Design and performance analysis of pneumatic disc brake of a new energy commercial vehicle

Shichao Fu$^1$ and Haiming Sun

Hubei University of Automotive Technology, Shiyan 442002, China

$^1$E-mail: 201911053@huat.edu.cn

Abstract. The brake is the key mechanism of automobile braking system, which makes the vehicle reach the expected driving state by outputting braking force to the wheel. SolidWorks was used to model the pneumatic disc brake of EQ1180GEVJ new energy commercial vehicle. The finite element software ANSYS workbench was used to conduct the finite element analysis of the brake disc, brake clamp and bracket, and the stress cloud diagram under the braking condition, as well as the first 10 natural frequencies and the main vibration pattern diagram were obtained. The results show that the maximum deformation of the brake disc is in the lower position on both sides of the brake disc, and the maximum stress is in the position connected with the wheel hub. The maximum stress of the brake clamp is at the connection between the connecting arm and the brake chamber. The maximum stress of the support is at the connection between the support and the suspension. The first ten natural frequencies of the three parts are between 240Hz and 3000Hz. By analysing the mode, we can obtain the position where the mode which is prone to sudden vibration frequency change and the system may generate the maximum vibration amplitude when resonance occurs. The results provide some references for further optimization of disc brake.

1. Introduction

There are mainly two types of brake: drum brake and disc brake. The brake system of traditional truck adopts full drum brake because of the low cost and simple structure of drum brake [1]. With the development of the automobile industry, the advantages of disc brakes are becoming more and more obvious. Compared with the drum type, disc brakes have better wading ability, brake stability and heat dissipation. The development trend of the braking system is to match the disc brake for the truck. The general administration of quality supervision, inspection and quarantine has made mandatory requirements in GB7258-2017 "Technical conditions for motor vehicle operation safety" [2].

For the research of the brakes, Zheng Tao, Chinese, adopted the method of extracting the amplitude of the complex mode vibration, summarized the main vibration parts of each order noise mode of the disc brake, and selected the parts with the largest overall amplitude as the optimization focus [3]. The research on brake squealing involves tribology, nonlinear dynamics, contacting mechanics and instability theory [4, 5]. Li Ming-Lei used ABAQUS to extract the support and reaction forces of the shell to obtain the load conditions for topology optimization, and determined the necessary design variables, responses, constraint conditions and objective function parameters for topology optimization and used Optistruct for topology optimization, then the calculation results were obtained [6]. Other researchers have studied the thermal field and stress field during braking, as well as the change of friction coefficient caused by heat, etc. [7-10] through the analysis of the brake's heat-structure coupling.
But there is less performant analysis of pneumatic disc brake for new energy commercial vehicle. This paper takes the pneumatic disc brake of a truck as the research object, designs its size, models it in Solidworks, and uses ANSYS Workbench to analyze the stress and prestress modes of the main parts of the brake under braking conditions.

2. Calculation of braking system
The main functions of the automobile brake system are: to make the vehicle to reduce the speed of the appropriate speed down to stop; When driving downhill, keep the car at a proper and steady speed. Make the car stop reliably in place or on a ramp. The design of the braking system should have sufficient braking capacity, good reliability, short lag time, good braking thermal stability, good braking water stability. Table 1 shows the vehicle parameters.

There are three basic working conditions when vehicle is braking. (1) The front wheel is locked and de-skidded, then the rear wheel is locked and de-skidded. (2) The rear wheel should be locked and de-skidded, then the front wheel should be locked and de-skidded. (3) The front and rear wheels are locked and de-skidded at the same time [11]. In these three conditions, the third condition is the best.

| Attribute        | Parameter      |
|------------------|----------------|
| total mass(kg)   | 9295           |
| curb weight(kg)  | 3350           |
| tyre size        | 9.00-20-12PR   |
| wheel base(mm)   | 3950           |
| $h_g$(mm)        | 1025           |
| $a$(mm)          | 1894           |
| $b$(mm)          | 2056           |

2.1. Calculation of normal reaction force
For the calculation of the normal reaction force of the ground facing the front and rear wheels, the rolling resistance distance, air resistance distance and the inertial force distance of the rotating parts are ignored. According to Figure 1, the calculation is as follows:

$$F_z L = G b + m \frac{d}{dt} \frac{h_g}{h_s} \tag{1}$$

$F_z$ is the normal reaction force of the ground facing the front wheel and the coefficient of synchronous adhesion $\varphi_0 = 0.5$, order $\frac{du}{dt} = z g$, $Z$ is the braking strength, and the ground normal reaction force can be obtained as:

$$F_z = G(b + z h_s)/L \tag{2}$$

If, as in the third working condition, the front and rear wheels are locked at the same time $\frac{du}{dt} = \varphi_0 g$. The normal reaction force of the ground acting on the front and rear wheels is
\[ F_z = \frac{G}{L}(b + \varphi h_z) \]  

(3)

Ground adhesion is \( F_z \), road adhesion coefficient \( \varphi = 0.8 \).

\[ F_\varphi = F_z \varphi \]  

(4)

Finally, the maximum braking torque of the front axle under full load is calculated as

\[ M_\mu = F_\varphi \cdot r = 5232.4 \, N \cdot m \]  

(5)

At this point, the braking torque of the single brake is \( M_\mu = 2616.2 \, N \cdot m \).

2.2. Calculation of tension force

For the calculation of the tensile force, assume that the friction surface of the lining is in contact with the brake disc, and the distribution of unit pressure is uniform, then the braking force distance of the brake is

\[ M_\mu = 2fF_0 R \]  

(6)

In general, you take \( R \) is equal to the mean radius \( R_m \) or the effective radius \( R_e \).

\[ R_m = \frac{R_1 + R_2}{2} \]  

(7)

Therefore, when the brake is locked, the pressing force required for fast friction lining is as follows:

\[ F_0 = \frac{M_\mu}{2fR_m} = 20863 \, N \]  

(8)

According to the simplified model, the contact area between the simplified friction plate and the brake disc is 0.01 \( m^2 \), and the applied pressure on each friction plate is 1.7MPa.

3. Construction of brake finite element model

3.1. Brake design principles

In the new energy truck disc brake design, there are usually several design principles: (1) the principle of standardization, automotive brakes have been standardized, most parts can be universal, so the design should be in accordance with the industry standards. (2) Safety and reliability. Due to the complicated braking conditions, the strength and quality of brake materials should be guaranteed and the cooling capacity should be guaranteed. (3) To reduce public hazards, braking noise should be low, but also to reduce the vibration of braking, especially for new energy vehicles.

3.2. Braking design

Pneumatic disc brake mainly consists of brake disc, brake clamp, support, friction plate, force transfer lever, friction gasket, brake chamber, return device and so on. Pneumatic braking, power transmission lever from chamber of thrust, occurs rotation by the push rod to the brake disc. One end of the push rod is in contact with the inner friction plate, when the inside friction plate pressure the brake disc, brake caliper floating pull by to the other side of the friction brake disk, the friction plate for both sides hold the brake disc, friction, kinetic energy is converted to the friction heat is gradually slow down the car.

Brake disc is the most important part of the disc brake. In the braking, disc brake bears a large load, so there is a large thermal load and the surface temperature can reach 800°C. Thus in such a complex working environment, brake disc is easy to cause some friction, distortion and scratch damage. In order to ensure that the brake disc can work well, many factors should be considered in the design, such as its heat dissipation, wear resistance, corrosion resistance, torsion resistance. Its structure and thickness need to be fully optimized and designed.

Brake clamp and mounting frame must have certain stiffness and strength, and can withstand certain impact and vibration. Brake nippers are generally made of ductile iron or aluminum alloy die-
cast parts. The sliding clamp disc brake is a block. Generally, a gap will be left in the middle of the outer edge of the brake to check and remove the brake block.

Generally, brake block is inlaid, riveted or glued with friction block and metal back plate. Sometimes, in order to insulate heat, a layer of insulation layer would be used between the friction block and back plate and in order to reduce the vibration, a layer of shock absorber would be added. Its shape is generally designed according to the actual situation of the needs of brakes. Its thickness is between 14 and 22 on medium and heavy duty vehicles.

3.3. Modeling

Pneumatic disc brake mainly consists of brake disc, brake clamp, support, friction disc, force transfer lever, friction gasket, brake air chamber, return device and other components, the whole braking process is mainly through the air chamber thrust to make the friction disc locked on the brake disc, relying on friction to achieve braking. SolidWorks was used to model the brake disc, clamp body and bracket to be analyzed, as shown in Figure 2.

![Figure 2. Simplified model of brake disc.](image)

4. Static analysis

In the braking process, when the friction disc just locked the brake disc, sliding friction is into static friction and the brake disc is in the maximum force, in order to verify the strength of brake disc, the combination of brake disc and friction plate is statically analyzed. The analysis process here is as follows:

First, the simplified model of brake disc, brake clamp and bracket are imported into ANSYS workbench, and the modified materials according to Table 2. Limit the translational degrees of freedom in the three directions of the brake disc to the degrees of freedom around X and Y. Apply the maximum braking torque to the threaded hole of the brake disc. The torque is $M = 2616.2N \cdot m$.

| Thrust applied on both sides of the friction plate, and the thrust is 20863N. |
|---|---|---|
| **Table 2. Disc brake materials.** |  |
| parts | Brake disc | bracket | brake caliper |
|---|---|---|---|
| elasticity modulus/Pa | HT250 | QT500-7 | QT500-7 |
| 1.05E11 | 1.62E11 | 1.62E11 |
| Density/kg/m$^3$ | 7220 | 7000 | 7000 |
| Poisson | 0.3 | 0.293 | 0.293 |

When the brake clamp is tightened and the brake disc is locked, the brake clamp is subjected to the maximum load. For this working condition, static analysis is conducted. Firstly, constraints are applied to the return hole to limit its rotation around the x y z axis and the displacement of the z y axis. The load is applied to the position of the lateral friction plate and the position of the lever hole of the brake chamber 20863N.
For the support, it is connected to the vehicle bridge, and the simplified connection bolt is replaced by the fixed support constraint. When the return force of 50N is applied to the return hole to simulate braking, the return effect of the support on the brake clamp is realized. The stress cloud diagram as shown in Figures 3, 4 and 5 was obtained by solving the above three analyses.

Through analysis the results, the maximum principal stress of 55.2MPa can be seen from Figure 3, and its material tensile strength of gray cast iron was 250MPa. Figure 4 brake pliers maximum principal stress occurs when braking return brake chamber wall in contact point size 162.6MPa. The biggest stress occurs in Figure 5 and the maximum principal stress of the bolt is 1.1MPa, the material of QT500-7 tensile strength above 500MPa, this can get enough strength.

5. Modal analysis
Vibration is a common physical phenomenon. The resonance and fatigue damage caused by vibration will seriously endanger the service life and safety of mechanical structures. Modal analysis can directly reflect the inherent characteristics of mechanical structures. Each mode has a specific natural frequency, damping ratio, modal mode, and parameters such as natural frequency and modal mode of components measured are of great significance to the safety of mechanical structures [12, 13]. The differential equation of multi-degree-of-freedom damping system is:

$$\begin{bmatrix} M \end{bmatrix}\ddot{X} + \begin{bmatrix} C \end{bmatrix}\dot{X} + \begin{bmatrix} K \end{bmatrix}X = \{F(t)\}$$

(9)

Where: $[M]$, $[C]$, $[K]$ are the mass matrix of the system, the damping matrix of the system and the stiffness matrix of the system respectively; $[X]$ is displacement vectors, and $\{F(t)\}$ is external force vectors.

In general, because damping has little influence on modal analysis, it is treated as undamped. The equation of free motion can be obtained as follows:

$$\begin{bmatrix} M \end{bmatrix}\ddot{X} + \begin{bmatrix} K \end{bmatrix}X = \{0\}$$

(10)

Structural free vibration of harmonic vibration, the displacement of sine function

$$x = x\sin(\omega t)$$

(11)

Plug in type (10), The corresponding characteristic equation is

$$\begin{bmatrix} K \end{bmatrix} - \omega_n^2 \begin{bmatrix} M \end{bmatrix}X = \{0\}$$

(12)

$\omega_n$ is the circular frequency of vibration and $f = \frac{\omega_n}{2\pi}$ is natural frequency of vibration.

Through modal analysis, the natural frequency and modal array of a structure can be obtained, and the analysis of the inherent parameters of the structure can provide a basis for vibration fault diagnosis and structural optimization design of the equipment. Using ANSYS workbench, the results of the front static analysis were used to analyze the prestressed modal of the finite element model of the brake disc, brake clamp and bracket. The natural frequencies of the first ten orders are extracted through post-processing. Figure 4 shows the specific data.
Because there are similar modes in modal modes, only a few different modes are selected for analysis. The first, third, sixth, and tenth order modes of the support are analyzed, as shown in Figures 6–9.

Figure 6. The bracket first mode.  
Figure 7. The bracket third mode.  
Figure 8. The bracket 6th mode.  
Figure 9. The bracket 10th mode.

According to the vibration mode animation and Figures 6–9, it can be seen that there are corresponding vibration modes in the natural frequencies of order 1, 3, 6 and 10. The first order vibration mode of the support is mainly manifested as the middle vibration in the Z direction relative to the center line. The third mode of vibration is mainly manifested as a torsional deformation about the z-axis, and the strip deformation is the largest, with the corresponding frequency value of 690.41Hz. The vibration mode of the sixth order is mainly manifested as the vibration along the X-axis, the deformation on both sides of the strip is the largest, and the corresponding frequency value is 1160.3Hz. The vibration mode of the tenth order is mainly manifested as the vibration along the Y-axis, and the deformation is the largest at the three-minute point of the long strip, and the corresponding frequency value is 1911.9Hz.

The modes of the first, fourth, seventh and tenth order modes of the brake clamp are taken for analysis, as shown in Figures 10–13.

Figure 10. The brake clamp 1st mode.  
Figure 11. The brake clamp 4th mode.  
Figure 12. The brake clamp 7th mode.  
Figure 13. The brake clamp 10th mode.

Figure 14. The brake disk 1st mode.  
Figure 15. The brake disk 4th mode.  
Figure 16. The brake disk 7th mode.  
Figure 17. The brake disk 9th mode.

Combined with the vibration mode animation and Figures 10–13, it can be seen that there are corresponding vibration modes in the natural frequencies of 1, 4, 7, and 10 orders. The first order vibration mode of the brake clamp is mainly manifested as torsion around the X-axis, and the deformation of the lateral bottom of the brake clamp is the largest, the corresponding frequency value is 245.86Hz. The fourth order vibration mode is mainly manifested as a vibration along the z-axis. The bottom deformation of the brake chamber is the largest, and the corresponding frequency value is
1051.2Hz. The seventh order vibration mode is mainly manifested as the vibration in the X Z plane. The deformation of the lateral bottom of the brake clamp is the largest, and the corresponding frequency value is 1838.4Hz. The tenth vibration mode is mainly manifested in the X Y plane vibration, and the deformation at the bottom of the brake clamp is the largest, the frequency value 3072.2Hz.

Figures 14~17 shows the analysis of the vibration modes of the braking disc in order 1, 4, 7 and 9. Combined with the vibration mode animation and Figures 14~17, it can be seen that there are corresponding vibration modes in the natural frequencies of order 1, 5, 7 and 9. The first vibration mode of the brake disc is mainly manifested as the oscillation of the brake coil around the Y-axis. The mode of the fifth order vibration is mainly shown as oscillating along the Z axis, and the maximum deformation occurs at the positive four angles, with the corresponding frequency value of 1634.9Hz. The seventh mode of vibration is mainly represented by the oscillation around the X-axis. The maximum deformation occurs at the four oblique angles, and the corresponding frequency value is 2187.1Hz. The ninth mode of vibration is mainly manifested as the oscillation around the brake disc along the Z axis. The maximum deformation occurs at the Saturday of the circle, the frequency value 2701Hz.

| Table 3. The first ten natural frequencies of the three parts. |
|-----|-----------------|-----------------|-----------------|
| Order | brake disc (HZ) | brake clamp (HZ) | Bracket (HZ) |
| 1 | 710.04 | 245.86 | 493.09 |
| 2 | 993.12 | 535.34 | 529.58 |
| 3 | 1188.8 | 764.74 | 690.41 |
| 4 | 1473. | 1051.2 | 734.05 |
| 5 | 1634.9 | 1126.6 | 909.85 |
| 6 | 1707.9 | 1295.4 | 1160.3 |
| 7 | 2187.1 | 1838.4 | 1201.7 |
| 8 | 2240.2 | 2345.7 | 1272.6 |
| 9 | 2701. | 2634.2 | 1360.4 |
| 10 | 2782.2 | 3072 | 1911.9 |

From the above analysis, it can be seen that the natural frequency and vibration mode of the friction disc have certain rules, showing obvious symmetrical characteristics. It can be seen from Table 3 that the second order frequency (535.34Hz) of the brake clamp and the second order frequency (529.58Hz) of the support are close to each other, which is easy to cause resonance. Once the external excitation frequency reaches the periphery of this frequency, it will cause strong vibration of the whole system. Therefore, it is necessary to further optimize the parts to minimize the probability of resonance.

6. Conclusions
The static analysis and modal analysis of the main parts of disc brake are carried out by establishing the three-dimensional model of disc brake.

(1) through static analysis, it is concluded that during the braking process, the maximum deformation of the brake disc occurs at the lower position of the both sides of the brake disc, and the maximum stress occurs at the position connected with the wheel hub; The maximum stress of the brake clamp occurs at the connection between the connecting arm and the brake chamber. The maximum stress of the support occurs at the connection between the support and the suspension.

(2) through the analysis and comparison of the first ten order modes of the brake disc, brake clamp and bracket, it is found that the second order frequency of the brake clamp is close to that of the bracket, which is easy to cause resonance.

(3) the strip should be strengthened to ensure the stability of the bracket during braking because during braking the vibration mode displacement of the disc brake bracket mainly occurs on the side strip; According to the analysis of the brake clamp, found that the vibration mode displacement occurs at the bottom of the outside brake clamp and the bottom of the brake chamber during braking,
therefore, these two places should be further strengthened. The vibration displacement of the brake disc mainly occurs in the four sides, which should be strengthened to reduce the damage of vibration to the brake disc.

References
[1] Liu J 2017 Application status and development trend of automobile brake[J]. Agricultural Development and Equipment (5) 72-74
[2] GB 7258-2017 Technical conditions for safe operation of motor vehicles[S]
[3] Zheng Tao 2018 Disc brake simulation optimization based on complex mode analysis[D]. Zhejiang University
[4] Chen Frank 2009 Automotive disk brake squeal an overview[J]. Int. J. Vehicle Design Vol. 51 (Nos. 1/2) 39-69
[5] Wang Wenzhu, Li Jie, Liu Gang, et al. 2019 Simulation of brake noise optimization for automobile disc brakes[J]. Computer simulation 36(1) 171-175
[6] Li Minglei 2014 Research on cae-based structure optimization design of automobile disc brake[D]. Changsha: Hunan University
[7] Merten Stender 2018 Impact of an irregular friction formulation on dynamics of a minimal model for brake squeal[J]. Mechanical Systems and Signal Processing 107 439-451
[8] B H Maruthi, H L Guruprasad and Yogesh Kumar 2014 Transient thermal and structural analysis of rotor disc of disk brake[J]. Indian J. Sci. Res 5(2) 81-88
[9] Zhang Tingting, Wang Guoquan, Wang Shuwen, et al. 2019 Study on transient analysis method of disc brake temperature field[J]. Computer Simulation 36(1) 181-186
[10] Huang Jian, Kong Lingyang and Li Weimin 2015 Analysis of temperature field and stress field of disc brakes[J]. Mechanical Design and Manufacturing vol (2) 143-145
[11] Wang Wangyu 2004 Automobile design[M]. Beijing: Machinery Industry Press
[12] Zheng Gang, Xie Fangwei, Wang Hewei, et al. 2013 Modal analysis of key parts for disc brakes[J]. Mechanical Design and Manufacturing phase II (2) 172-173+176
[13] Yang Longbao 2013 Performance analysis of automotive disc brakes based on ANSYS Workbench[D]. Guangxi: Guangxi University