Experimental study on load rejection process of a model

tubular turbine

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Abstract: To obtain the dynamic parameters variation characteristics of a model tubular turbine during the load rejection process, a model test method was designed and carried out in this paper. Firstly, the model test system, the test method and the model tubular turbine parameters were introduced. Then, the experiments of different schemes were performed and the results were analyzed. Evidently, the closing rules of guide vanes and runner blades have influence on the quality of transient process. When the guide vanes undergo linear needle closure law, the effect of prolonging the blade closing time on reducing the maximum speed appreciation and the maximum reverse water thrust is obvious, and when the guide vanes is closed in two stages, the longer the blade closing time, the lower the maximum reverse hydraulic thrust value. With the same closing law, the segmental closure law with guide blade first fast and then slow is more obvious than the linear needle closing law in reducing the maximum reverse water thrust, and the selection of closure segment point of guide vanes has a certain effect on the appreciation of the maximum rotating speed. Thus, the model test method could be used in transient operation improvement at hydropower plants.

Keywords: Model tubular turbine; load rejection; closing rule; transient characteristics

1. Introduction

With the sharp deterioration of the contradiction between energy supply and environmental protection in the world, low-head turbines have become the top priority in the development and utilization of renewable energy due to low investment, short construction period and small negative impact on the environment [1-2]. At present, the tubular turbine is an ideal type because it can adapt to the conditions of large flow and low water head, and has dual regulation function.

In the daily operation of conventional hydropower stations, frequent changes in operating conditions inevitably lead to transient process of turbines [6-9]. Load rejection is a common transient process, and dynamic parameters such as rotational speed, mass flow rate, torque, axial thrust and static pressure fluctuate greatly. In the process of load rejection, due to the increase of rotating speed and system water pressure, the flow pattern in the channel becomes more complicated, which leads to drastic changes in the change rule of the characteristics outside the unit, resulting in additional dynamic load and strong hydraulic pulsation and vibration, thus posing a huge threat to the safe operation of the whole power station [10]. Therefore, it is of great significance to ensure the operation safety of power station to adopt proper regulation and control scheme for turbine components in the transition process.

At present, the experimental research on hydraulic machinery and system transition process is mainly based on field measurement. The flow water system pressure fluctuations, turbine pressure vibration and swing, shaft centerline orbit parameters during the unit normal starting and stopping, the load increase or decrease, and the load rejection process were measured and analyzed to evaluate
and improve the unit and the hydraulic system transition process quality [11] [12]. However, due to the high cost of the test and the imperfection of the similarity rule of the original model, there are few researches on the transition process model test. Limited studies involve centrifugal pump starting process [13], PIV measurement of transient flow field during oblique flow pump starting process [14], starting and stopping of high-head Francis model unit [15], pressure pulsation characteristics during variable-speed and variable-flow process [16], and dynamic characteristics of variable-speed transition process of model pump turbine [17], etc.

In this study, the test system for model tubular turbine transition process was specially built. Then, the test method and the model tubular turbine parameters were introduced, and the different closing law of guide vanes and runner blades during load rejection of tubular turbine was designed. Finally, the experiments of different schemes were performed and the results were analyzed to reveal the model turbine transient characteristics.

2. Test system

2.1 Test bench structure

The test bed is composed of pumps, pipelines, valves, water pools, unit test section and so on, shown as Figure 1. Circulating water is fed into the upper pool by circulating pump from the bottom pool, then flows into the lower pool through model unit, and then enters the flow measurement weir tank and returns to the bottom pool. Both upstream and downstream pools have overflow plates to adjust the water level to meet the requirements of test head and stable upstream and downstream water level. Figure 2 is the schematic diagram of the model unit test device, which is composed of DC power measuring motor 2, electromagnetic clutch 3, torque meter 4, model runner 7, guide blade operating mechanism 9 to 11 and propeller blade operating mechanism 14 to 16.

![Figure 1](#)

**Figure 1.** The schematic diagram of the model test loop.
Figure 2. The schematic diagram of model device test, where 1-photoelectric speed measuring device; 2- power measuring motor; 3- electromagnetic clutch; 4- torque meter; 5- coupling knot; 6-connecting rod mechanism; 7 - runner; 8- bevel gear; 9- control ring; 10- nut – lead screw; 11- stepping motor; 12- preloaded spring; 13- pressure sensor; 14- nut - lead screw; 15- cycloid needle wheel reducer; 16- DC motor.

2.2 Measurement and control system

The measurement and control device was designed by Programmable Logic Controller (PLC) to complete automatic control, automatic data acquisition, recording, data processing and data graph output. The software of measurement and control device adopts "menu" design technology and modularization of program calculation. Therefore, it is only necessary to adjust the working condition before load rejection and keep it stable. Once the instruction is issued, all the control work is automatically completed by the microcomputer measurement and control device.

2.3 Parameters measurement method

2.3.1 Measurement method of torque and rotating speed

The dynamic torque and rotation speed of the turbine runner were measured by a high-precision JCZ intelligent torque speed sensor. In the case of static check, the torque measurement accuracy was ± 0.1%, the measurement accuracy of the rotation speed was ± 0.1%, and the frequency response was 10 kHz. The speed and torque signals were input into the microcomputer for recording and storage.

2.3.2 Measurement method of blades and guide vanes opening

In dynamic test, the path length of guide vanes and runner blades was reflected by a multi-coil helical potentiometer so as to obtain the guide vane opening and blade opening, and total resistance tolerance is ±1% full scale (FS). Before the dynamic test, the potential of the potentiometer was calibrated with the opening of the guide vanes and blades. The computer directly collected the change of potential value.
2.3.3 Measurement method of water pressure

For water pressure, conventional PX series pressure sensors with high precision and fast response speed were installed at the corresponding measuring points during the dynamic test. The measurement accuracy is ± 0.1% FS.

2.3.4 Measurement method of axial force

Axial forces were measured using resistance strain pressure sensors. An auxiliary shaft was drawn from the discharge cone of the runner, and a disk was mounted to the end of the auxiliary shaft with a set of bearings. Three tension pressure sensors were evenly distributed at the same radius of the disc. Meanwhile, three adjustable springs were set on the back of the disc to preload pull pressure sensors. Preloading is greater than the possible maximum negative axial force, and this causes pull pressure sensors in the dynamic test to be in a state of compression, which improves the measurement accuracy. To overcome the influence of static friction force, the sensors were calibrated directly by hanging weights. The output signal of a sensor was input to the PLC. Owing to problems of manufacturing and installation, the accuracy of the measurement of axial force is slightly low. In the static test, the sensitivity was ± 9.8 N. In the dynamic test, due to the addition of a set of blade operating mechanisms, the sensitivity was ± 14.7 N and the error was ± 4%.

3. Test methods

In the test process, under a certain guide blade opening and blade angle, the operating condition of the turbine can be changed by the unit speed, which is realized by changing the armature voltage of the DC motor, leaving the head unchanged. Thus, the model machine can be operated at a similar working condition according to the initial simulation conditions. The electric energy generated by the DC power measuring motor is fed into the power grid through the thyristor three-phase full-bridge rectifying inverter.

The stepper motor rotates the nut via the coupling to make the screw rod of the push-pull rod move forward and backward, driving the control ring so as to close or open the guide vanes through the connecting rod mechanism 6. The stepping motor 12 can rotate 2.4° per pulse. The pulse generator sends out different clock pulses to the stepping motor controller through the PLC program according to the guide vane closing rule, which changes the output rectangular wave frequency. The pulse is divided into five phases by the stepper motor controller, and the power is amplified and sent to the stepper motor to control the operation of the guide vane. To change the frequency of the pulse, the closing speed of the guide vane can be changed.

The speed regulating motor drives the screw rod to rotate through the cycloid needle wheel reducer, causing the nut connected to the blade operating shaft to move forward and backward. The blade operating cross is driven by the blade operating shaft and rotates the blades by the connecting rod mechanism to open or close them. The blade rotating speed relies on the armature voltage of the speed regulating motor. Tests confirm the RB closing linearity, and the closing time can be adjusted between 2 and 40 s. The entire blade operating device is suspended on a disc connected to an auxiliary shaft to measure axial forces so as to internally balance the blade operating force without affecting the measurement of axial forces during the transient process.

4. Model parameters

Some basic parameters of bulb tubular model turbine are as follows: nominal diameter of model runner is 0.25 m, guide vane number is 16, guide vane tape angle is 60 °, and runner blade
number is 4. The inertia moment of the model unit is 2.88 N.m². The designed water head is 1.0m, the rotation speed is 712 r/min, and the flow rate is 170 l/s. Figure 3 shows a schematic diagram of the structure of the tubular turbine. The closure law of guide vanes and runner blade are provided in Figure 4.

Figure 3. 3D sketch of the tubular turbine model.

Figure 4. Schematic diagram of related parameters of closure law of guide vane and blade.

5. Test results and analysis

5.1 Experimental study on the transient process under different blade closing rules

Case 1: Under the design water head condition, the load and the guide blade closure law is the same. The guide vanes were closed in a straight line with a closing time of 5 seconds. The test results obtained by using different blade closing rules are shown in Table 1.
Table 1. Test results of different closing time of blade with the same linear needle closing law of guide vanes.

| Scheme | $T_z$ / s | $n_{\text{max}}$ / % | $p_{\text{max}}$ / % | $F_{\text{max}}$ / % |
|--------|-----------|----------------------|----------------------|----------------------|
| 1      | 18.6      | 47.9                 | 27.6                 | -58.5                |
| 2      | 34.9      | 46.6                 | 26.3                 | -55.1                |
| 3      | 50        | 44.3                 | 25.6                 | -54.2                |

Table 1 shows that under the same law of guide blade straight line closing, with the increase of blade closing time, the maximum values of parameters such as rotation speed, pressure and reverse hydraulic thrust all decrease. Therefore, when the guide blade straight line is closed, lengthening the blade closing time is beneficial to the transition process of the unit. Especially to reduce the maximum speed on the appreciation of maximum reverse hydraulic thrust effect is more obvious. Therefore, lengthening the blade closing time can be used as one of the measures to slow down the speed of the power plant and reduce the maximum reverse hydraulic thrust.

Case 2: Under the design water head condition, the load and the guide blade closure law is the same. The guide vanes were closed using the same two-stage closing rule. The test results obtained using different blade closing rules were shown in Table 2.

Table 2. Test results of different closing time of blade with the same two steps closing law of guide vanes.

| Scheme | $T_z$ / s | $n_{\text{max}}$ / % | $p_{\text{max}}$ / % | $F_{\text{max}}$ / % |
|--------|-----------|----------------------|----------------------|----------------------|
| 1      | 27.7      | 39                   | 17.3                 | -43.1                |
| 2      | No movement | 38.9                 | 17.2                 | -39.3                |

As can be seen from Table 2 above, the longer the blade closing time is, the greater the maximum reverse hydraulic thrust value is, which is the same as the conclusion above that the guide blade straight line is closed. Based on the above experimental results, it can be seen that the longer the blade closing time is beneficial to reduce the maximum reverse hydraulic thrust when the blade is involved in the transition process of adjustment.

5.2 Experimental study on the transient process under different guide vane closing rules

The test results obtained by using different guide vanes closing rules were shown in Table 3. It can be seen from Table 3 that under the design head working condition and with the same load, the guide blade adopts different closing rules at the same blade closing rule, and the changes of maximum speed appreciation, maximum pressure appreciation and maximum reverse hydraulic thrust are all great. Therefore, the closing rule of guide vane has a great influence on the quality of transition process.

Table 3. Test results of different closing time of guide vanes with the same closure rules of blades.

| Scheme | $T_z$ / s | $Y_d$ / ° | $T_{\alpha 1}$ / s | $T_{\alpha 2}$ / s | $n_{\text{max}}$ / % | $p_{\text{max}}$ / % | $F_{\text{max}}$ / % |
|--------|-----------|-----------|-------------------|-------------------|----------------------|----------------------|----------------------|
| 1      | 24.7      | 27.7      | 5                 | 20                | 39.2                 | 27.6                 | -52.7                |
| 2      | 24.7      | 34.7      | 8                 | 32                | 42.6                 | 16.5                 | -44.9                |
| 3      | 24.7      |           | 8                 | -                 | 41                   | 21.9                 | -56.2                |

5.3 Variation of parameters in load rejection process of the model tubular turbine
Taking the test results of 100% load rejection at design water head condition as an example, Table 4 illustrates the closing scheme of guide vane and blade in the load rejection test. The test results are provided in Table 5, and Figure 5 demonstrates the parameter change curve of the model test during the load rejection.

| Test scheme | $T_z$/s | $Y_d$/° | $T_{s1}$/s | $T_{s2}$/s |
|-------------|---------|---------|------------|------------|
|             | 40.0    | 34.7    | 5.0        | 25.0       |

Table 5. Test results of 100% load rejection at design head condition.

| $H$/m | $a_0$/° | $\varphi_0$/° | $n_{max}$/% | $P_{max}$/% | $F_{max}$/% |
|-------|---------|--------------|-------------|-------------|-------------|
| 1.00  | 69.3    | 15.28        | 34.0        | 28.0        | -51.4       |

Figure 5. Variation of normalized dynamic parameters in the load rejection.

According to the test results, the maximum speed of the unit increases by 34%, the maximum pressure appreciation is 28%, and the maximum negative axial force appreciates by 51.4% when the design head is used to dump the full load.
From the pressure variation curve in Figure 5, during the time when guide vane and blade do not operate, the flow rate increases due to the climb of rotating speed, so there is an obvious pressure decrease on the pressure process curve before and after the guide vanes. However, there is an obvious pressure rise on the draft tube pressure process curve, which is determined by the flow characteristics of the tubular unit. Furthermore, pressure variation after the guide vanes is much larger than that before the guide vanes. At this time, the guide vanes have a large opening, and the pressure change before guide vanes and after guide vanes is similar, which is opposite to the pressure change of the draft tube. Therefore, the blade will become the interface of water hammer pressure when the opening of the guide blade is large, which is a unique phenomenon of tubular unit.

From the experimental axial force and rotating speed curve, it can be seen that the maximum reverse water axial force can also appear at the segmented point as long as the position of the segmented point is appropriate, which indicates that the position of the segmented point can control the maximum reverse axial force. Meanwhile, the speed rise quickly reaches the extreme value, which is due to the small inertia moment of tubular unit, so it is easy to over-speed in the actual project, which should be paid attention to in operation.

6. Conclusions

The power consumption of model test is small, which is easy to observe and measure, and the initial working conditions and regulating elements are easy to adjust, and accidents are generally not easy to occur. Therefore, this paper carried out the test study of dynamic process of the model tubular turbine from the perspective of economy, flexibility and safety. Dynamic test of hydraulic turbine model has complexity and particularity, which needs to rely on the test bed with higher degree of automation, as well as the accurate and rapid response actuator and data acquisition system. According to the dynamic characteristics of hydraulic turbine model test, the closing rules of guide vanes and blade have influence on the quality of transit process. When the guide vanes undergo linear needle closure law, the effect of prolonging the blade closing time on reducing the maximum speed appreciation and the maximum reverse water thrust is obvious, and when the guide vanes is closed in two stages, the longer the blade closing time, the lower the maximum reverse hydraulic thrust value. With the same closing law, the segmental closure law with guide blade first fast and then slow is more obvious than the linear needle closing law in reducing the maximum reverse water thrust, and the selection of closure segment point of guide vanes has a certain effect on the appreciation of the maximum rotating speed. Therefore, in order to optimize the quality of the transition process, it is necessary to use the optimization method to find the best closing law. Furthermore, the moment of inertia of tubular unit is small, the rotating speed rises rapidly and reaches the extreme value quickly. Attention should be paid to prevent the unit from over-speed in actual operation.

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