IOP Conference Series: Materials Science and Engineering

PAPER • OPEN ACCESS

Friction-thermal-vibration Coupling Analysis of Large-megawatt Wind Turbine Brake

Friction-thermal-vibration Coupling Analysis of Large-megawatt Wind Turbine Brake

S Zhang, P Xin, J Yin, Y Liu*, Z Sha, F Ma and D Yang

School of Mechanical Engineering, Dalian Jiaotong University, Dalian, China

liuyu_ly12@126.com

Abstract. To establish a mutual contact relationship between the brake disc and the brake pad, simulation of the brake disc in the actual working conditions braking deceleration, according to the wind of the brake and brake piece of actual size and related parameters of brake disc material in ABAQUS to establish 3D transient friction-thermal-vibration coupling finite element model. The brake process is verified with the analysis of brake and brake disc contact, from the coupling results, this paper summarizes the temperature and stress distribution, the research on the temperature in time and space distribution. In this paper, the stress of brake stress is analyzed, and the distribution of stress is summarized, and the importance of the coupling study on the stress analysis of brake is analyzed.

1. Introduction

The braking process of the large- megawatt wind turbine brake under the actual working condition is very complicated. Friction-thermal-vibration coupling will have a great influence on the braking performance. Study on friction-thermal-vibration coupling is of great significance on the design and optimization of the wind turbine brake. In recent years, many studies were carried out on the field of thermal-structural coupling analysis and vibration analysis respectively. The heat conduction equation and thermal stress equation are used to describe the thermal structure coupling process by mathematical analytic method. Zagrodzki[1] et al established the 2D symmetric model of the brake and analyzed the thermo-elastic problem. The finite element method was used to describe the temperature and the thermal stress. Lu[2] et al took the train disc brake as the research object, established a 3D transient thermal structural coupling finite element analysis model and simulate the actual braking conditions. The results showed that the uneven distribution of temperature and equivalent stress on the contact surface leads to the reducing of actual contact area. Ghosh[3] through the analysis of the vibration of the brake disc with variable thickness, it is found that the temperature caused by the friction in the braking process has a certain influence on the vibration. The finite element model of automobile disc brake is set up by Dai[4] showed the influence of the coupling state of temperature field and stress field on the vibration of brake. In this paper, considering friction-thermal-vibration coupling, the vibration coupling dynamic force is put forward, and the friction-thermal-vibration coupling model of the braking process is established with ABAQUS software.

2. Vibration analysis in large-megawatt wind turbine brake

2.1. Vibration analysis of brake pressure during the braking process

Vibration can be regarded as uniform force when the disc brake is braking urgently. Assuming that the braking pressure of the unit area on the upper surface of the brake pad is p, the compression force F generated by each element is:

$$\mathrm{d}F = kpr\mathrm{d}\theta\mathrm{d}r \tag{1}$$

where k is the effective contact area coefficient in the brake disc/pad, r is the radius of the brake disc, θ is the angle of the radial boundary of the brake pad.

According to the knowledge of thermodynamics, the friction heat generated during braking will cause thermal expansion of brake pad, and its expansion thickness in axial direction $\Delta\delta_s(r,t)$ is:

$$\Delta\delta_s(r,t) = \frac{\frac{\pi}{6}a_s\delta_s\mu kp\left(r^3 - r_0^3\right)\left[\theta(t) - \theta(t_0)\right]}{C_a\gamma_a\left(\delta_s + \delta_f\right)} \tag{2}$$

where a_s and a_f represent the thermal expansion coefficient in the brake disc/pad respectively. The deformation of the brake pad in the axial direction is a linear function relationship with the three square of the radius of the brake. With the increase of the radius, the deformation of the material will also increase.

2.2. Effect of deformation on brake pressure

The initial state of the brake disc is uniform in shape, and in the working process it will become thin in the outer thickness. The distribution of the pressure on the brake sluice is also small, and the relationship between the braking force and the radius on the brake sluice is obtained by converting the type (2) to the brake disc.

$$p\left(r^3\right)r^3 = \lambda\Delta\delta(r,t) + p\left(r_0^3\right)r_0^3 \tag{3}$$

Considering the integer variable is not very large, the expansion thickness is assumed to be a fixed value. Therefore, a new expression of pressure distribution on the brake sluice is proposed and defined as a vibration coupling dynamic force.

$$p\left(r^3\right)r^3 = C \tag{4}$$

That is to say, the three power product of the brake pressure and the brake disc radius is the fixed value C, so that the pressure distribution on the brake pad has certain regularity.

The vibration and noise are caused by the heat deformation of the material, which leads to the change of the braking force during the whole braking process. The expression of the vibration coupling force can better express the distribution of brake pressure on the brake Sluice at every moment. Combined with the temperature field and stress field, the actual braking condition can be simulated more effectively, and the analysis results are closer to the actual working conditions.

3. Establishment of friction thermal vibration coupling model

3.1. Assumptions

During the braking process, a large amount of heat is generated by the friction in brake disc/pad. The heat is distributed in a gradient on the brake disc. The heat is transmitted to the interior of the disc and in the surrounding air. The main heat transfer mode is heat conduction and heat convection. As the temperature of the braking process is very high, there will be a small amount of heat radiation. When models of the brake disc/pad are established, the following assumptions are made, which are material of the brake disc/pad are isotropic, and parameters of materials do not change with temperature variation. The friction coefficient μ in the brake disc/pad is constant during the braking process. Wear of the materials during the braking process is ignored, there is only elastic deformation. During the braking process, all of the friction work is converted to heat, and the heat dissipation of the brake disc is non-uniform.

3.2. Determination of related parameters

According to a certain type of large-megawatt wind turbine brake, material of the brake pad is copper-based powder metallurgy, and the brake disc's is high thermal conductivity cast iron. The relevant parameters of the brake disc/pad under different temperature are shown in Table 1 and Table 2:

Table 1. Material parameters of copper-based powder metallurgy.[5]

Temperature ℃	Thermal conductivity w/(kg·℃)	Specific heat capacity J/(kg·℃)	Modulus of elasticity MPa	Linear expansion coefficient /℃
100	380	436	1.72×10^{5}	1.11×10^{-5}
200	364	436	1.57×10^{5}	1.12×10^{-5}
300	352	436	1.41×10^{5}	1.12×10^{-5}

Table 2. Material parameters of highly conductive cast iron.[6]

Temperature ℃	Thermal conductivity w/(kg ℃)	Specific heat capacity J/(kg ℃)	Modulus of elasticity MPa	Linear expansion coefficient /℃
100	57	497	0.98×10^{5}	1.12×10^{-5}
200	74	525	0.88×10^{5}	1.20×10^{-5}
300	74	574	0.84×10^{5}	1.23×10^{-5}
400	74	634	0.71×10^{5}	1.27×10^{-5}
500	74	688	0.67×10^{5}	1.30×10^{-5}

3.3. Determination of vibration boundary conditions

When the friction-thermal-vibration coupling model is established, the influence of the spindle vibration on brake pressure should be fully considered. The braking pressure on the brake pad is applied dynamically when the model is established. In this paper, the vibration boundary of the brake pad is based on the data of the whole acceleration a of the brake pad through dynamic simulation with ADAMS software, which is transformed by the Newton's second law. The initial brake pressure F is 17000N, the initial spindle speed ω is 1000rmp and the friction coefficient is 0.3, and the acceleration-time variation curve of the brake pad is shown in Figure 1, in which the positive and negative scale of the ordinate represent the direction of acceleration (up is positive and down is negative). Combined with the proposed brake vibration coupling dynamic equation, the braking pressure at each moment is the force at the center radius of the brake pad, and the brake pressure on the brake pad is rearranged at every moment. Same method is used with different working conditions. The friction-thermal-vibration FE analysis model is presented in Figure 2.

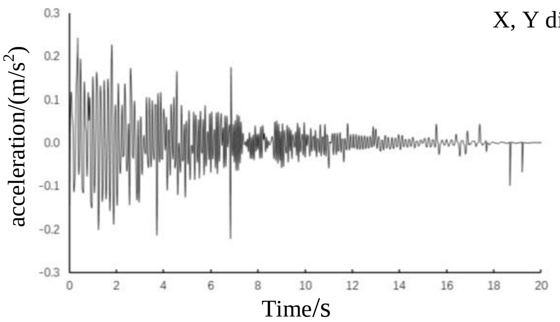

Figure 1. Acceleration-time variation curve of the brake pad.

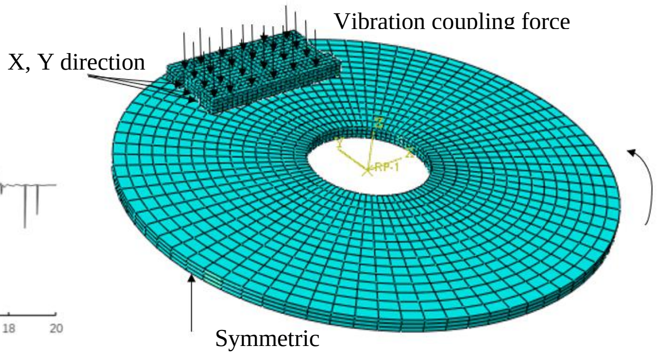

Figure 2. Friction-thermal-vibration coupling finite element model of the disc brake.

4. Simulation results and analysis of brake disc temperature and stress

The distribution contour of the transient temperature of the brake disc when the initial brake pressure is 17000N, the initial brake disc speed is 1000rpm, and the friction coefficient is 0.3 is shown in Figure 3. As can be seen, the high temperature is concentrated in the friction area, and the temperature rise in the non-friction area is not obvious. The maximum temperature is related to the position of the contact area. The initial contact temperature was the highest and then concentrated in the central annular area where the radius was 300mm. The temperature rise near the friction area spreads to both sides.This is because at the initial stage of braking, frictional heat flux input dominates the distribution of the temperature field. Because the brake is a three-dimensional model, as the braking process progresses, due to the thermal deformation of the friction material, the heat flow input to the brake disc is not axis symmetric. As the brake disc speed decreases, the heat input intensity gradually decreases, and the effect on the distribution of the temperature field gradually decreases.

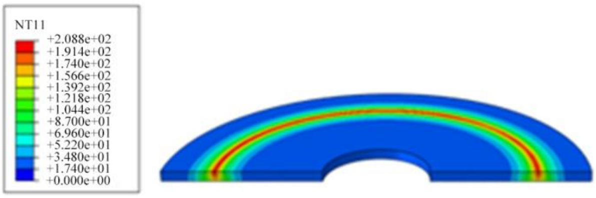

Figure 3. Brake disc temperature distribution contour.

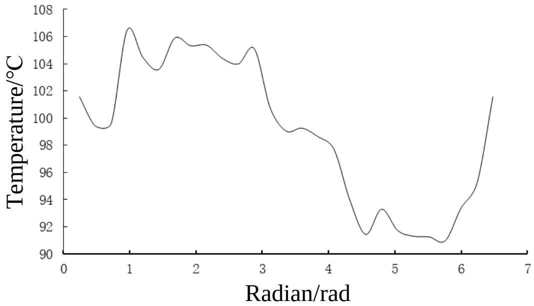

Figure 4. Temperature variation curve of the direction of rotation of the brake disc (0.50s).

To quantitatively describe the circumferential distribution of the temperature shown in Figure 3, 26 equally spaced nodes at the disc surface contact radius R_1=300mm in the counterclockwise direction are chosen to draw the temperature-arc curve in Figure 4. The starting point on the curve is the node at the corresponding radius in the positive direction of the Y axis. The ending point on the curve is the node closest to the Y axis on the right side of the Y axis. The temperature difference at the time of 0.50s is not obvious, and the temperature fluctuates counterclockwise from the left end of the contact with the brake pad until it contacts the right end of the brake pad again.

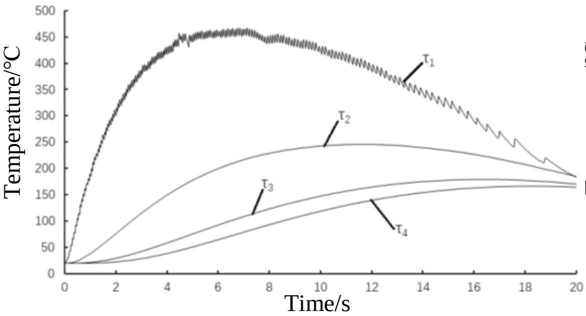

Figure 5. Nodes temperature of the brake disc in the axis-direction.

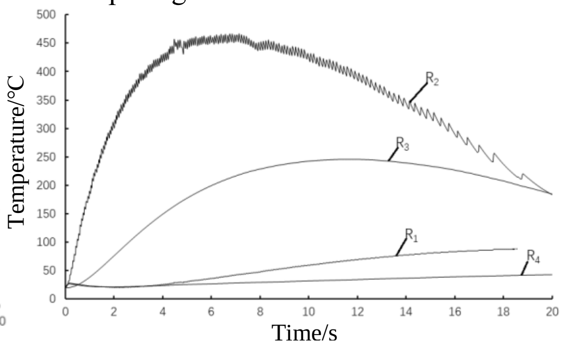

Figure 6. Nodes temperature of the brake disc in the radial-direction.

Figure 5 is the curve of the temperature of nodes along the Z axis selected at the radius of the disc R_1=300mm along the Z axis at τ_1=0, τ_2=6mm, τ_3=12mm, τ_4=20mm and plot the temperature of the node over time. The temperature rise of each node is different in Figure 5. The temperature rise at the contact surface of the brake disc is relatively rapid and much higher than that of the internal node of the brake disc. It can be found that only the temperature rise of the contact region node fluctuates. In the axial direction, the nearer the contact surface is, the more obvious the temperature rise during the braking process and the gradual transfer of frictional heat flow to the inside are. The temperature rise of the inner node of the brake disc far from the contact surface is obviously delayed compared with that of the node near the brake disc contact area. This is because the internal node heat flow input along the Z-axis direction except the nodes in contact area are mainly derived from the thermal conduction of the upper nodes. However, the heat transfer distance is proportional to the time, and the upper layer needs a certain amount of time to accumulate heat flow to the internal materials. Node τ_2 will output to the lower layer in addition to the input of the upper layer heat flow during the braking process. In the first 12 s, the heat flow input of Node τ_2 is larger than the heat flow output. The node temperature rises and reaches the maximum temperature in the 12th second. After that, the heat flow output is larger than the heat flow input, and the node temperature begins to decrease.

Select nodes with the radius R_1=275mm, R_2=300mm, R_3=325mm, and R_4=350mm in the positive direction of the Y-axis of the brake disk contact surface. Extract the temperature value of each node at each moment and draw the curve of the node temperature as shown in Figure 6. It can be seen that there is a temperature gradient with a large temperature difference in the radial direction of the brake disc during the braking process. The temperature changes at R_2 and R_3 in a parabolic shape. The temperature at R_2 varies in the form of fluctuations during the braking process, and the decreasing trend is more slowly than the rising trend. However, it can be seen that the temperature at R_2 is higher than that at R_3 at each moment. The temperature rise at R_1 and R_4 are relatively gentle, and the temperature of R_1 and R_4 in the first 4s are basically the same, but the temperature at R_1 after 4s is always higher than that at R_4. The brake disc contact surface undergoes a complicated heat exchange process, and the three heat exchange strengths occurring at every moment and every position are different. The temperature rise R_2 and R_3 is not only due to the frictional heat flow contact between the brake disc and the brake pad, but also the input of friction heat flow in the vicinity of the high temperature conduction form, which causes the obviously temperature rise. A temperature gradient with a large temperature difference is formed on the contact surface of the brake disc. After the above-mentioned nodes separating from the brake pad, the heat exchange form becomes heat dissipation into the air. Due to the deceleration, the input of heat flow generated by friction is reduced in each circle, but the output gradually increases, the input of heat flow in the early stage is larger than the output, the temperature of the node rises and reaches the maximum value, and the output of heat flow in the later stage is greater than Input, the node temperature decreases.

5. Effect of brake pressure on transient coupling of the brake disc

Select the node whose radius of the contact surface of the disc is R=300mm in the positive direction of the Y-axis as the research object. Extract the temperature at different times during the braking process and draw a time-varying temperature diagram as shown in Figure 7. It can be seen that under the three conditions, as the brake pressure increases, the brake disk temperature change rate also increases. In the early stage of braking, the stress fluctuation amplitude is small, and then the change amplitude gradually increases. The larger the initial brake pressure, the greater the stress change. This situation will aggravate the thermal instability of the brake disc and reduce the service life of the brake disc.

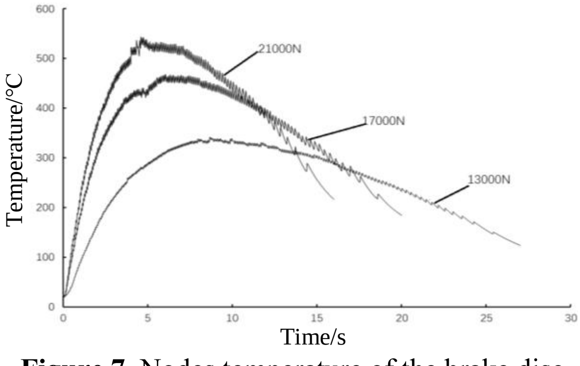

Figure 7. Nodes temperature of the brake disc at different initial pressure(R=300mm).

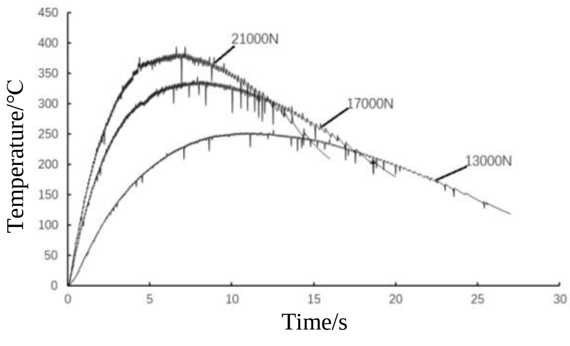

Figure 8. Nodes von mises stress of the brake disc at different initial pressure(R=300mm).

This is mainly because the relative speed of the brake is relatively high and the change frequency of stress is high, so the initial equivalent stress increases rapidly, but with the braking process, the speed of the brake disc gradually decreases and the equivalent stress change frequency of the contact area of the brake disc becomes smaller, so the stress variation characteristics of the large value fluctuation shown in the diagram show. In addition, the greater the stress of the larger braking force will increase the amplitude of the equivalent stress fluctuation. The equivalent stress value of the node in the positive direction of the Y axis at the radius of R=300mm under the three working conditions is drawn, and the equivalent stress time variation of the node under the action of different initial brake pressure shown in Figure 8 is drawn. It can be seen from the image that the equivalent stress time variant curve is lagging behind the corresponding temperature time curve.

6. Conclusion

Based on the heat and stress analysis of the megawatt wind electric brake, the coupling problem of the sliding temperature field, the stress field and the vibration between the brake disc and the brake disc is fully considered. The finite element model of the friction thermal vibration coupling analysis is established, and the braking deceleration process is simulated, and the disc brake is obtained by using the non linear multi physical field method. The transient temperature field and the stress field. Through the analysis of the results of the analysis, the following conclusions are drawn. The temperature value in the circumferential direction of the brake disc is decreasing from the contact position of the brake disc and brake disc. The temperature field of the friction region of the brake disc is circular distribution and produces a clear temperature gradient in the radial and axial direction. The temperature rise near the contact surface of the brake disc is far away from the temperature at the contact surface. The deeper the depth is, the more obvious the hysteresis characteristic of temperature is rise. Increasing the initial brake pressure will aggravate the uneven distribution of the temperature and stress field of the brake disc, and with the increase of the initial brake pressure, the maximum of the temperature and stress will increase and the change rate increases obviously in the process of braking, so the use of brake pressure can be reduced to increase the use of the brake. In addition, by comparing the equivalent stress time-varying curve and temperature curve, we can see that the equivalent stress change has obvious hysteresis compared with the temperature.

Acknowledgement

The authors gratefully acknowledge the financial support of National Natural Science Foundation of China (No. 51475066 and No. 51675075), Natural Science Foundation of Liaoning Province (No. 2015020114).

References

[1] ZAGRODZKI P.Analysis of thermomechanical phenomena in multidisc clutches and brakes[J].*Wear*,1990,140:291-308.
[2] Lu Chao, Gao Chenghui, Huang Jianmeng. Study on contact pressure distribution of train brake pad contact interface [J]. *Mechanical design and manufacturing engineering*, 2013, 7:47-51. (In Chinese)
[3] Ghosh N C.Thermal effect on the transverse vibration of spinning disk of variable thickness[J].*Journal of Applied Mechanics*,1975,42(2):358.
[4] Dai changju. Analysis of vibration and noise generation mechanism and transfer path of automotive disc brake [D]. *Shenyang: Northeastern University*, 2014. (In Chinese)
[5] Liu Jianxiu. Research and development of copper based powder metallurgy friction materials and their high temperature fatigue and impact properties [D]. *Henan: China Institute of engineering and physics*, 2003-12-10. (In Chinese)
[6] Liu Li. High thermal conductivity cast iron for brake discs [J]. *World automobile*, 1992 (5):62-67. (In Chinese)