

On the Coupling Stiffness in Closed-Loop Coupling Disc Brake Model through Optimization

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ABSTRACT

The study and prevention of unstable vibration is a challenging task for vehicle industry. Improving predicting accuracy of braking squeal model is of great concern. Closed-loop coupling disc brake model is widely used in complex eigenvalue analysis and further analysis. The coupling stiffness of disc rotor and pads is one of the most important parameters in the model. But in most studies the stiffness is calculated by simple static force-deformation simulation. In this paper, a closed-loop coupling disc brake model is built. Initial values of coupling stiffness are estimated from static calculation. Experiment modal analysis of stationary disc brake system with brake line pressure and brake torques applied is conducted. Then an optimization process is initiated to minimize the differences between modal frequencies predicted by the stationary model and those from test. Thus model parameters more close to reality are found. Unstable mode prediction results using parameters before / after optimization are compared with those from brake noise bench test. The results after parameter optimization are in good correlation with test result and illustrate obvious improvement of predicting accuracy. Finally, the presented method is used to study the relationship between the coupling stiffness and brake conditions, i.e. brake-line pressure. The result shows accordant with literature.

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INTRODUCTION

Due to its complex nature brake squeal has been a challenging problem for researchers and engineers since 1930's and much work is still needed to further the understanding of the mechanism [1, 2, 3, 4]. However, recent years much progress and insight have been gained in theory and engineering practice. For example, modifying the structural parameter of brake components can lighten the squeal problem to a satisfactory level [5, 6].

Many different modeling and analyzing methods had been developed. Complex eigenvalue analysis is one of the common used methods. For disk brake systems, Liles [7] included friction coupling between rotor and pad as off diagonal terms in the stiffness matrix and used a complex eigenvalue analysis to assess the stability of a brake system. Guan, Jiang, etc [8, 9, 10, 11] built closed-loop coupling disk brake model based on modal synthesis theory and developed a set of analysis methods such as substructure modal composition analysis, substructure modal parameter sensitivity analysis, feed-in energy

analysis, etc.. These methods can analyze the influence of components' structural parameters to squeal and determine which one is dominant. The analysis results reached good agreement with those from test.

In closed-loop disc brake model, the dynamic characteristics of components are expressed in the form of modal parameters. The contacts between substructures are represented by spring and friction couplings between coupled nodes [8]. Stiffness of coupling springs is very important parameter in the model. Choosing suitable values is essential to successful simulation and prediction of brake squeal.

A few discussions can be found on determining coupling stiffness values in public literature. Chen [12] declared that it should be 1000 times of local stiffness. Jiang [14] believed that the coupling stiffness should make the coupled structure mechanical equivalent to the real structure. Abu Bakar [13] provided the following guidelines to determine stiffness values of coupling springs when building a FE (finite element) disc brake assembly model:

- Any components that allow to slide between them the spring value is set to a very low stiffness e.g. around 0.5 N/m;
- Any components that restrict movement at any directions e.g. between bolt and caliper arm, the spring value is set a very high stiffness e.g. around 1×10^{10} N/m;
- Any components that may experience to vibrate e.g. back plate, the spring stiffness is set around 1×10^6 N/m.

Obviously these references only provided some principle suggestions on this issue. For a specific modeling task, the stiffness values are to be tuned based on a matter of trial-and-error process, which is laborious, time consuming and the result largely depends on researcher's experience and discretion[12, 13].

In this paper, a method to determine the values of coupling stiffness between brake pad and disc rotor and between inner pad (piston pad) and caliper based on modal test and optimization is presented. Experiment modal analysis of stationary disc brake system with brake-line pressure and brake torques applied is conducted. Modal parameters are extracted. Then an optimization process is initiated to minimize the differences between experimental and model predicted modal frequencies. Unstable mode prediction results using parameters before / after optimization are compared with those from brake noise bench test. The comparison shows the improvement of predicting accuracy. Finally, the presented method is used to study the relationship between the coupling stiffness and brake-line pressure. The result shows accordant with literature.

CLOSED-LOOP COUPLING MODEL

Closed-loop coupling brake model is based on modal synthesis method. The dynamic behavior of brake components is expressed in the form of modal parameters which are calculated from their individual FE (finite element) model. Then the components are assembled to become a system model [8, 14].

The free vibration equation of the brake system is:

$$M\ddot{u} + (K - K_f)u = 0 \quad (1)$$

where M and K are discretized mass and stiffness matrices respectively, u is nodal displacement vector, K_f is coupling matrix referring to the relationship between components, which includes spring and friction coupling. K_f is an asymmetric matrix.

Constructing coordinate transformation equation as:

$$u = \Phi q$$

$$\Phi = \begin{bmatrix} \Phi_A & & & & \\ & \Phi_B & & & \\ & & \Phi_C & & \\ & & & \Phi_D & \\ & & & & \Phi_E \end{bmatrix} \quad (2)$$

where $\Phi_A \sim \Phi_E$ are modal shape matrices of components which are modal mass normalized respectively; q is modal coordinate vector in component mode space Φ .

From Eq.1 and Eq.2 it can derived:

$$\ddot{q} + K_{sys}q = 0 \quad (3)$$

$$K_{sys} = \begin{bmatrix} \lambda_A & & & & \\ & \lambda_B & & & \\ & & \lambda_C & & \\ & & & \lambda_D & \\ & & & & \lambda_E \end{bmatrix} - \Phi^T K_f \Phi \quad (4)$$

where diagonal matrices $\lambda_A \sim \lambda_E$ are modal frequency matrices of each component respectively.

Complex eigenvalues and corresponding eigenvectors can be obtained by eigenanalysis of Eq.3. Those modes that eigenvalues have positive real parts are unstable modes, and may result in squeal. The imaginary part of the eigenvalue represents the unstable frequency.

K_f is the coupling matrix. For coupling surface between friction material of brake pad and disc rotor, the coupling matrix of a coupled node pair is:

$$K_f = \begin{bmatrix} -K_1 & 0 & K_1 & 0 \\ -\mu K_1 & 0 & \mu K_1 & 0 \\ K_1 & 0 & -K_1 & 0 \\ \mu K_1 & 0 & -\mu K_1 & 0 \end{bmatrix} \quad (5)$$

where K_1 is coupling stiffness between friction block and disc rotor and μ is friction factor.

For coupling surface between backplane of inner pad and caliper, the coupling matrix is:

$$K_f = \begin{bmatrix} -K_2 & 0 & K_2 & 0 \\ 0 & 0 & 0 & 0 \\ K_2 & 0 & -K_2 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \quad (6)$$

where K_2 is coupling stiffness between backplane of inner brake pad and caliper housing.

K_f consists of coupling stiffness and friction factor. Determining the coupling stiffness values is an important task in the modeling process. To improve the correlation between model predicted and experimental results, the contact stiffness should reflect the actual situations when squeal occurs.