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Modal Analysis of Automobile Brake Disc

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***Abstract*—The vibration problem of automobile brake disc in the braking process, establish the brake disc model by 3D software CATIA, finite element modal analysis of automotive brake disc is carried out by using the finite element software Ansys-workbench based on modal analysis theory, their front five order modals are extracted, compared with the LMS test, the maximum error is less than 5%, the results are of great value for reference, and provide guidance to reduce the brake disc in the manufacture of vibration and structure optimization.**

***Keywords*— Brake Disc; Mode; Error**

I. INTRODUCTION

The brake disc is the main component of the braking system. Its performance directly affects the driving safety of vehicle. when the vehicle is braking, the excitation frequency is close to the natural frequency of the brake disc, which will cause resonance, generate severe vibration and noise, and affect ride comfort. Therefore, it is necessary to analyze the modal characteristics of the brake disc.

II. MODAL ANALYSIS THEORY

$$M\ddot{x}+C\dot{x}+Kx=f \qquad (1)$$

Where, M is the mass matrix, C is the damping matrix, K is the stiffness matrix, F is the load vector, and the F and X are the functions on the T (time).

When f=0, the Laplace transform is carried out on the left and right sides of the (1) type, and the results are as follows:

$$\left(s^2M+sC+K\right)X(s)=F(s) \qquad (2)$$

Where, F (s) is the Laplace transform of the excitation force.

X (s) is the displacement response.

$$Z(s)=s^2+sC+K \qquad (3)$$

Finishing (2) and (3) form:

$$X(s)=Z(s)^{-1}F(s)=H(s)F(s) \qquad (4)$$

Where, H (s) is a transfer function matrix.

The elements of the L line and the P column in the transfer function can be expressed in the Fourier domain:

$$H_{l,p}(w)=\sum\frac{\phi_{lr}\phi_{pr}}{m_r\left[\left(w_r^2-w^2\right)+j2\xi_r w_r w\right]} \qquad (5)$$

Where, $w_r^2=\frac{k_r}{m_r}$, $\xi_r=\frac{c_r}{2m_r w_r}$,

mr is the mode mass of order r, wr is the modal frequency of order r, ξr is the modal damping ratio of order r, and Ør is the Modal shape of order r.

III. PFINITE ELEMENT ANALYSIS

A. *Experimental object*

Brake disc, as shown in Figure 1.

Fig. 1. *Brake disc*

TABLE I. MATERIAL PROPERTIES

Density	Modulus	Poisson ratio
7100(kg/m^3)	6.4e+10	0.27

B. *Finite element analysis*

The 3D model was created by CATIA.

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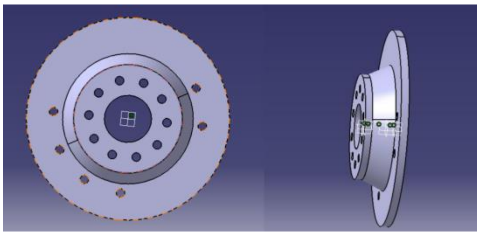
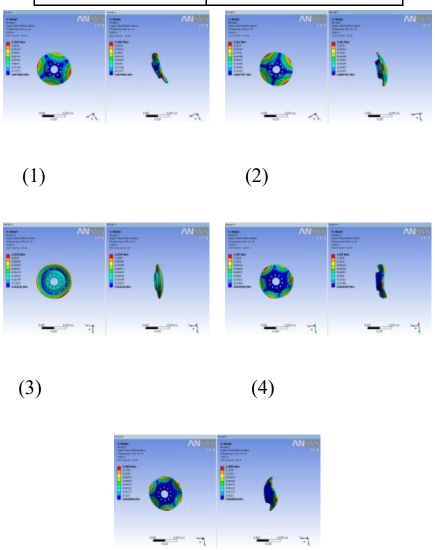


Fig. 2. *Figure 2 Brake disc model*

Meshing and modal analysis of the model into ANSYS-Workbench. The modal frequencies of the first five steps in Table 2 and modal shapes of each order shown in Figure 3.

TABLE II. FINITE ELEMENT MODAL FREQUENCY

Order number	Frequency（Hz）
1	880.14
2	889.23
3	1195.19
4	1567.11
5	1701.04



(1) (2) (3) (4) (5)

Fig. 3. Figure 3 Modal shape

C. *Test analysis*

The test system consists of three parts: vibration excitation system; response acquisition system; analysis and processing system.

The test equipment is shown in Table 3.

TABLE III. EQUIPMENT

Number	Instrument	Number	Model
1	hammer	1	Lianneng
2	force sensor	1	ICP
3	acceleration sensor	5	ICP
4	signal generation and data acquisition	1	LMS SCADASIII 316w
5	analysis and processing	1	Test.Lab 11B

1) Test scheme

a) Support method

During the test, the brake disc was installed with a rubber rope suspension to make it free.

Fig. 4. *Figure 4 Suspension*

b) Layout of measuring points and oscillating point

Four acceleration sensors are uniformly arranged on the outer ring of the brake disc, and an accelerometer is arranged on the upper part of the inner ring to enable it to represent the shape of the brake disc and avoid the location of the modal node. The blue point (measuring point) and the red point (oscillating point) shown in Figure 5.

Fig. 5. *Measuring point and oscillating point*

c) The establishment of geometric model

The coordinate system of the brake disc takes the working plane as the XY plane. The center of the disc is the origin (0, 0, 0). The center of the circle perpendicular to the working plane is Z axis, and the coordinate system is established.

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Start the LMS software and enter the modal analysis interface. On this basis， enter the coordinates of measuring points in the table 4 to complete the model.

TABLE IV. COORDINATES OF MEASURING POINTS

1	s:1	-0.110	0	0.050
2	s:2	0	-0.110	0.050
3	s:3	0.110	0	0.050
4	s:4	0	0.110	0.050
5	s:5	-0.050	-0.030	0

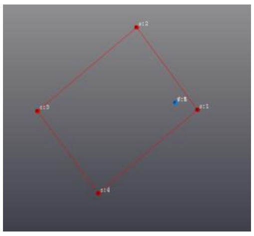


Fig. 6. *Geometric model*

d) Setting of parameters

The bandwidth of the test is set to 2048Hz and the number of lines is set to 6400. It does not need to add windows to the hammering signal. In order to reduce the impact of ambient noise and other contingency factors on the experimental test, 3 percussion is required at each measuring point.

2) Analysis of the results

The LMS Test Lab modal analysis module is opened, and the data collected can be obtained. As shown in Figure 7.

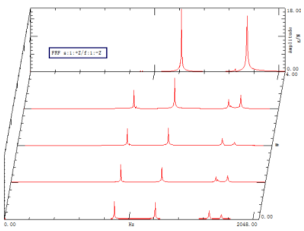


Fig. 7. *Waterfall*

LMS PloyMAX is used for fitting. According to the results of the finite element analysis, the frequency range of 700 - 2000HZ is selected and the stabilization diagram is obtained.

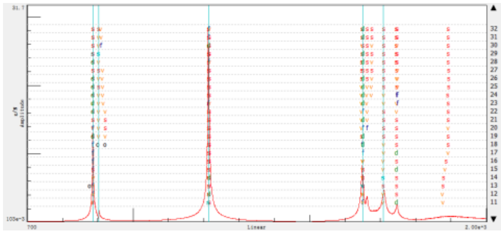
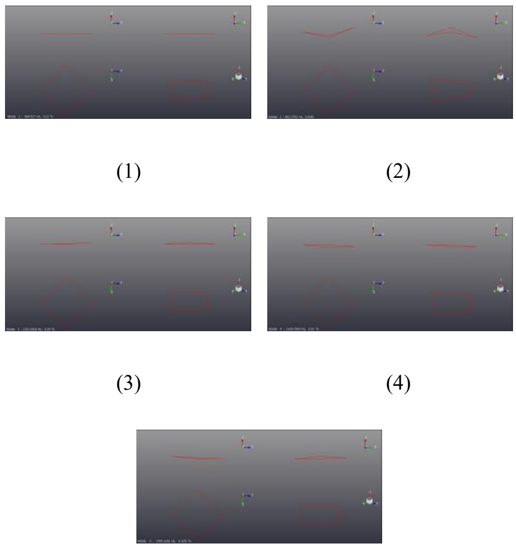


Fig. 8. *Stabilization diagram*

The modal parameters obtained after analysis are shown in the table 5.

TABLE V. MODAL PARAMETERS

Order number	Frequency（Hz）	Damping ratio
1	884.527	0.11%
2	901.379	0.04%
3	1202.043	0.09%
4	1639.758	0.03%
5	1705.328	0.31%



(1) (2) (3) (4) (5)

Fig. 9. *Mode of vibration*

3) Analysis of accuracy

Use LMS Ploymax to extract modalities and perform fitting calculations based on modal numbers. The correlation between the obtained results and the experimental results is over 85%, indicating that the modal number extracted is relatively complete and has high accuracy.

The modes of each order are calculated by MAC, and the modal correlation of each order is mostly within 35%, which shows that it has higher orthogonality. The first and second order modes have higher MAC degrees and different modes of vibration. They have similar two order modes in the result of finite element analysis, so they are kept at the same time.

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TABLE VI. CALCULATION OF MAC

（%）	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5
Mode 1	100	65.553	1.564	1.305	2.103
Mode 2	65.553	100	29.961	31.003	17.983
Mode 3	1.564	29.961	100	14.012	64.157
Mode 4	1.305	31.003	14.012	100	0.0878
Mode 5	2.103	17.983	64.157	0.0878	100

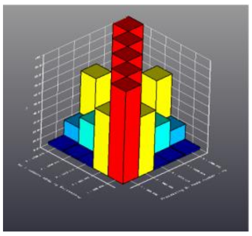

Fig. 10. *MAC*

To sum up, the modal test results of the brake disc have high accuracy and meet the practical application requirements of the engineering.

IV. SUMMARY

The results of the test are compared with the results of the finite element analysis. The results are shown in Table 7. It can be seen that the results of the two methods are very close, and the maximum error result is only about 4.5%.

TABLE VII. ERROR

Order number	Test（Hz）	Finite element（Hz）	Error
1	884.527	880.14	0.50%
2	901.379	889.23	1.35%
3	1202.043	1195.19	0.06%
4	1639.758	1567.11	4.43%
5	1705.328	1701.04	0.25%

The reasons for the error are:

- The material of the brake disc is a composite material. The material property of the finite element software is ideal, and there is an error in practice.
- In actual experiments, there may be some noises in the environment, which will affect the accuracy of acceleration sensors, and will produce errors in the process of signal transmission, both of which will eventually result in errors.
- The error of the test and software simulation is inevitable. However, through the above comparison, the error is very small, so we can combine the two methods to analyze the mode of the brake disc. The two methods are of reference.

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