

Automotive Brake Squeal Simulation and Optimization

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ABSTRACT

This work carries out complex modal analyses and optimizations to resolve an 1800 Hz front brake squeal issue encountered in a vehicle program development phase. The stability theory of complex modes for brake squeal simulation is briefly explained. A brake system finite element model is constructed, and the model is validated by the measurement in accordance with the SAE 2521 procedure. The key parameters for evaluating the stability of the brake system complex modes are determined. The modal contributions of relevant components to unstable modes are analyzed and ranked. Finally, in order to resolve the squeal issue, the design improvements of rotor, caliper and pad are proposed and numerical simulations are carried out. The obtained results demonstrate that the optimized rotor and pad design can alleviate the squeal issue significantly while the optimized clipper design could essentially eliminate the squeal issue. The analytical solutions obtained from the proposed pad design are validated by the experimental measurements.

CITATION: Yang, S., Sun, Z., Liu, Y., Lu, B. et al., "Automotive Brake Squeal Simulation and Optimization," *SAE Int. J. Passeng. Cars - Mech. Syst.* 9(1):2016, doi:10.4271/2016-01-1298.

INTRODUCTION

Brake system complaints are among the top vehicle quality concerns as demonstrated by market surveys such as J.D. Power IQS (Initial Quality Study). Among those brake related complaints, NVH problems often claim the lion's share. The most notorious brake system NVH issues are squeal, rattle, groan, judder and moan.

The automobile brake squeal problem is usually a single frequency event, characterized by high intensity up to 120 dB(A), poor sound quality, and a frequency that falls into a wide range from 0.9 to 18 kHz. Its mechanisms are characterized by excitation, modal coupling and mode lock-in. It is well accepted from a dynamics perspective that it is caused by brake system instability, represented by a self-sustained energy-feed-in cycle due to the physics of negative damping. Thus, when a brake system possesses intrinsic characteristics prone to instability and a proper excitation source happens to be present, the brake system may generate squeal. The understanding, analysis and prevention of the issue are further complicated by the nonlinear and random nature of the problem, such as contact mechanism, stick-slip process and tribological effects. Therefore, once the problem emerges, it is difficult to resolve, and in fact it is one of the toughest issues during the brake system development effort.

In recent years, considerable efforts have been devoted to the understanding and resolution of the problem as evidenced in many published literatures, e.g. [1][2][3][4][5][6]. OEM's and brake system suppliers have carried out extensive studies on the mechanism, simulation methodology, and experimental technology. This work concentrates on a typical brake squeal problem encountered in a

prototype vehicle for a vehicle program. During a program development stage, a front wheel brake system repeatedly developed a typical 1800 Hz brake squeal noise. The factors influencing such event are many, such as the pad, caliper, anchor bracket, knuckle, other brake components, and the interactions of them. Many proposed efforts, such as various bench test and vehicle measurement, could not find an effective fix. In this work, based on the theory of the stability of the complex mode for brake systems, a finite element model of the brake system is established first, and then the complex modal analysis is carried out. The evaluation method and factors that influence brake squeal are looked into. And finally, the proposed resolution measure is investigated and validated by testing.

1. ROOT CAUSES OF UNSTABLE MODES

For a linear vibration system with N degrees of freedom and viscous damping, the dynamic mathematical expression can be written as:

$$M\ddot{u} + C\dot{u} + Ku = Q,$$

where u is the displacement vector, M is the mass matrix, C is the damping matrix which includes the friction effect, K is the stiffness matrix, and Q is the external load. In the case of Q being 0, the above equation becomes

$$M\ddot{u} + C\dot{u} + Ku = 0,$$

which describes the inherent properties of the system. In general, the damping matrix and the stiffness matrix are asymmetric, thus the above equation can not be solved for real modes. In this case, it necessitates the complex mode analysis

In general, the above equation has a solution of the form:

$$u = \phi e^{\lambda t},$$

where λ and ϕ are the complex eigenvalues and eigenvectors respectively. The complex eigenvalues are of the form $\lambda_j = \alpha_j \pm i\omega_j$, where $i = \sqrt{-1}$, α_j and ω_j represent the real part and imaginary part respectively with $j = 1, 2, \dots, N$.

From the nature of the solution, it is seen that the real part of the eigenvalue represents the attenuation of the vibration. When $\alpha_j < 0$, the complex mode will cause the system to undergo a damped vibration with decreasing amplitude, and therefore this mode is a stable one. When $\alpha_j = 0$, this complex mode represents an undamped vibration, which could be considered as a critical stable mode. And when $\alpha_j > 0$, the complex mode represents a vibration with increasing amplitude, and thus it diverges as an unstable mode. The imaginary part of the eigenvalue represents the frequency of the vibration mode.

If the contact mechanism and the friction behavior between the brake rotor and pad are not considered, the system would possess a positive-definite M , a symmetric C and a symmetric K . This situation will not lead to the occurrence of any unstable modes. If the contact friction mechanism is included in the formulation, after discretization and linearization, the following equation is derived:

$$M\ddot{u} + (C + C_f)\dot{u} + (K + K_f)u = 0$$

where C_f and K_f are the friction damping matrix and friction stiffness matrix respectively due to the contact mechanism and friction effects between the rotor and pad. It is this addition of the extra quantities that leads to the likelihood of unstable modes.

Figure 1 shows the modeling of the contact mechanism for the rotor and pad. Suppose the element coordinate system is (U, V) , the coefficient of friction is μ , and the contact stiffness is k . Then the normal force and tangential force between points n, m , can be expressed as:

$$\begin{Bmatrix} F_{Un} \\ F_{Vn} \\ F_{Um} \\ F_{Vm} \end{Bmatrix} = \begin{bmatrix} -k & 0 & k & 0 \\ \mu k & 0 & -\mu k & 0 \\ k & 0 & -k & 0 \\ -\mu k & 0 & \mu k & 0 \end{bmatrix} \begin{Bmatrix} U_n \\ V_n \\ U_m \\ V_m \end{Bmatrix}$$

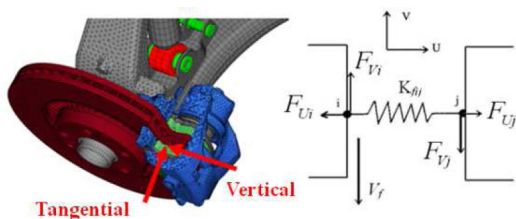


Figure 1. Finite element contact model on contact surface between rotor and pad.

It is apparent that the element stiffness matrix is asymmetric, and all elements connecting the contact surface have asymmetric stiffness matrix. Therefore, $K + K_f$ is asymmetric. This means that the eigenvalues could be of a complex nature, and their real part might be positive.

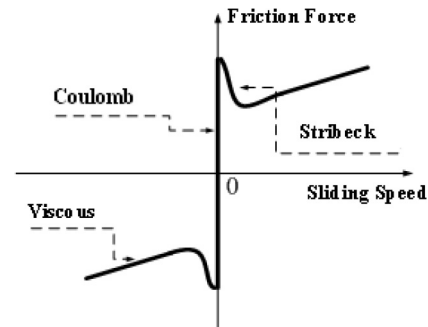


Figure 2. Stribeck friction model.

The establishment of C_f requires detailed information regarding the relationship between the friction force and relative speed. Figure 2 displays the familiar Stribeck model for the relation between the friction force and relative speed [7][8]. It is evident that in the Stribeck region, the friction coefficient is negative. As a result, the damping matrix $C + C_f$ might be negative.

2. ESTABLISHMENT OF FINITE ELEMENT COMPLEX MODE ANALYSIS MODEL

The brake squeal is usually caused by unstable high frequency modes between 0.9 to 18 kHz. In order to carry out the analysis, it is necessary to establish an accurate and reliable brake system finite element model that can correctly capture the characteristics of such high frequency modes. To ensure the model accuracy, the friction coefficient, the material properties of the pad and silencing sheet, hydraulic cylinder oil film, bearing, stiffness of the suspension joints, tire and bushings, are all obtained by testing. Based on the measured data, the models of brake system and the suspension system are assembled, and a brake system complex stability analysis model is established as shown in Figure 3.

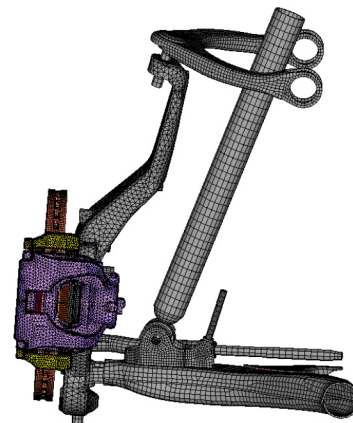


Figure 3. Brake squeal complex mode analysis finite element model.

The model contains 251269 solid and shell elements with 359192 nodes, 12 contact friction pairs including pad, rotor, and caliper, as well as 5 other friction pairs in the suspension system.

The anisotropic material properties of the friction pad are obtained by ultrasonic testing [9]. By comparing the modal results obtained by testing with the calculated modal results using these measured material properties, the material parameters are modified so as to ensure that the modal results from testing and simulation are correlated. In this way, a correct set of material parameters is obtained. The comparison of the first and second modes for the pad from the testing and simulation is shown in Figure 4. The modified material properties are listed in Table 1, where E is the elastic modulus, G the shear modulus, and ν the Poisson's ratio, with 1 denoting the vertical direction and 2 the tangential direction.

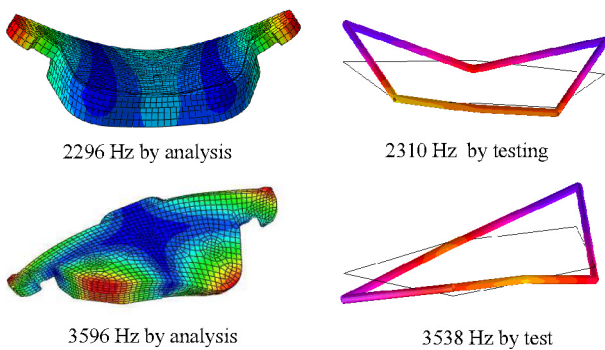


Figure 4. Comparison of simulation and test results for the first bending and torsion modes.

Table 1. Measured and modified friction pad anisotropic material properties.

	E_1	E_2	E_3	ν_{12}	ν_{13}	ν_{23}	G_{12}	G_{13}	G_{23}
Measured	10	10	2.8	0.19	0.32	0.32	4.2	2.1	2.1
Modified	6.5	6.5	1.8	0.19	0.32	0.32	2.7	1.3	1.3

3. BRAKE SYSTEM COMPLEX MODE STABILITY ANALYSIS

3.1. Simulation Setup

Brake squeals occur in a specific brake pressure, speed, temperature, and friction condition. In theory, it is desirable to analyze the stability of brake system considering all design variables involved in a complicated brake system, along with many operating and environmental conditions such as brake pressure, friction coefficient and rotating speed. In reality, as these parameters are continuously changing, it is impractical and impossible to carry out such a thorough and exhaustive calculation covering all possible combinations of parameters and events. Based on past simulation works for many programs and the specific situation of the current brake system, this work establishes a load case matrix for analyses as shown in Table 2.

Table 2. Operating condition matrix used for analyses.

Rotating Speed	Friction coefficient	Brake Pressure(Bar)				
Forward 5 r/s	0.4	3	8	16	24	30
	0.5	3	8	16	24	30
	0.6	3	8	16	24	30
Backward 2 r/s	0.4	3	8	16	24	30
	0.5	3	8	16	24	30
	0.6	3	8	16	24	30
	0.7	3	8	16	24	30

3.2. Stability Criterion

Whether an unstable mode could lead to brake squeal or not also depends on the nature of negative damping as well as the modal frequency [10]. This work assesses the stability of the brake system by using the system's negative damping coefficient which is defined by

$$c_j = \frac{-2\alpha_j}{\omega_j},$$

where α_j and ω_j are the real part and imaginary part of the j th eigenvalue respectively. If the negative damping coefficients are of the same value, the squeal propensity is construed the same. In this case, the real part of the eigenvalue, i.e α_j , is linearly proportional to the frequency, which is equivalent to saying that a higher frequency mode can afford to have a higher real part of the eigenvalue.

In theory, the mode with its negative damping coefficient less than 0 is an unstable mode. However, a more realistic assessment is that only when the negative damping coefficient reaches a certain value, the brake squeal would be likely to occur. Therefore, in order to assess whether a brake system is imminent to become unstable, it is important to establish a critical value of the negative damping coefficient which signals the inception of instability.

According to past experience and specific requirements of the current project, the critical negative damping coefficient is taken as -0.01. If a negative damping coefficient of a system is less than this value, it is construed as the occurrence of brake squeal is more likely. It needs to point out here that this requirement is not considered very stringent.

3.3. Simulation Result

Using the load condition matrix in table 2 and the analysis procedure outlined in Section 3.1, 35 complex modal analyses are carried out. The computer resource involves a top line desktop, and the computation takes more than 130 hours of runtime. There are many simulation results obtained, and only the main results corresponding to 2 friction coefficient values of 0.5 and 0.6 are selected to be displayed here in Figures 5 and 6 respectively. In each diagram, one friction coefficient is combined with 5 brake pressure values and one rotating speed under the forward driving condition,.