EFFECT OF CHAMFERED BRAKE PAD PATTERNS ON THE VIBRATION SQUEAL RESPONSE OF DISC BRAKE SYSTEM

En-Cheng Liu¹, Shih-Wei Kung², Syh-Tsang Jenq¹*, Chie Gau¹, Hsin-Luen Tsai³, Cheng-Ching Lee⁵, and Yu-Der Chen⁴

¹ Department of Aeronautics and Astronautics, National Cheng Kung University, Tainan, Taiwan, R.O.C.
² Dassault Systemes Simulia Central, Cincinnati, Ohio, USA
³ Department of Electronic engineering, Kao Yuan University, Kaohsiung county, Taiwan, R.O.C.
⁴ Aeronautical Systems Research Division, Chung-Shan Institute of Science & Technology, Taichung, Taiwan

Abstract

This paper is concerned with the disc brake squeal problem for automotive vehicles. The purpose of the present research tries to reduce the instability of the squeal mode and to improve the comfortless discord volume in the car. In the current study, the ABAQUS/Standard (Implicit method) finite element numerical model was constructed to perform the dynamic contact vibration analysis of a disc brake system containing a brake disc, brake pads, caliper, and brake shoe. Brake pads with specific chamfer patterns and a brake disc with cooling ribs for ventilation and heat transfer were constructed numerically in the present system. The effect of brake pad chamfer pattern on the unstable frequency distribution, which may result in the brake squeal phenomenon, is studied. A disc rotational velocity of 2.5 rad/s was studied numerically when the pad was uniformly pressurized to 200 psi (1.379 MPa) on the brake shoe. The Lanczos method is used to find the natural frequencies of brake system in question, and then use the complex eigenvalue method to extract the unstable squeal high frequencies between the brake disc and brake pads for specific chamfer patterns with dry friction. A lining friction coefficient of 0.4 was selected to in the current analysis in order to determine the uncomfortable brake squeal vibration frequencies distribution with specific chamfer patterns.

Keywords: brake squeal, friction, chamfer, Lanczos method, complex eigenvalue method.

1. Introduction

Brake squeal problems are always exist in the brake system of the vehicle, there are many engineers and academics more and more attach importance to friction induced vibration problem of brake squeal. Brake noise frequency can be classified into three categories frequency region. The three categories are judder, groan/moan, and squeal as shown in Figure 1. Among this frequency, engineers and academics used experimental, analytical, and computational methods to analysis and reduce the brake squeal problems. Many engineers and academics used the complex eigenvalue method to predict the brake squeals in the unstable region. Kung et al. [1] were used the complex eigenvalue method for a detailed analytical study in order to obtain a better understanding of this solution technique. Modal participation factors were calculated to examine the modal coupling mechanism. Parametric studies were also performed to find out the effects of friction coefficient and rotor stiffness. In order to reduce or eliminate squeal, it was very important to understand the coupling mechanism so that the key component(s) can be modified accordingly, Kung et al. [2] demonstrated a quantitative method to define system mode shapes using the concept of modal participation factors. This method was implemented on a front disc brake system to identify the modal coupling mechanism associated with its high frequency squeal. Complex eigenvalue analysis was carried out and the squeal frequency was correlated. Shi et al. [3] were used the complex eigenvalue analysis for brake noise analysis and noise reduction in GM. Finite element models were validated with component modal testing. Also dynamometer and vehicle test results were compared with finite element results. From several vehicle level brake noise analyses, their experience shows that complex mode shapes were capable of depicting the noise generating mechanisms, and the complex eigenvalues are able to depict the noise frequencies. Bajer et al. [4] were performed the complex eigenvalue extraction at a deformed configuration; nonlinear effects were taken into account in the modal analysis. An example case was used to illustrate the importance of friction-induced damping and nonlinear effects in brake squeal analysis. It was found that the inclusions of lining wear, geometric nonlinearities, and positive as well as negative frictional damping effects had significant influence on the brake squeal predictions. Yue et al. [5] in order to better understand the mechanism of squeal generation, this study started with the component modal alignment analysis around problem frequencies based on the component EMA (Experimental Modal Analysis) data in free-free condition. Liu et al.[6] were presented a simple finite element model as a rapid engineering tool for designing pad shape to reduce high frequency brake squeal and discussed the mechanism, effect, and design methodology of pad shape on squeal. Liu et al. [7] and [8] were performed using the commercial available code – ABAQUS/Standard (Implicit method) to calculate the natural frequencies of brake disc model extract the complex eigenvalues of the contacted brake disc and pads model. The unstable squeal frequencies of the brake disc compressed by a pair of brake pads were determined.

* Corresponding author, Professor, Email: stjeng@mail.ncku.edu.tw, Fax: (Taiwan)-6-208-3641.
2. Equation of motion of the disc brake system

The equation of motion for a disc brake system can be written as follows:

$$[M]\dddot{u} + [C]\dot{u} + [K]u = [F_{\text{external}}].$$  \hspace{1cm} (1)

where $[M]$ is the mass matrix, $[C]$ is the damping matrix, $[K]$ is the stiffness matrix, $\{u\}$ is the generalized displacement vector, and $[F_{\text{external}}]$ is external force matrix, respectively. The forcing term $[F_{\text{external}}]$ can be as the function of the velocity as follows:

$$[F_{\text{external}}] = [f] \{\dot{u}\}.$$  \hspace{1cm} (2)

A homogeneous equation of can be obtained by combing equation (1) and (2) by moving the forcing term to the left hand side

$$[M]\dddot{u} + [C - f]\dot{u} + [K]u = 0.$$  \hspace{1cm} (3)

From equation (3), the determination of instability formulation is derived as

$$\zeta = \frac{\omega_n C_n}{\sqrt{MK}} < \frac{C - f}{2\sqrt{MK}} < 0,$$  \hspace{1cm} (4)

where $\omega_n = \sqrt{\frac{K}{M}}$, $\zeta = \frac{C_n}{C_c}$, $\lambda_c(D) = \frac{\omega_n(D)}{2\omega_n}$, and $C_c = 2M\omega_n = 2\sqrt{MK}$. In addition, $\omega_n$ is the undamped natural frequency of the disc brake system, $\zeta$ is damping ratio, $C_c$ is critical damping coefficient and $\lambda_c = a + bi$ is the eigenvalues of the brake system, where $a$ is the real part and $b$ is the imaginary part of the eigenvalue. If $C < f$ occurs, the disc brake system will appear negative damping effect such that the disc brake system occurs unstable phenomenon.

3. Finite Element Model

A passenger car front brake system with a ventilated disc and single-bore floating caliper is used here as an example [4]. High frequency squeals were observed in dynamometer tests as shown in Figure 2. The finite element model consists of a rotor, caliper, bracket, piston, and pads as shown in Figures 3 and 4. This model is meshed with 26,056 elements and 38,120 nodes.

The brake lining material is assumed to be transverse isotropic. Other components are modeled as linear elastic isotropic materials as shown in Table 1. Boundary conditions are applied at the center of the hub and at the bolted location of the bracket. Interfaces between components are modeled as contact pairs with master and slave surfaces. These contact pair surfaces include rotor-lining surfaces, caliper-pad surfaces, piston-pad surfaces, and bracket-shoe abutment surfaces.

A commercial finite element software Abaqus/Standard is selected. The analysis flow chart is shown in Figure 9. Several non-linear static steps are performed first to calculate steady state equilibrium of the system in a specific braking condition. Within these static steps, the brake pressure is applied and rotor angular motion is imposed. Then natural frequencies of the system at braking condition are extracted by using the Lanczos eigenvalue solver. Finally, complex modes are extracted in the complex eigenvalue step and unstable squeal frequencies are identified.

To study the effect of lining chamfer, the original pad shape (Figure 6) are modified to the new pad shape (Figure 7 and 8) as suggested by Ref. [6] (Figure 5). This modification includes 21 mm parallel chamfers, center slot, as well as top and bottom center V-chamfer. The finite element model are revised by swapping the inner and outer pad models with the new ones. Contact surfaces are redefined and the procedure described above is performed again. Results are discussed in the following session.
4. Results and Discussion

The focus of this study is on the reduction of high frequency squeals which ranges from 5 kHz to 16 kHz. In order to cover such a wide frequency range, 200 modes are requested in the complex eigenvalue calculation.

In this case study, the basic parameters chosen for the parametric studies are friction coefficient and brake disc rotation speed. The friction coefficient $\mu = 0.4$, and brake
disc rotation speed is 2.5 rad/s. The results of complex eigenvalue analysis is in the form of $\lambda = a + bi$, where $a$ is the real part and $b$ is the imaginary part of the eigenvalue which presents frequency of the unstable mode.

All calculated eigenvalues are then plotted on a complex plane as shown in Figure 10. Note that the left half plane shows the stable region while the right half plane shows the unstable region. It can be seen that the squeal frequencies of 6, 7, 10, 13 and 16 kHz from the test data(4) are all predicted in the complex eigenvalue results as unstable modes. It is also seen that the instability of most unstable modes are reduced when the chamfer design is applied. Table 2 listed the predicted modes and their instabilities before and after the lining chamfer.

Table 2 The FEM analyzed unstable modes and frequencies corresponding to dynamometer test results(4).

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>68</td>
<td>6.3353 kHz</td>
</tr>
<tr>
<td>110</td>
<td>10.105 kHz</td>
</tr>
<tr>
<td>139</td>
<td>12.085 kHz</td>
</tr>
<tr>
<td>152</td>
<td>13.3 kHz</td>
</tr>
<tr>
<td>187</td>
<td>15.848 kHz</td>
</tr>
</tbody>
</table>

5. Conclusion
In the present study, a commercial finite element code - ABAQUS/Standard (Implicit method) is used to predict the dynamic instability of a contact vibration disc brake system before and after a specific lining chamfer treatment. This type of lining chamfer(6) consists of a center slot, 21-mm parallel chamfers on both leading and trailing side, and V-chamfers at the top and bottom center. Major squeal frequencies that observed in the dynamometer test(4) are predicted in the complex eigenvalue analysis as unstable modes. The effect of lining chamfer is also predicted and the results show reductions of instability for many modes. The FEM predicted frequencies of the unstable squeal modes corresponding to test results are 6.3353, 10.105, 12.085, 13.3, and 15.848 kHz. It can be seen that the squeal frequencies of 6, 7, 10, 13 and 16 kHz from the test data are all predicted in the complex eigenvalue numerical results as unstable modes. It is also seen that the instability of most unstable modes are reduced when the chamfer design is applied. Table 2 listed shows the predicted modes and their instabilities before and after adopting the lining chamfer. Further investigation in component motions is needed to get the insight of this lining change so that...
optimum lining shape can be proposed. This analysis provides designers with a powerful analysis tool for the brake squeal vibration study of an advanced disc brake system.

Acknowledgement
The authors are grateful to the Department of Industrial Technology, Ministry of Economic Affairs (MOEA), Taiwan, R.O.C. for their support of the research presented under the Contract No. 97-EC-17-A-99-R9-0800.

References


