Research on the Impact of Double-fed Wind Turbine Generator Drive System Modeling on Torsional Vibration Frequency

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ABSTRACT: With the continuous increase of the wind power development strength in recent years, the rapid wind power development causes a series of problems. Based on the mechanism of the torsional vibration of double-fed wind turbine generator system, this paper researches the impact of the main drive system modeling method on the torsional vibration frequency. This paper first establishes mathematical models of three mass blocks and two mass blocks of the drive system, and then calculates the natural vibration frequency of two kinds of mathematical models by use of the small perturbation method and Dunkerley method, and also obtains the sensitivity of mechanical system parameters to the natural vibration frequency through calculation of the sensitivity, and finally obtains the difference of different mathematical models under different fault conditions through the numerical calculation and example simulation, thus providing a theoretical basis for correctly selecting the mathematical models.

Keywords: double-fed wind turbine generator system; torsional vibration; natural frequency; small perturbation method; Dunkerley method

1 INTRODUCTION

At present, the megawatt high-capacity wind turbine generator system (WTGS) is subject to the variable-speed constant-frequency models, which mainly adopt double-fed permanent-magnetic direct-drive technologies. For the common characteristic of these two models, the speed governing and conversion of electrical energy are dependent on a high-power converter. In China, the double-fed variable-speed constant-frequency induction generator becomes the most widely used commercial model due to its advantages, which can track the optimal wind speed in a wider range and improve the utilization of wind energy to a greater extent. However, in the actual operation of the double-fed induction generator, the randomness and impact of the wind speed, and the snap action of the power electronic devices and the perturbation of the power grid in a high-power converter are likely to cause the shafting vibration between the wind turbine and generator, and even lead to malfunction, thus causing shutdown. The typical faults caused by the shafting vibration mainly manifest in the blade fracture, shaft and bearing cracks or even broken shaft, broken teeth of gearbox and so on [17]. It not only causes huge maintenance costs, but also wastes a lot of wind energy resources.

In terms of the damages to the torsional vibration of shaft system, this paper focuses on the calculation of the natural frequency of the main drive system of the double-fed generator unit through the small perturbation method and Dunkerley method, and also analyzes the impact of the structural parameters of the wind power drive system on the torsional vibration frequency by the use of the sensitivity analysis, and compares with different modeling methods, so as to determine the applicable conditions of the modeling methods.

This paper consists of five parts: the second part is the three mass and two mass block modeling for the double-fed wind turbine generator drive system; the third part is the calculation of the natural vibration frequency of the drive system by the use of the small perturbation method and Dunkerley method; the fourth part is the numerical calculation and simulation contrast for different modeling methods, and focuses on comparing with the simulation effect of different modeling methods under different conditions of the system fault. The fifth part is the conclusion of the whole paper.
DOUBLE-FED WIND TURBINE GENERATOR DRIVE SYSTEM MODELING

The simulation of the torsional vibration of shaft system mostly adopts the mass concentration model. The mass concentration model is to concentrate the rotational inertia on the shaft segment of the device on each part based on the concrete structure of the shaft system, which is represented by the mass block, and then the ideal spring is used to connect with the concentrated mass blocks, in order to simulate the flexibility between the shaft segments. The shaft system of the double-fed wind turbine generation system is usually equivalent to the five mass blocks, three mass blocks, two mass blocks and a single mass block. Comparatively speaking, the five mass blocks can be used to further accurately describe the dynamic behavior of the shaft system of the wind turbine system, so that the simulation results will be more close to reality. However, the shaft system of the wind turbine system does not exist in isolation, and it is necessary to consider the generator, control system, power system and many other dynamic conditions in the process of simulation, so it will increase the difficulty of calculation, and the simulation is generally carried out after appropriate simplified processing. If it is simplified to a single mass, the difficulty of calculation is greatly reduced, but it is difficult to accurately reflect the dynamic characteristics of the important link in the research of the relevant issues of the torsional vibration due to ignoring the stiffness coefficient and damping coefficient. Therefore, this paper carries out a deep analysis and contrast of the torsional vibration of the shaft system of the double-fed wind power generation system mainly by the use of the three mass and two mass blocks.

2.1 Mathematical model of three mass blocks

The wind turbine rotor, gearbox and generator are equivalent to a concentrated rotational inertia, corresponding to the connection between three mass blocks via the ideal spring. Thus, the main drive system can be simplified as the torsional vibration mechanical model shown in Figure 1. The differential equation can be established according to the basic laws of Newtonian dynamics, and its differential equation of motion can be expressed as

\[ J_{wt} \frac{d\omega_{wt}}{dt} = k_{wt \_ 1}(\theta_{1} - \theta_{wt}) - D_{wt} \omega_{wt} + D_{wt \_ 1}(\omega_{1} - \omega_{wt}) + T_{wt} \]

\[ J_{1} \frac{d\omega_{1}}{dt} = k_{wt \_ 1}(\theta_{1} - \theta_{wt}) + k_{1 \_ g}(\theta_{g} - \theta_{1}) - D_{1} \omega_{1} + D_{wt \_ 1}(\omega_{1} - \omega_{wt}) \]

\[ J_{g} \frac{d\omega_{g}}{dt} = k_{1 \_ g}(\theta_{g} - \theta_{1}) - D_{g} \omega_{g} + D_{1 \_ g}(\omega_{g} - \omega_{1}) - T_{g} \]  

(1)

Where: the subscript \( wt, 1 \) and \( g \) respectively represent the following three mass blocks: wind wheel, gearbox and generator. \( T \) represents the torque; \( J \) represents the rotational inertia; \( D \) represents the damping of shaft system; \( \omega \) represents the revolving speed; \( \theta \) represents the angle of torsion; \( k_{c} \) represents the torsional stiffness; \( k_{wt \_ 1} \) and \( k_{1 \_ g} \) respectively represent the coupling rigidity between the mass blocks, which can be obtained by the following equation:

\[ \frac{1}{k_{wt \_ 1}} = \frac{1}{k_{wt}} + \frac{1}{2k_{1}} \]

\[ \frac{1}{k_{1 \_ g}} = \frac{1}{k_{1}} + \frac{1}{2k_{g}} \]

2.2 Mathematical model of two mass blocks

The influence of the gearbox is equivalent to the wind turbine and the rotational inertia and damping of the coupler is equivalent to the motor rotor, that is, the wind turbine and generator respectively correspond to two mass blocks, as shown in Figure 2. Due to the combination with the low-speed shaft and high-speed shaft based on the model of three mass blocks, for the rotational inertia, damping, torque and other parameters in the model, there is a need to convert the
high-speed shaft to the low-speed shaft or convert the low-speed shaft to the high-speed shaft. The differential equation can be established according to the basic laws of Newtonian dynamics. Based on the principle of the unchanged kinetic energy, energy loss and power before and after the conversion of the mass blocks [58], after conversion the high-speed shaft to the low-speed shaft, the differential equation of motion of the two mass blocks can be obtained [155]:

\[
J_w \frac{d\omega_w}{dt} = k_s (\theta_e - \theta_w) - D_w (\omega_e - \omega_w) + T_w
\]

\[
J_g \frac{d\omega_g}{dt} = k_s (\theta_e - \theta_g) + D_w (\omega_e - \omega_g) - T_g
\]

\[
d\theta_e = \omega_w
\]

\[
d\theta_g = \omega_g
\]

In the above equation, \(D_w\) is the damping coefficient of the shaft system between two mass blocks; \(k_s\) is the torsional rigidity of the revolving shaft of two mass blocks. They can be calculated by the following equation:

\[
\frac{1}{k_s} = \frac{1}{k_{ws}} + \frac{1}{k_{wg}}
\]

3 ANALYSIS OF TORSIONAL VIBRATION OF SHAFT SYSTEM OF WIND TURBINE GENERATOR

3.1 Natural frequency of shaft system of wind turbine generator

3.1.1 Small perturbation method

For a dynamical system, its motion state can be described by a set of differential equations. Therefore, the equation of motion of the shaft system of the wind turbine generator can be written as [166, 167]:

\[
\begin{cases}
\dot{x} = f(x, y, \mu) \\
0 = g(x, y, \mu)
\end{cases}
\]

Where: \(x\) is the differential variable of the system; \(y\) is an algebraic variable; \(\mu\) is a parameter of the system.

When the system is subjected to a small perturbation, after linearization of the model of the shaft system at the equilibrium point, the equation of model’s dynamic state can be obtained: \(f(\bullet)\) is used to describe the dynamic behavior of the shaft system; \(g(\bullet)\) is used to describe the relationship between the differential variable and algebraic variable. After linearization of the equation (1) at the equilibrium point, then:

\[
\begin{cases}
\Delta \dot{x} = f_0(x_0, y_0, \mu_0) \Delta x + f_1(x_0, y_0, \mu_0) \Delta y \\
0 = g_0(x_0, y_0, \mu_0) \Delta x + g_1(x_0, y_0, \mu_0) \Delta y
\end{cases}
\]

If \(g_1\) is a non-singular, the above equation can be simplified as:

\[
\Delta \dot{x} = A(x_0, y_0, \mu_0) \Delta x
\]

Where: \(A = f_0 - f_1 g_0^{-1} g_1\) is the state matrix, which describes the dynamic characteristic of the nonlinear system, and its eigenvalue can be used to calculate the system vibration frequency and vibration mode [168].

1) Natural frequency of model of three mass blocks

After linearization of the equation (1) at the equilibrium point, then:

\[
\begin{bmatrix}
X_{sm} \\
\dot{X}_{sm}
\end{bmatrix} = \begin{bmatrix}
A_{sm} \\
B_{sm}
\end{bmatrix} U_{sm}
\]

Where:

\[
A = \begin{bmatrix}
D_w & 0 & -k_{ws} & -k_{ws} \\
0 & -D_w & J_w & J_w \\
0 & -D_w & J_w & J_w \\
1 & 0 & 0 & 0
\end{bmatrix}
\]

\[
B = \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0
\end{bmatrix}
\]
\[ \begin{bmatrix} \Delta \omega_{mx} & \Delta \omega_{m1} & \Delta \omega_{mg} & \Delta \theta_{mx} & \Delta \theta_{m1} \end{bmatrix}^T; \]

\[ \begin{bmatrix} \frac{1}{J_{wt}} & 0 & 0 & 0 & 0 \\ 0 & \frac{1}{J_1} & 0 & 0 & 0 \\ 0 & 0 & \frac{1}{J_g} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}; \]

\[ \begin{bmatrix} \Delta \omega_{mx} & \Delta \omega_{m1} & \Delta \omega_{mg} & \Delta \theta_{mx} & \Delta \theta_{m1} \end{bmatrix}^T; \]

\[ \begin{bmatrix} \Delta T_{wt} & \Delta T_{g1} & \Delta T_{g2} & 0 & 0 \end{bmatrix}^T. \]

Equation (6) is an equation of the dynamic state of the equivalent model of shaft system of three mass blocks. According to the vibration theory, in the absence of external torque, \( \Delta T_{wt} = \Delta T_{g} = 0 \). After calculation of the eigenvalue and characteristic vector of \( X_m = A_{sm} X_m \), the natural frequency and vibration mode of the drive system of three mass blocks can be obtained. According to the linear algebra theory, the solution to the natural frequency and main vibration mode of the drive system can be translated into solving the following equation:

\[ A_{sm} P = \lambda P \]  

(7)

\( \lambda \) is the eigenvalue of \( A_{sm} \), and the corresponding non-zero column vector \( P \) is the eigenvector corresponding to \( \lambda \). The imaginary part of the conjugate complex roots in the eigenvalues is the natural frequency, and the eigenvector \( c \) is the corresponding vibration mode [169]. Therefore, the solution of the vibration frequency of the shaft system and vibration mode is translated into solving the determinant in the equation (7), that is, to solve the determinant:

\[ |\lambda I - A_{sm}| = 0 \]  

(8)

2) Natural frequency of model of two mass blocks

After linearization of the equation (2) at the equilibrium point, the equation of the model’s dynamic state of two mass blocks can be obtained:

\[ \dot{X}_{lm} = A_{lm} X_{lm} + B_{lm} U_{lm} \]  

(9)

Where:

\[ A_{lm} = \begin{bmatrix} D_{lm} & -D_{lm} & -k_{lm} & k_{lm} \\ J_{wt} K_{g1} & J_{wt} K_{g2} & J_{wt} K_{g3} & J_{wt} K_{g4} \\ \frac{D_{lm}}{J_g K_{g1}} & -\frac{D_{lm}}{J_g K_{g2}} & -k_{lm} & k_{lm} \\ 0 & 0 & 0 & 0 \end{bmatrix}; \]

3.1.2 Dunkerley method

The drive shaft of the wind power generation system is a strong coupling system. The rapid estimation of the fundamental frequency of the system is very crucial to the research of the torsional vibration of the shaft system. The Dunkerley method can be more easy and fast to estimate the fundamental frequency of the system oscillation.

According to the basic laws of Newtonian dynamics, the dynamic model described by the equation (6) can be written as the differential equation of the non-damping system:

\[ J \frac{d^2 \theta}{dt^2} + k \theta = T \]  

(10)

Where:

\[ J = \text{diag}[J_{wt} \quad J_1 \quad J_g]; \quad \theta = [\theta_{wt} \quad \theta_1 \quad \theta_g]^T; \]

\[ k = \begin{bmatrix} k_{wt,1} & -k_{wt,1} & 0 \\ -k_{wt,1} & k_{wt,1} + k_{s1,g} & -k_{s1,g} \\ 0 & -k_{s1,g} & k_{s1,g} \end{bmatrix}; \]

\[ T = [T_{wt} \quad 0 \quad T_g]^T. \]

If the non-diagonal element of the matrix \( k \) is not 0, it indicates mutual coupling between the mass blocks and mutual effect of the dynamic characteristics of the adjacent modules. In the research of the characteristics of the free oscillation of the drive system, generally assuming that the external torque is 0, the equation (10) can be written as:

\[ J \frac{d^2 \theta}{dt^2} + k \theta = 0 \]  

(11)

Three oscillation frequencies of the system can be obtained by solving the equation (11), of which the
minimum non-zero frequency is the fundamental frequency, which is the main vibration mode of the system for the wind turbine.

The essence of solving the equation (11) is to solve the eigenvalue and modal vector, which can be translated into the following equation [168]:

$$\frac{1}{\omega^2} K P = J P$$

(12)

Where: \(\omega\) is the natural frequency; \(P\) is the corresponding modal vector of the natural frequency.

Assuming that \(M = K^{-1} J\), then

$$\begin{bmatrix}
k_{11}J_{wt} & k_{12}J_1 & k_{13}J_g \\
k_{21}J_{wt} & k_{22}J_1 & k_{23}J_g \\
k_{31}J_{wt} & k_{32}J_1 & k_{33}J_g
\end{bmatrix}$$

The equation (12) can be written as [169]:

$$\frac{1}{\omega^2} P = MP$$

The characteristic equation of the above equation is:

$$\begin{vmatrix}
k_{11} & -\frac{1}{\omega^2} & k_{12} & k_{13} \\
k_{21} & k_{22} - \frac{1}{\omega^2} & k_{23} \\
k_{31} & k_{32} & k_{33} - \frac{1}{\omega^2}
\end{vmatrix} = 0$$

After expansion, the cubic algebraic equations of \(\frac{1}{\omega^2}\) can be obtained:

$$\left(\frac{1}{\omega^2}\right)^3 - \left(\frac{k_{11}J_{wt} + k_{22}J_1 + k_{33}J_g}{\omega^2}\right)\left(\frac{1}{\omega^2}\right)^2 \cdots = 0$$

(13)

Assuming that the equation roots are respectively \(\frac{1}{\omega_1^2}, \frac{1}{\omega_2^2}\) and \(\frac{1}{\omega_3^2}\), then the above equation can be written as:

$$\left(\frac{1}{\omega^2} - \frac{1}{\omega_1^2}\right)\left(\frac{1}{\omega^2} - \frac{1}{\omega_2^2}\right)\left(\frac{1}{\omega^2} - \frac{1}{\omega_3^2}\right) = 0$$

After expansion:

$$\left(\frac{1}{\omega^2}\right)^3 - \left(\frac{1}{\omega_1^2} + \frac{1}{\omega_2^2} + \frac{1}{\omega_3^2}\right)\left(\frac{1}{\omega^2}\right)^2 \cdots = 0$$

(14)

To contrast the equation (13) and equation (14), then

$$\frac{1}{\omega_1^2} + \frac{1}{\omega_2^2} + \frac{1}{\omega_3^2} = k_{11}J_{wt} + k_{22}J_1 + k_{33}J_g$$

In the actual project, the value of high frequency is far greater than that of the low frequency, so \(\frac{1}{\omega_1^2}\) and \(\frac{1}{\omega_2^2}\) are neglected, and then the Dunkerley equation can be obtained:

$$\frac{1}{\omega_3^2} \approx k_{11}J_{wt} + k_{22}J_1 + k_{33}J_g$$

(15)

Assuming that \(\frac{1}{\omega_1^2} = k_{11}J_{wt}, \frac{1}{\omega_2^2} = k_{22}J_1, \frac{1}{\omega_3^2} = k_{33}J_g\), then the above equation can be expressed as:

$$\frac{1}{\omega_3^2} \approx \frac{1}{\omega_1^2} + \frac{1}{\omega_2^2} + \frac{1}{\omega_3^2}$$

(16)

The above equation shows that the fundamental frequency of the system of three mass blocks can be approximate to the natural frequency of the system when each concentrated mass block exists in isolation. That is to say, the Dunkerley method can be used to convert the solution of the natural frequency of the strong coupling system into the calculation of the natural frequency of each independent mass block. Of course, the equation (15) is obtained after neglecting the item of high frequency, so that the calculated natural frequency is slightly smaller than the actual value. Due to a big difference between the high frequency and fundamental frequency of the drive system in the wind turbine generator system, the results calculated by the Dunkerley equation are relatively accurate.

In the practical engineering application, the rotational inertia of the wind turbine \(J_{wt}\) is much larger than that of the gearbox \(J_1\). Therefore, the equation (16) can be further simplified. In order to grasp the essence of the research content, for the related issues of the main drive system, this paper converts the multi-degree-of-freedom system into the two-degree-of-freedom system constituted by the wind turbine and generator rotor as the analysis model.

3.1.3 Sensitivity of natural frequency to mechanical parameters

The process from calculation of the state matrix to the calculation of the eigenvalue indicates that each element of the state matrix has varying degrees of impact on the eigenvalue, while the impact of the structural parameters of each element on the eigenvalue is also different. Assuming that \(\psi_i\) and \(\phi_i\) are respectively the left eigenvector and right eigenvector related to the eigenvalue \(\lambda_i\). The following equation can be used to investigate the sensitivity of the eigenvalues to the mechanical parameters [170]:

$$\frac{\partial \lambda_i}{\partial \mu} = \frac{\partial \lambda_i}{\partial \mu} \frac{\partial \phi_i}{\partial \mu}$$

(17)
4 NUMERICAL CALCULATION AND SIMULATION OF NATURAL FREQUENCY

4.1 Numerical calculation

With the research object of 1.5MW of wind turbine generator drive system, the mechanical parameters of three mass blocks and two mass blocks are shown in Appendix A and Appendix B. The natural frequencies of the shaft system can be respectively calculated by the use of the small perturbation analysis method and Dunkerley method, and the calculation results are shown in Table 1. The natural frequency of the model of two mass blocks is 1.9725Hz, which is sole after calculation by the use of the small perturbation method, and corresponding to the torsional vibration frequency of the main shaft; the natural oscillation frequencies of the mode of three mass blocks are 1.8873Hz and 20.4367Hz, of which the former corresponds to the natural frequency of the torsional vibration of the main shaft, and the latter corresponds to the natural torsional frequency of the high-speed shaft. The Dunkerley method can only be used to calculate the fundamental frequency, with the result of 1.7993Hz and 1.7862Hz. Obviously, the natural frequency of the model of three mass blocks and two mass blocks is similar.

The sensitivity of rotational inertia, damping coefficient and torsional stiffness of three mass blocks and two mass blocks to the natural frequency of the main shaft can be calculated according to the equation (17), as shown in Table 2 to Table 4.

4.2 Simulation analysis

Taking the system shown in Figure 3 as an example, the double-fed wind turbine generator Gw is connected to an infinite busbar. The voltage of infinite busbar is $U_{\infty}$; the line inductive reactance is $x_e$, and the line resistance is neglected. Under the wind speed disturbance as shown in Figure 4, if the three-phase short-circuit fault occurs in the busbar B2 at $t=0.5s$, the duration is 0.5s. Figure 5(a) is the transient response curve of the revolving speed. The solid line and dashed line respectively correspond to three mass blocks and two mass blocks. Thus, the oscillation amplitude and frequency of these two modeling methods are similar. When the fault duration is increased to 0.8s, the response of the shaft system is completely different, as shown in Figure 5 (b). Fault causes the rotor to swing. The amplitude of the model of two mass blocks gradually reduces after a period of

Table 1. Eigenvalues calculated based on the small perturbation method and Dunkerley method.

<table>
<thead>
<tr>
<th>Small perturbation method</th>
<th>Dunkerley method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model of three mass blocks</td>
<td>Model of two mass blocks</td>
</tr>
<tr>
<td>$-4.289 \pm j1.8873$</td>
<td>$-4.633 \pm j1.9725$</td>
</tr>
<tr>
<td>$-7.759 \pm j20.4367$</td>
<td>$-25.006$</td>
</tr>
<tr>
<td>$-18.767$</td>
<td>$-41.377$</td>
</tr>
</tbody>
</table>

Table 2. Sensitivity of $f=1.8873$Hz to mechanical parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>$J_{\text{wt}}$</th>
<th>$J_1$</th>
<th>$J_g$</th>
<th>$k_{\text{swt,1}}$</th>
<th>$k_{\text{s1,}g}$</th>
<th>$D_{\text{wt}}$</th>
<th>$D_1$</th>
<th>$D_g$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensitivity</td>
<td>0.814</td>
<td>0.665</td>
<td>0.732</td>
<td>4.123</td>
<td>5.778</td>
<td>0.165</td>
<td>0.143</td>
<td>0.207</td>
</tr>
</tbody>
</table>

Table 3. Sensitivity of $f=20.4367$Hz to mechanical parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>$J_{\text{wt}}$</th>
<th>$J_1$</th>
<th>$J_g$</th>
<th>$k_{\text{swt,1}}$</th>
<th>$k_{\text{s1,}g}$</th>
<th>$D_{\text{wt}}$</th>
<th>$D_1$</th>
<th>$D_g$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensitivity</td>
<td>0.768</td>
<td>0.879</td>
<td>1.003</td>
<td>5.854</td>
<td>6.175</td>
<td>0.236</td>
<td>0.334</td>
<td>0.209</td>
</tr>
</tbody>
</table>

Table 4. Sensitivity of $f=1.9725$Hz to mechanical parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>$D_u$</th>
<th>$J_{\text{wt}}$</th>
<th>$J_g$</th>
<th>$K_{\text{gmr}}$</th>
<th>$k_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensitivity</td>
<td>0.276</td>
<td>1.884</td>
<td>2.067</td>
<td>2.281</td>
<td>5.348</td>
</tr>
</tbody>
</table>
oscillation, while the model of three mass blocks will eventually lose stability after increasing oscillation of the revolving speed during the period of fault. Figure 6 is the time-domain response of the torsion moment of two mass blocks and three mass blocks under the same fault condition, indicating that the frequency component of the torsion moment of the moment of three mass blocks is richer than that of the model of two mass blocks. The eigenvalue can be calculated by the prony method. The main vibration frequency of the model of two mass blocks is 7.137Hz, while the main vibration frequency of the model of three mass blocks is 1.8099Hz. Compared with the numerical calculation results of the natural frequency, the main vibration frequency of three mass blocks is closer to the natural frequency of the shaft system, thus causing serious damage to the shaft system due to resonance.

Figure 3. Simulation system of a single double-fed generator set switching in infinite busbar.

Figure 4. Wind speed disturbance model.

Figure 5. Rotor speed waveform of different shaft system models under the condition of power grid fault.

Figure 6. Rotor speed waveform of different shaft system models under the condition of power grid fault.

5 CONCLUSION

This paper calculates the eigenvalues of the torsion systems of two mass blocks and three mass blocks by the use of the small perturbation method and the Dunkerley method, in order to determine the natural oscillation frequency of the torsional vibration system. In the numerical calculation, the model of three mass blocks considers the flexibility of the gearbox and the model is closer to the physical model, so the natural oscillation frequency is smaller than that of the model of two mass blocks. The impact of the structural parameters on the natural oscillation frequency is analyzed through the calculation of the sensitivity. For the model of two mass blocks or three mass blocks, the impact of the torsional stiffness on the natural oscillation frequency is the most obvious, while the impact of the damping coefficient is the least. In the simulation analysis, when the wind turbine is switched in the infinite power grid, under the condition of small perturbation, the revolving speed response of the model of two mass blocks and three mass blocks is basically
the same. When the perturbation time is too long, the system of two mass blocks remains stable, while the revolving speed of the model of three mass blocks increases oscillation. According to the frequency distribution of the torque time-domain response of the model of three mass blocks, the generation of increasing oscillation is because the main vibration frequency is close to the natural vibration frequency, thus causing violent torsional vibration between the mass blocks.

6 ACKNOWLEDGMENT

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APPENDIX A

\[ J_{wi} = 9.5 \times 10^6 \text{ kg} \cdot \text{m}^2, J_i = 1123 \text{ kg} \cdot \text{m}^2, J_s = 70 \text{ kg} \cdot \text{m}^2, \]

\[ k_{\text{int}} = 2.3 \times 10^8 \text{ N} \cdot \text{m} / \text{rad}, k_i = 1.8 \times 10^8 \text{ N} \cdot \text{m} / \text{rad}, \]

\[ k_{\text{ge}} = 1.8 \times 10^8 \text{ N} \cdot \text{m} / \text{rad}, D_i = 812439 \text{ N} \cdot \text{m} \cdot \text{s} / \text{rad}, \]

\[ D_s = 73472 \text{ N} \cdot \text{m} \cdot \text{s} / \text{rad}. \]

APPENDIX B

\[ J_{wi} = 73472 \text{ N} \cdot \text{m} \cdot \text{s} / \text{rad}, J_i = 70 \text{ kg} \cdot \text{m}^2, \]

\[ k_{\text{ge}} = 8.52 \times 10^7 \text{ N} \cdot \text{m} / \text{rad}, D_i = 73472 \text{ N} \cdot \text{m} \cdot \text{s} / \text{rad}. \]

REFERENCES


