amplitude of pressure fluctuation. Point C is far away from the dramatic changes in the volute section, so the amplitude of pressure fluctuation is lower than the other points.

![Figure 3. Locations of monitor points.](image)

3.2 Frequency domains of pressure fluctuations

The Fourier transform results of pressure fluctuation at different points are shown in figure 5. Analysis of the amplitude of pressure fluctuation corresponding to the characteristic frequency is shown in table 2.

| Table 2. Relative amplitude of pressure fluctuation (\( \Delta H/H \)) |
|-----------------|-----------------|-----------------|-----------------|
| A               | B               | C               | D               |

3
| $\Delta H/H\%$ | 8.2% | 7.2% | 5.6% | 6.8% |

**Figure 4.** Pressure of each point at time domain.
Dominant frequencies of the pressure fluctuations between the vanes and the runner inlet are different along the circumferential position. The dominant frequency is the blade passing frequency at point A corresponding to the position of volute outlet. The pressure fluctuation is mainly affected by the number of runner blades at point A, and there is a low frequency which is 0.7 times of the rotating frequency. The lower frequency component may be related to the stall vortex near the vaneless space. The second frequency at point A is twice of blade passing frequency, and the amplitude of the second frequency component is 90% of the dominant frequency component. The dominant frequency is twice of the blade passing frequency at point B, and the second frequency is the blade passing frequency. The amplitude of the rotating frequency component has a higher proportion in the spectrum of pressure fluctuation. The dominant frequency and second frequency at point C are the same as point B. The amplitude corresponding to the second frequency is approximately 70% of the dominant frequency component. Point C also has a low-frequency which is 0.7 times of rotating frequency. The dominant frequency is twice of the blade passing frequency at point D, and the second frequency is the blade passing frequency. The amplitude of the second frequency component at point D is 95% of the dominant frequency component. The amplitude of blade passing frequency decreases at clockwise beginning from point A. The amplitude of twice of the blade pass frequency begins to decrease after the first increase in the clockwise direction from the point A, and it has a maximum value at the point B.

3.3 Flow analysis
Streamlines in the runner at runaway condition of the hydraulic turbine are shown in Figure 6. Large scale vortex appears at the suction side of the blade in the one of the passage of the runner. The large scale vortex locates just adjacent to the volute outlet section, which may be the reason for causing the increase of the amplitude of pressure fluctuation.

4. Conclusions
When a hydraulic turbine operates at runaway point, the position of the maximum amplitude of the pressure pulsation locates near the tongue of the volute. The dominant frequency changes along the circumferential position at the runaway condition in the vaneless space. The dominant frequency of the pressure fluctuation in the vaneless space near the volute tongue is the blade passing frequency, and the dominant frequency at other point far away from the volute tongue is the twice of the blade passing frequency. The increase of the amplitude of pressure fluctuation near the volute tongue can be attributed to the large-scale vortex in the runner.
Acknowledgments
The authors would like to thank project 51406010 supported by National Natural Science Foundation of China and project supported by open Research Fund Program of State key Laboratory of Hydroscience and Engineering, NO. sklhse-2014-E-02.

References
[1] Derakhshan S and Nourbakhsh A 2008 Exp. Thermal and Fluid Science 32 (3) 800-7
[2] Williams A A 1994 J. Power and Energy 208 (1) 59-66
[3] Yang J H and Wang X H 2011 J. Drainage and Irrigation Machinery Eng 29 (4) 287-29 (in Chinese)
[4] Yang S S, Kong F Y, Xue L et al. 2012 Transactions of the Chinese Society for Agricultural Machinery 43 (7) 104-7 (in Chinese)
[5] Fernandez J, Blance E, Parrondo J, et al. 2004 J. Power and Energy 218 (4) 265-71
[6] Yang J H, Gong C H, Xia S Q, Luo K K and Li H L 2014 J. Drainage and Irrigation Machinery Eng 32 (2) 113-118 (in Chinese)
[7] Hitoshi I, Junichiro F and Yoshiyuki N 1999 Study of reverse running pump turbine Proc. 3rd ASME/JSME Joint Fluids Eng. Conference (San-Francisco, California, USA, 18-23 July 1999) pp1-6
[8] Shahram D and Ahmad N 2008 Exp Thermal and Fluid Science 32 800-7
[9] Shahram D, Bijal M and Ahmad N 2009 J. Fluid Eng. 131 (2) 8-34