Analysis of Dynamic Stresses of Pump-Turbine Runner during Load Rejection Process in Turbine Mode

Chen Funan\textsuperscript{1}, Yang Xiaolong\textsuperscript{3}, Bi Huili\textsuperscript{1}, Mao Zhongyu\textsuperscript{1}, Luo Yongyao\textsuperscript{1}, Liujingshi\textsuperscript{2}, Fan Hongang\textsuperscript{1}, Wang Zhengwei\textsuperscript{1}

\textsuperscript{1}Department of Energy and Power Engineering, Tsinghua University (Beijing, CN);
\textsuperscript{2}Harbin Electric Machinery Company Limited.
\textsuperscript{3}Construction and Management Branch of CSG Power Generation Co.Ltd.

wzw@mail.tsinghua.edu.cn

Abstract. During load rejection process, the rotation rate, head and discharge change a lot and the internal flow of the runner is complex, leading to fatigue cyclic loads on the runner. In some cases, fatigue cracks appear at the stress concentration location of the runner. This paper studied the dynamic stresses of a pump-turbine runner during load rejection process in turbine mode using finite element method. The hydraulic loads were obtained from CFD analysis, and the clearance flow was taken into consideration in the CFD simulation. The variation of performance characteristics was calculated through the one-dimensional method of characteristic (1D-MOC). The results show that the pressure on outer surfaces of runner changed the stress of the runner, and the rotation rate had a great influence on the dynamic stresses of the runner.

1. Introduction

Pumped storage units play an important role in the power grid, solving the problem of frequency modulation, phase modulation as well as peak regulation\textsuperscript{[1]}. In addition, as a way of energy storage, pumped storage units can absorb wind and solar energy on a large scale, improving the power quality\textsuperscript{[2; 3]}. In order to meet the dispatching demand of power grid, pumped storage units are obliged to work in different operating modes, causing the occurrence of transient process. The variation of performance characteristics is severe during transient process, and the flow field changes fast and complicatedly, which can cause serious damage on the pump-turbine runner. As the core part of pumped storage units, the damage of runner will bring great loss to power station.

Many researches focused on the unsteady flow characteristics during the transient process. Xiaolong Fu\textsuperscript{[4]} et al studied the transient flow of a pump-turbine during load rejection process. The authors found that unsteady vortex flows appeared when the torque on blades were near to zero and impacted the guide vanes and stay vanes region in reverse pump operating process. Nicolle J et al\textsuperscript{[5]} conducted a CFD simulation for a hydraulic turbine start up and the results showed that rotating stall phenomenon occurred at speed-no-load.

Nowadays there are some studies discussing the dynamic stresses in transient process by experiment\textsuperscript{[6-8]} although measuring the transient dynamic stresses of runner is a difficult task because it’s hard to distinguish whether the strain is caused by mechanical vibration or by the deformation of the runner. Numerical methods can be a better way to solve the problem. He Lingyan et al\textsuperscript{[9]} calculated
the dynamic stresses of runner during start up and discussed the resonance phenomenon in the process of runner acceleration. However, there are still many problems that need further study.

In this paper, the dynamic stresses of runner during load rejection process were studied using fluid–structure coupling method. The change of performance characteristics was calculated by 1D-MOC. The discharge of the unit as well as the pressure at the outlet of draft tube were also obtained through 1D-MOC and were used as the boundary conditions of CFD analysis. The fluid structure interaction (FSI) simulation was conducted to obtain the dynamic stresses of runner. Stress distribution and stress concentration were discussed, and the change of stress and its influencing factors were analyzed in this paper.

2. Computational domain and numerical method

The prototype of pump–turbine consists of draft tune, casing, 16 stay vanes, 16 guide vanes, 5 blades and 5 splitter blades, as shown in figure 1. In attention, the clearance between runner and head cover as well as the clearance between runner and head cover as well as the clearance between runner and bottom leakage ring were also taken into account, as shown in figure 2.

![Figure 1. The pump-turbine](image1)

![Figure 2. The clearance of runner](image2)

The layout of three-units with one diversion tunnel was chosen during the construction of the pumped storage power station. Load rejection of three units at the same time is one of the most dangerous transient processes, and the safety of runner need to be guaranteed during this process.

The external characteristics such as discharge, head, torque on the runner, rotation rate etc. were calculated by 1D-MOC. 13 typical time points were chosen for further analysis, including initial operating condition, the operating condition with maximum rotating rate and with maximum reverse discharge.

The hydraulic loads in 13 time points were obtained by CFD simulation. The mesh of whole unit was shown in figure 3. The boundary conditions in the simulation were all gained through the results of 1D-MOC. SST k-ω turbulence model was used for its better prediction of boundary layer separation.
The finite element model of runner was shown in figure 4. It was made of stainless steel and the properties were shown in table 1. The external forces on the runner include gravity, centrifugal force and hydraulic forces. As the time interval of 13 time points was longer than 1.9s, Euler Force was not taken into account.

Table 1. The properties of stainless steel

| Properties          | Value       |
|---------------------|-------------|
| Young's modulus     | $2.1 \times 10^{11}$ Pa |
| Poisson's ratio     | 0.3         |
| Density             | 7850 kg/m$^3$ |

2 nodes were selected to check the grid independence. R1 is located at the circle area near the fixed support, and R2 is located at the fillet of leading edge and band. The result of grid independence check is shown in figure 5. A total node number of 305,505 is enough for the stress analysis.
3. Results and discussion

3.1. Variation of performance characteristics
The process of load rejection of three units at the same time was calculated. Three units all operated at 100% load and at rated head and the emergency occurred at 0s and the guide vanes of the three units were closed according to the set rule after a delay of 0.2s. The time history of performance characteristics of 1# unit was shown in figure 6. The 13 time points calculated by CFD simulation are also shown in figure 6. The discharge and torque of the runner both decreased as guide vane closed and became negative after few seconds. The reverse torque can reach 0.69 times of the rated value, sometimes causing severe damage on the shaft. When the torque dropped to zero, the rotation rate reached the maximum value, which was 1.336 times of rated rotation rate. The guide vane fully closed at 60s and discharge was close to zero. At the same time, 1# unit was at no load.

3.2. Dynamic stresses of pump-turbine runner
The stress distribution of runner at 0s was shown in figure 6. During the load rejection process, the stress distribution was similar to that at 0s. The stress concentration located at three regions: the annular area near the leak stop ring, shown in figure 6(a); as well as the blending part between the suction side of blade and band ring, shown in figure 6(b); the T-joint of leading edge and crown, shown in figure 6(c). The shape of this pump-turbine runner is more flat than that of Francis turbine,
and its rotation rate is high (500 rpm to 668 rpm), therefore the centrifugal force is of great importance on the stress of runner. And the high rotation rate caused the high stress at annular area near the leak stop ring.

![Figure 7. The Von-mises stress distribution of the runner](image)

The pressure on the outer surface (OS) of runner also plays a significant role on the stresses of runner. The surfaces of the runner can be divided into 2 parts: inner surface (IS) and outer surface, as shown in figure 8. The pressure on OS was not taken into account in traditional FSI simulation, and the result of this method was shown in figure 9(a). The maximum stress was 450MPa and located at the T-joint of leading edge and band ring. However, the result can be totally different when pressures on both OS and IS were used as hydraulic loads, as shown in figure 9(b). The maximum stress was 110 MPa and the stress at T-joint of leading edge and band ring was smaller than that at T-joint of leading edge and crown.

![Figure 8. The surfaces of the runner](image)
Figure 9. The stress distribution of runner using different FSI simulation methods

Four monitor points were set in the high stress area of the runner, whose location were shown in figure 7. The time history of the dynamic stress is shown in figure 10. It was similar with that of rotation rate, indicating that rotating rate was the mean factor influencing dynamic stress.

Moreover, the dynamic stress of s1 was almost equal to that of s3, and was larger than s4. The dynamic stress of s2 was the lowest one. Point s1 located at the depression part of the crown, and the mechanical strength is lower than other parts. Meanwhile, this location is closed to the fixed support region, and the centrifugal force of the whole runner works on this annual area, therefor its stress is largest. The geometry changes at T-joint of leading edge and band ring, causing stress concentration. The splitted blades suffered larger stress because of its thinner thickness.

Figure 10. The dynamic stress of the runner and rotating rate vs time
4. Conclusion
The dynamic tresses of a pump-turbine runner during load rejection process were calculated using FSI method. The performance characteristics were simulated by 1D MOC. At the same time, the discharge of casing inlet and the pressure of draft tube outlet were also calculated, and used as the boundary conditions of CFD analysis. The pressure on the outer surface of runner was analyzed because clearance flow was taken into consideration, and it changed the stress distribution of runner. The high rotation rate in load rejection process resulted in high centrifugal force, which was the main reason of the high stress in annular area of crown.

References
[1] Deyou L, Xiaolong F, Zhigang Z, Hongjie W, Zhenggui L, Shuhong L and Xianzhu W 2019 Investigation methods for analysis of transient phenomena concerning design and operation of hydraulic-machine systems—A review Renewable and Sustainable Energy Reviews 101 26-46
[2] Liu X, Luo Y and Wang Z 2016 A review on fatigue damage mechanism in hydro turbines Renewable and Sustainable Energy Reviews 54 1-14
[3] Trivedi C, Gandhi B and Michel C 2013 Effect of transients on Francis turbine runner life: a review J of Hydraulic Research 51(2) 121-32
[4] Xiaolong F, Deyou L, Hongjie W, Guanghui Z, Zhenggui L and Xianzhu W 2018 Analysis of transient flow in a pump-turbine during the load rejection process J of Mechanical Science and Technology 32(5) 2069-78
[5] Nicolle J, Giroux A and Morissette J 2014 CFD configurations for hydraulic turbine startup 27th IAHR Symposium on Hydraulic Machinery and Systems
[6] Gagnon M, Nicolle J, Morissette J and Lawrence M 2016 A look at Francis runner blades response during transients 28th IAHR symposium on Hydraulic Machinery and Systems
[7] Gagnon M, Tahan S, Bocher P and Thibault D 2010 Impact of startup scheme on Francis runner life expectancy 25th IAHR Symposium on Hydraulic Machinery and Systems
[8] Huang X, Chamberland-Lauzon J, Oram C, Klopfer A and Ruchonnet, N 2014 Fatigue analyses of the prototype Francis runners based on site measurements and simulations 27th IAHR Symposium on Hydraulic Machinery and Systems
[9] Lingyan H, Lingjiu Z, Soo-Hwang A, Zhengwei W, Yusuke N and Sadao K 2019 Evaluation of gap influence on the dynamic response behavior of pump-turbine runner Eng Computation 36(2) 491-508.