

Study on a Closed-Loop Coupling Model without Coupling Spring

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ABSTRACT

Closed-loop coupling model, based on complex eigenvalue analysis, is one of the most popular and effective methods for brake squeal analysis. In the model, imaginary coupling springs are used to represent the normal contacting force between coupled nodes. Unfortunately, the physical meaning of these coupling springs was seldom discussed and there's no systematic method to determine the value of spring stiffness. Realizing this problem, this paper, based on finite element model and modal synthesis technique, develops a new closed-loop coupling disc brake squeal model without introducing imaginary coupling springs. Different from the traditional model where two nodes at coupling interface are connected through a spring, these node-pairs in the new model are assumed to remain in tight contact during vibration. Details of the model, including force analysis, coordinate reduction and transformation and complex eigenvalue decomposition are given in this paper. Finally, the presented method is applied on the modelling of a squealing disc brake, which shows good correlation between model prediction results and those from bench test. Besides, because there are less indeterminate model parameters, the time for parameters tuning process is greatly reduced.

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INTRODUCTION

Brake squeal causes significant warranty costs and strongly degrades customers' perceptions of the quality and reliability of vehicles. Severe and continuing vibration of braking system may also lead to early fatigue failure of relevant components. The problem becomes more prominent due to the increase of driving speed, the application of new frictional materials and lightweight structural designs [1].

Brake squeal is a complicated dynamic problem which involves multi-disciplines and multiple spatial and time scales. Lots of research works have been carried out through experimental, analytical and numerical methods in the last few decades. Although the understanding of the mechanism of brake squeal is still insufficient and incomplete, much progress has been made in analyzing, diagnosing and formulating structural modification scheme to suppress the squeal [2].

The analytical methods can be divided into frequency-domain and time-domain methods. Frequency-domain method is widely used. In this method, direct finite element (FE) model or modal synthesis model is built, the system's unstable modes or squeal modes are calculated using complex eigenvalue analysis, and further analysis methods can be used to investigate the influences of system parameters on the unstable modes, such as components' geometrical profiles, material properties, modal properties and frictional behaviors. Time-domain method, on the other hand, is based on

nonlinear and transient analysis of the system, and is suitable for cases where the vibrating system is far from its equilibrium steady sliding state, and the effects of nonlinearities are non-negligible or the system parameters are time dependent but not periodic [3-4]. Additionally, the transient analysis of brake squeal is usually conducted on the basis of the result of complex eigenvalue analysis. So the computation efficiency is low [5].

Liles[6] was the first researcher who built a frequency-domain model for disc brake squeal analysis. The friction between disc rotor and pads was incorporated as "geometric instability" into the model. A complex mode formulation was derived for the system. The real part of the eigenvalue indicates the stability (or lack of it) of the system. If the real part of an eigenvalue is positive, the corresponding imaginary part was thought to be a possible squeal frequency. He constructed the friction stiffness matrix using relative displacements between mating surfaces. Later, Nack presented a detailed instruction on how to build the frictional stiffness matrix and clearly stated the meaning of using complex eigenvalue analysis [7, 8].

Guan and her research team [9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19] considered brake squeal as a self-excited vibration problem due to the instability of the frictional closed-loop coupling of the system. It's caused by the mismatch of structural parameters of brake components. Adjusting the structural parameters matching properly is an effective way to solve this problem. Based on modal synthesis and FE method, a closed-loop coupling model of disc brake was

Where μ and K are the friction coefficient and stiffness of coupling springs respectively. For those coupling surfaces without friction, K_f is simplified as:

$$K_f = \begin{pmatrix} -K & 0 & K & 0 \\ 0 & 0 & 0 & 0 \\ K & 0 & -K & 0 \\ 0 & 0 & 0 & 0 \end{pmatrix} \quad (3)$$

In the model, the dynamic characteristic of each substructure (component) is represented by its modal parameters, and the coupling stiffness matrix reflects the coupling relationships between substructures. In this way, the motion differential equation of the system can be established. Transform the nodal physical coordinates to substructure modal space, the character equation of the system can be derived. By solving this eigenvalue problem, the eigenvalues and the corresponding eigenvectors can be obtained. The imaginary part of the eigenvalue represents the frequency, while the real part stands for the damping of this mode. Therefore, those modes in which the real parts of eigenvalue are positive (negative damping) are unstable modes and may lead to squeal. And higher real part indicates stronger squeal propensity.

Obviously, the assumption of coupling spring is a foundation of closed-loop coupling model, and the stiffness values have dominant effects on the accuracy of the model. However, to the best of authors' knowledge, this important assumption has not been well justified from a physical aspect. What's worse, determining the value of this essential parameter deeply relies on empirical treatment in most cases, and there is still no systematic and rigorous method to solve this issue in public literatures.

NEW CLOSED-LOOP COUPLING MODEL

To build a new closed-loop coupling model, following modelling assumptions are made to the system:

1. The system can be linearized approximately when it's vibrating closely to its equilibrium states,
2. Direction of friction force remains unchanged during vibration,
3. When vibrating, the coupling node pairs remain in tight contact in normal direction of the coupling interface.

It is clear that the new model inherits the first two assumptions of the traditional one whereas sets-up new coupling conditions of the nodes.

Substructures and Force Analysis

The objective disc brake is divided into 6 substructures named A to F. The interactions of different substructures are shown in Figure 2.

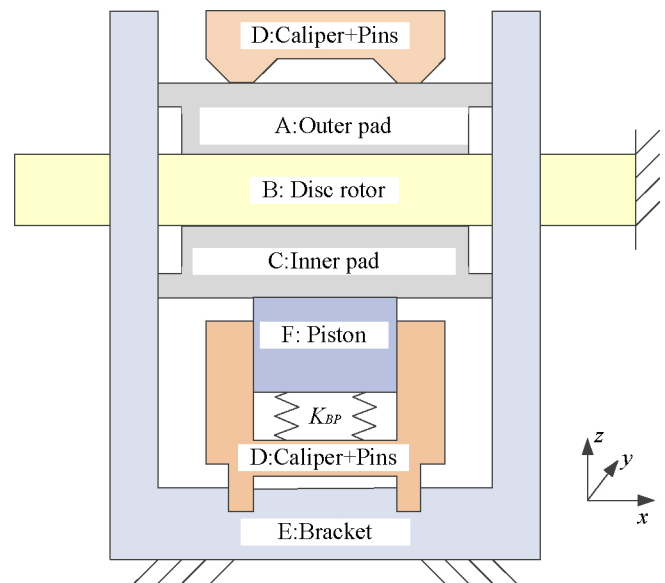


Figure 2. Schematic of new model

In Figure 2, the coupling node pairs are tightly connected with each other in normal directions of most coupling interfaces. So the nodal displacements of relevant node pairs at coupling interface are identical. It should be noted that a spring with stiffness K_{BP} is placed between the head of the piston and the inner bottom surface of piston housing of caliper to simulate the elasticity of the hydraulic pipeline and oil. Table 1 illustrates the coupling relationships on all coupling surfaces.

Table 1. Coupling relationships between substructures

Substructure 1	Substructure 2	Normal direction	Description of coupling form
A	B	z	Tightly contacted in normal direction
C	B	z	In-plane friction caused by relative movement
A	D	z	Tightly contacted in normal direction
A	E	x, y	Free sliding along coupling surfaces
C	E	x, y	
C	F	z	
D	E	x, y	
D	F(piston wall)	x, y	
D	F(piston head)	z	Spring coupling in normal direction
			Free sliding along coupling surfaces

Figure 3 shows the coupling forces from other substructures during braking.

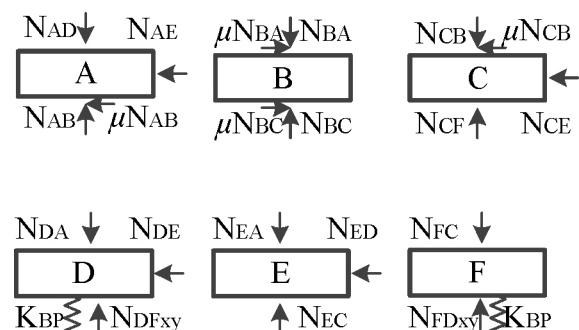


Figure 3. Coupling forces from other substructures