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Multi-body Dynamic Analysis of the Braking Process for Megawatt Wind Turbine Disc Brake

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Multi-body Dynamic Analysis of the Braking Process for Megawatt Wind Turbine Disc Brake

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Abstract. The dynamic characteristics of brake during the braking process are studied by the method of multi-body dynamics, in order to reduce the vibration of disc brake in the emergency braking process, improving the braking efficiency of the brake of the megawatt wind turbine. The flexible multi-body dynamics model of the mechanical assembly for the whole megawatt wind turbine disc brake is built, including the brake spring, active brake caliper, passive brake caliper and compensatory mechanism. The simulation results under the common working conditions are compared with the test results under the same working condition to verify the model. The influence trends of braking parameters, such as initial braking speed, spring braking force and friction coefficient between the brake pairs, on the dynamic characteristics of the brake are analyzed, which provides the theoretical basis for reducing the severe vibration during the emergency braking process in this paper.

1. Introduction

With the development of wind power industry, megawatt wind turbine has gradually become the key of wind turbine research in the world[1]. Its braking system is of vital importance in the efficient operation of wind power generation[2].The problem of vibration and other unstable braking is easy to be found in the urgent braking process of brake which is the main device in the brake system, because speed of main shaft is high and rotary inertia is large in the megawatt wind turbine, this brings the hidden danger to the normal operation of the wind turbine[3].As an important part of the wind turbine braking system, wind turbine brake's performance directly affects the safe operation of wind turbine.

It is necessary to do the further study on the dynamic characteristics of the wind turbine disc brake, in order to solve the problem of vibration and unstable braking during the braking process of the megawatt wind turbine disc brake. In the study of dynamic characteristics of disc brake, scholars in the world have made comparatively deep analysis using multi-body dynamics method. A multi-body dynamic model for vehicle brake was established by Kim et al, and the change of braking torque was measured by changing the thickness of the brake disc. It was found that the change amplitude of braking torque can be reduced by reducing the contact stiffness of the brake pair and increasing the torsional rigidity[4]. The parameters of the brake members were redesigned and the shape of the brake pad was optimized to make the force of brake pad uniform by Swift et al[5], which can be used to study the influence of contact area between the brake pairs on vibration of brake during the braking process . The change amplitude of brake torque was studied by using the model of disc brake with assembly error, and then the whole model of the brake was established to obtain the transmission way on vibration of brake and get the specific method of reducing the vibration by Gao et al.[6]. The vibration of the brake can be reduced by reducing the damping of the friction plate for the fixed caliper disc brake, in the study of the design of car brake by Ning Xiaobin and Zhang Wenming[7].

In this paper, the influence of the interaction among the brake components is taken into account in the braking process, the multi-body dynamic model of mechanical assembly for the whole megawatt wind turbine disc brake, including brake spring, active brake caliper, passive brake caliper and compensatory mechanism, is established, and the rationality of the model is verified. The influence trends of braking parameters on the dynamic characteristics of the brake are analyzed, which provides the theoretical basis for efficient braking of megawatt wind turbine disc brake.

2. Establishment of the multi-body dynamic model of the whole wind turbine disc brake

2.1. Establishment of the three-dimensional model for brake

Some components are included in the three-dimensional model of mechanical assembly for megawatt wind turbine disc brake, such as brake disc, up-pad and down-pad, compensation mechanism(includes compensation nut, compensation putter, compensation nut sleeve and piston), active brake caliper, passive brake caliper, end cover and brake spring. The outer diameter of the brake disc is 800mm, the inner diameter is 200mm, and the thickness is 40mm. The structural parameters of up-pad and down-pad are the same. The length of brake pad is 270mm and the width is 130mm. The size of guide grooves on both sides of the brake pad is 60mm×10mm. The material properties of the brake disc and brake pad are shown in Table 1[8]. The material of other components is Q345-b. Three-dimensional assembly model of mechanical assembly for wind turbine brake is shown in Figure. 1.

Table 1 Material attributes of brake disc and pads at room temperature[8]

brake element	material	density /(kg/m^3)	modulus of elasticity /MPa	Poisson ratio
brake pad	Copper based powder metallurgy	5250	1.80×10^5	0.30
brake disc	high thermal conductivity cast iron	7250	1.38×10^5	0.28

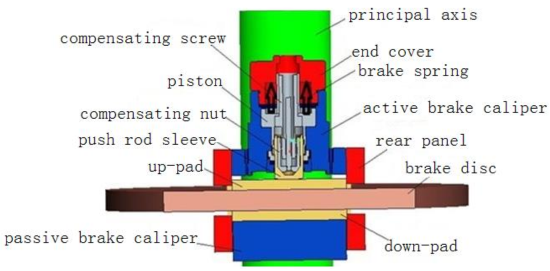
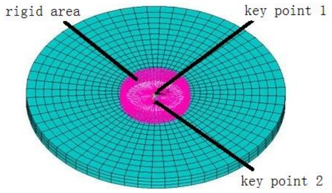


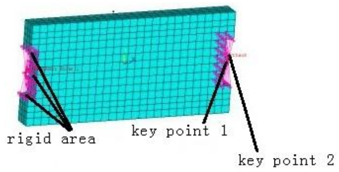
Figure 1. 3D solid model of mechanical assembly for wind turbine disc brake

2.2. Establishment of flexible body for the main components of brake

Vibration of brake disc is most obvious during braking process. Therefore, the brake disc is the most important represented component of braking vibration. Besides, the braking performance of the brake can be affected seriously by the contact situation between brake pairs of brake disc and brake pads. Brake disc and brake pads are the important components of brake which need to be flexible. The most critical step is to establish a rigid area for the components in the process. The flexible bodies of brake discs and pads, key points of connection with rigid bodies and rigid areas of components are shown in Figure. 2.



(a) flexible body of brake disc



(b) flexible body of brake pads

Figure 2 Flexible bodies of main parts for brake

2.3. Establishment of the dynamic model for brake

Three-dimensional model of mechanical assembly for brake is imported into the multi-body dynamics software and the flexible bodies of main components are replaced. According to the braking principle of the brake, constraint relationships are set up. And the constraint relationship for dynamics model of wind turbine disc brake are shown in Table 2.

Table 2 Joints of dynamic model for wind turbine disc brake

constraint link	constraint type	constraint link	constraint type
up-pad /down-pad and ground	translation	up-pad/down-pad and brake disc	contact
end cover and piston	spring	Brake disc and ground	revolute
Push rod sleeve and up-pad	contact	Passive brake caliper and down-pad	contact

The setting of “contact” in the constraint relationship is particularly important, and some settings are included in the contact attribute, such as stiffness coefficient, force index, damping coefficient and penetration depth, which are given by the materials of contact components and contact conditions. Friction between brake pairs is calculated by coulomb force. The dynamic or static friction coefficient, and the dynamic or static slip speed must be selected based on the materials of brake pairs and the actual operating conditions. The multi-body dynamics model of mechanical assembly for megawatt wind turbine disc brake is shown in Figure 3. The up-pad, down-pad and the brake disc are flexible bodies, but the others are rigid bodies in this model.

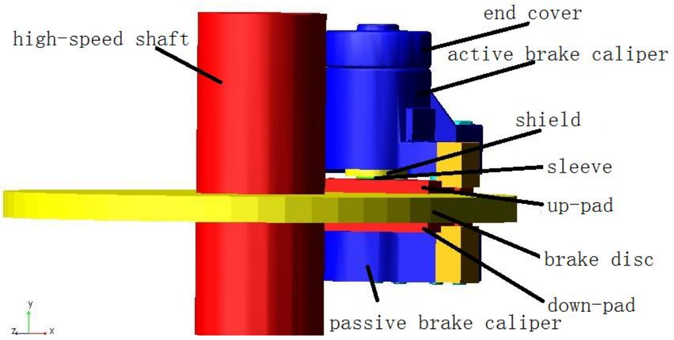
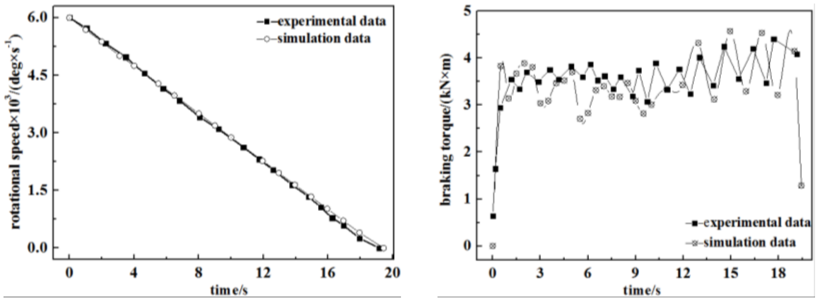


Figure 3 Flexible multi-body dynamics model of mechanical assembly for megawatt wind turbine disc brake

In Figure 3, the Y axis is the axial direction of the brake. The positive direction of Y is positive. It is found that the axial vibration of the brake is the most obvious during the braking process by simulation tests. Therefore, it is necessary to analyze the axial dynamic characteristics of the brake. Unless special instructions are given, the dynamic characteristics of the brake studied is about axial direction during the braking process in this paper.

3. Verification for the model of brake

The initial braking rotational speed of the brake disc is 1000r/min, the spring braking force is 17000N and the friction coefficient between the brake pairs is 0.3, under the common working condition. The simulation results obtained by the multi-body dynamics model of brake are compared with the test results obtained by test-bed to verify the rationality of the multi-body dynamics model of mechanical assembly for megawatt wind turbine disc brake. The working condition of the inertial test rig for wind turbine disc brake is that the environment temperature is 20℃, humidity is 30% and the moment of inertia is 600 kg/m^2. Through the acquisition device of test rig, such as sensor and amplifier device, the test curve of rotational speed-time and braking torque-time are reported in braking process, and the test results and simulation results are compared with each other. It is necessary to convert the unit of the rotational speed, since the unit of rotational speed in ADAMS is deg*s-1. The comparison curves between the simulation and test of braking results are shown in Figure 4 under the common working conditions.



a) Curve of comparison for rotational speed-time b) Curve of comparison for braking-time

Figure 4 Curve of comparison between simulation and test under common condition

It can be seen from Figure 4(a) that the simulation value of braking time is 19.44s, the test value is 19.16s and the error rate is 1.5% in this working condition. Besides, the slope of the simulation and test curve is basically consistent and the braking is completed by constant deceleration. The average value of braking torque is about 3468 N·m, which is found in the curve of comparison between simulation and test for the braking torque-time in the braking process under the common working condition, and the curve are fluctuated in the form of sine curve in the Figure 4(b). Simulation and test curve get risen at the end of braking, because the friction between brake pairs is turned into the static friction from dynamic friction, and the static friction is always higher than dynamic friction at that time. According to the comparison results, the simulation results corresponded well with test results under the same working condition.

It can be shown by comprehensive analysis that the simulation results of the multi-body dynamics model are in conformity with the actual working condition, so the model is reasonable and can be used for the subsequent simulation research for dynamic characteristic in the braking process.

4. Analysis for dynamic characteristics of brake during the braking process

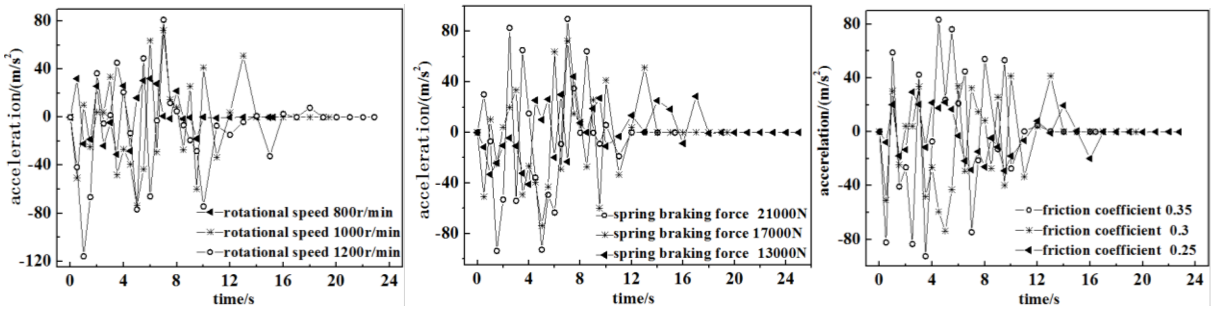
4.1. The influences of braking parameters on dynamic characteristics of brake caliper

The multi-body dynamics model is operated to study the influences on dynamic characteristics of braking parameters which are changed by equal gradient based on common braking condition. The simulation condition of different braking parameters are shown in Table 3.

Table 3 Simulation condition of different braking parameters

Simulation condition	initial braking rotational speed/(r/min)	friction coefficient between the brake pairs	spring braking force/N
1	800, 1000, 1200	0.3	17000
2	1000	0.3	13000, 17000, 21000
3	1000	0.25, 0.3, 0.35	17000

The influence trends of braking parameters on the dynamic characteristics of passive brake caliper are studied mainly, because the vibration of passive brake caliper is more severe than that of active brake caliper, according to the braking principle. Three kinds of working condition, as shown in Table 3, are respectively set in the simulations, such as initial braking rotational speed, spring braking force, friction coefficient between the brake pairs. The dynamic characteristic of passive brake caliper are shown in the Figure 5, under the conditions of different braking parameters.



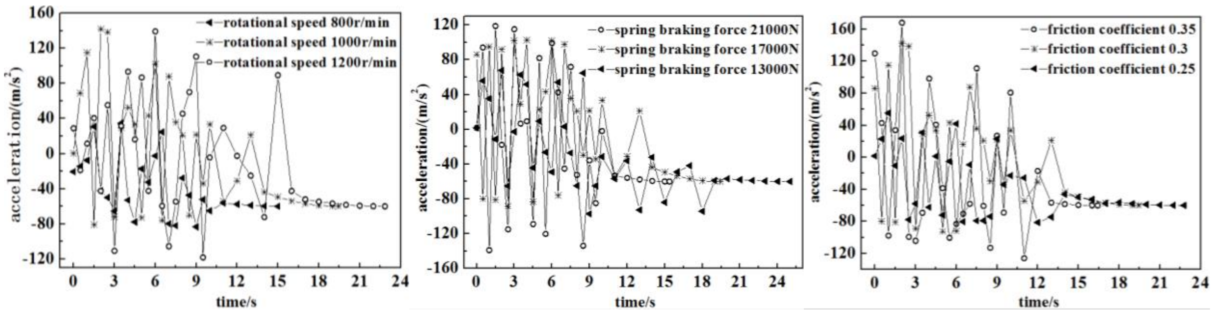
(a) (b) (c)

Figure 5 Dynamic characteristics of passive brake caliper under different braking parameters

It can be seen from Figure 5(a) that the higher the value of the initial braking rotational speed is, the longer the braking time is, and the larger the fluctuation for the axial acceleration of passive brake caliper is. The reason is that the higher initial braking rotational speed is, the larger the energy of the brake disc is, under the same external load. So it takes longer time to achieve braking. Besides, the larger the energy is, the severer the vibration of the brake is during the braking process. It can also be seen from Figure 5(b) that the larger the spring braking force is, the shorter the braking time of the brake is, and the larger the fluctuation for the axial acceleration of passive brake caliper is. The reason is that the larger the spring braking force is, the larger the external load is, and the braking time is obviously shorten. But the larger the spring braking force, the larger the deformation of the spring is, and the larger the instability of the spring braking force, and even the larger the impact to the brake disc is, which causes the severer vibration of the brake. Besides, the larger friction coefficient between the brake pairs is, the shorter the braking time of the brake is, and the larger the fluctuation for the axial acceleration of passive brake caliper is. The reason is that the larger the friction coefficient between the brake pairs is, the larger the friction resistance is, this shortens the braking time. But the larger friction resistance between the brake pairs is, the larger friction resistance will be resisted by the brake disc with the severer vibration, causing severer vibration of the brake, as shown in Figure 5(c).

4.2. Influence of brake parameters on the dynamic characteristics of brake disc

The dynamic characteristic of brake disc are shown in Figure 6, under the conditions of different braking parameters.



(a) (b) (c)

Figure 6 Dynamic characteristics of brake disc under the different braking parameters

It can be seen from Figure 6 that the larger the braking parameters are, such as initial braking rotational speed, the spring braking force and friction coefficient between the braking pairs, the larger the fluctuation for the axial acceleration of the brake disc is. Besides, the fluctuation degree for the axial acceleration of the brake disc in Figure 6 is larger than that of the passive brake caliper in Figure 5 under the same condition. Brake disc is the terminal component of the brake to achieve braking and the brake disc is equipped with a certain amount of energy at the early stage of braking owing to the higher initial braking rotational speed of the brake disc. But the passive brake caliper is stationary at the early stage of braking, and the passive brake caliper just is the transitive part of energy during the braking process. Therefore, the vibrations of the brake disc are severer than the passive brake caliper under the same working conditions.

5. Conclusions

Based on the theory of dynamics, the multi-body dynamic model of the mechanical assembly for megawatt wind turbine disc brake is established, and the rationality of the model is verified by simulation and test comparison curves of rotational speed-time and braking torque-time in braking process under common working conditions. The influence trends of braking parameters on the dynamic characteristics of the main brake members are studied during the braking process. It is concluded that with the increase of the initial braking rotational speed, the spring braking force and the friction coefficient between the brake pairs, the fluctuation tends for the axial acceleration of passive brake caliper and brake disc is increased roughly, indicating the vibration degree of the brake is also more severe. Besides, the vibration of the brake disc is severer than that of the passive brake caliper, indicating that the vibration of the brake can be reflected well by that of the brake disc during the braking process.

Acknowledgments

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