

On the Effect of Friction Law in Closed-Loop Coupling Disc Brake Model

Yongchang Du, Yingping Lv, and Yujian Wang
Tsinghua Univ.

Pu Gao
Beijing Institute of Technology

ABSTRACT

Brake squeal is a complex dynamics instability issue for automobile industry. Closed-loop coupling model deals with brake squeal from a perspective of structural instability. Friction characteristics between pads and disc rotor play important roles. In this paper, a closed-loop coupling model which incorporates negative friction-velocity slope is presented. Different from other existing models where the interface nodes are coupled through assumed springs, they are connected directly in the presented model. Negative friction slope is taken into account. Relationship between nodes' frictional forces, relative speeds and brake pressure under equilibrant sliding and vibrating states is analysed. Then repeated nodal coordinate elimination and substructures' modal coordinate space transformation of system dynamic equation are performed. It shows that the negative friction slope leads to negative damping items in dynamic equation of system. Then the complex eigenvalue analysis is applied to find out the unstable modes of the system. Finally, the presented method is applied on a disc brake system to study the effect of friction character on the stability of the system. Prediction results from models with/without considering negative friction slope effect is listed and compared with bench test results. It shows that with the introduction of negative friction slope, more unstable modes which correspond to test result can be obtained. So the model prediction results are more correlated with those from test. The accuracy of model prediction results is improved and the closed-loop coupling model is more close to the actual brake system.

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INTRODUCTION

Brake squeal is distracting, annoying and affects physical as well as mental health. Large amount of theoretical researches and engineering practices have been carried out on brake squeal issue for decades. Although there are still disputes between researchers over the cause of brake squeal[1,2], agreements regarding the squeal diagnosis, analysis as well as suppression methods have been reached and various analysis models for brake squeal have been developed[3], among which closed-loop coupling model is one of the most popular and effective models for brake squeal analysis[4, 5, 6, 7, 8, 9, 10, 11]. Based on this model, analysis methods, such as substructure modal composition analysis, sensitive analysis and feed-in energy analysis are developed. With these methods, further comprehensive studies on brake squeal from the point of view of structure design of the whole brake system is performed and squeal suppression measures as well as structure modification strategies are also made. This method has been demonstrated in practice as a good way to solve brake squeal issues[12, 13, 14, 15].

In most of current closed-loop coupling models[4-5,8-9], coupling node pairs on coupling surfaces are connected with each other through imaginary coupling springs. Therefore, the stiffness of coupling springs is a critical parameter to the model. Unfortunately,

there is currently neither rigorous explanation for the physical meaning of coupling springs nor systematic method on the determination of the coupling springs stiffness. To avoid these problems, the author of this paper built a new disc brake squeal closed-loop coupling model, in which relevant node pairs on coupling surfaces are assumed to remain in tight contact in normal direction of the interface during the vibration[16]. By comparing the prediction result from theoretical case study with bench test result, the validity and effectiveness of this new model is proved.

Besides the structural factors of the brake system, friction characteristics and working conditions also play important roles in brake squeal. Usually, the friction coefficient has two significant characters: 1) Static friction coefficient is greater than dynamic friction coefficient. 2) In certain area, friction coefficient decreases with growing relative velocity. The former may lead to stick-slip in system while the latter, namely friction coefficient negative-slope character, may result in negative damping effect in system. When the damping of the brake system itself is not great enough, the vibration of the system may be diverged and brake squeal be stimulated due to negative damping effect[17]. A number of researchers[18, 19, 20, 21] consider negative friction-velocity slope character as an important factor when dealing with brake squeal issues.

In a previous preliminary study on the new closed-loop coupling model, coulomb's friction law was used when dealing with friction forces between disc rotor and brake pads[16]. The friction coefficient remains constant during the vibration and the effect of relative sliding speed on friction coefficient is neglected. In this paper, to further explore this model, negative friction-velocity slope character is taken into account. Through force analysis under equilibrium sliding and vibrating states, the negative-slope character of friction coefficient is introduced into the dynamic equation of system. Complex eigenvalue analysis method is then adopted to find unstable modes of the system. Finally, a case study on a disc brake with squeal propensity is performed. Prediction results from models with/without considering negative friction-velocity slope effect is calculated and compared with bench test results. It shows that due to the introduction of negative-slope character, more unstable modes which correspond to the result from bench test can be obtained. It has demonstrated that with the negative friction-velocity slope character, the accuracy of model prediction result is improved and the closed-loop coupling model is more close to the actual brake system.

CLOSED-LOOP COUPLING MODEL

A disc brake closed-loop coupling model is presented in a former study[16]. It is assumed that during the vibration of the brake system, the relevant coupling nodes on coupling surfaces remain in tight contact in normal direction of the interface. In this way, the dynamic equation of the system is derived. And by performing coordinate reduction, modal coordinate transformation as well as complex eigenvalue analysis, the unstable modes of the system are obtained. The model structure and substructures involved are shown in Figure 1.

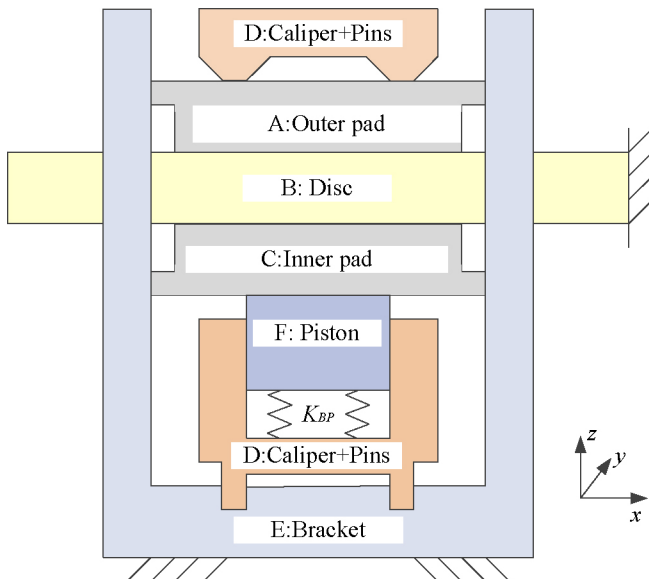


Figure 1. Model structure and substructures

Neglecting the damping of substructures, the dynamic equations of each substructure can be written as:

$$\begin{aligned} [M_A]\{\ddot{u}_A\} + [K_A]\{u_A\} &= \{F_A\} \\ [M_B]\{\ddot{u}_B\} + [K_B]\{u_B\} &= \{F_B\} \\ [M_C]\{\ddot{u}_C\} + [K_C]\{u_C\} &= \{F_C\} \\ [M_D]\{\ddot{u}_D\} + [K_D]\{u_D\} &= \{F_D\} \\ [M_E]\{\ddot{u}_E\} + [K_E]\{u_E\} &= \{F_E\} \\ [M_F]\{\ddot{u}_F\} + [K_F]\{u_F\} &= \{F_F\} \end{aligned} \quad (1)$$

where $M_A \sim M_F$ and $K_A \sim K_F$ are the discrete mass and stiffness matrices of substructure A~F respectively. $\{u_A\} \sim \{u_F\}$ and $\{F_A\} \sim \{F_F\}$ are nodal displacement vectors and force vectors respectively.

Group and rearrange the nodes of substructures according to their coupling conditions with other substructures. For instance the nodal displacement vectors are:

$$\begin{aligned} \{u_A\} &= \begin{Bmatrix} u_{A0} \\ u_{ABxy} \\ u_{ABz} \\ u_{AD} \\ u_{AE} \end{Bmatrix} & \{u_B\} &= \begin{Bmatrix} u_{B0} \\ u_{BAxy} \\ u_{BAz} \\ u_{BCxy} \\ u_{BCz} \end{Bmatrix} & \{u_C\} &= \begin{Bmatrix} u_{C0} \\ u_{CBxy} \\ u_{CBz} \\ u_{CE} \\ u_{CF} \end{Bmatrix} \\ \{u_D\} &= \begin{Bmatrix} u_{D0} \\ u_{DA} \\ u_{DE} \\ u_{DFxy} \\ u_{DFz} \end{Bmatrix} & \{u_E\} &= \begin{Bmatrix} u_{E0} \\ u_{EA} \\ u_{EB} \\ u_{ED} \end{Bmatrix} & \{u_F\} &= \begin{Bmatrix} u_{F0} \\ u_{FC} \\ u_{FDxy} \\ u_{FDz} \end{Bmatrix} \end{aligned} \quad (2)$$

First subscript of each vector component represents the substructure which the node belongs to. Second subscript stands for the substructure which the node is coupled with. Subscript 0 means the node is not coupled with other substructure. When necessary, optional subscripts x, y or z are used to indicate the direction of coupling.

Similarly, the dynamic nodal force vectors can be arranged as:

$$\begin{aligned} \{F_A\} &= \begin{Bmatrix} 0 \\ \mu N_{AB} \\ N_{AB} \\ N_{AD} \\ N_{AE} \end{Bmatrix} & \{F_B\} &= \begin{Bmatrix} 0 \\ \mu N_{BA} \\ N_{BA} \\ \mu N_{BC} \\ N_{BC} \end{Bmatrix} & \{F_C\} &= \begin{Bmatrix} 0 \\ \mu N_{CB} \\ N_{CB} \\ N_{CE} \\ N_{CF} \end{Bmatrix} \\ \{F_D\} &= \begin{Bmatrix} 0 \\ N_{DA} \\ N_{DE} \\ N_{DFxy} \\ K_{BP}(u_{FDE} - u_{DFz}) \end{Bmatrix} & \{F_E\} &= \begin{Bmatrix} 0 \\ N_{EA} \\ N_{EB} \\ N_{ED} \end{Bmatrix} & \{F_F\} &= \begin{Bmatrix} 0 \\ N_{FC} \\ N_{FDxy} \\ K_{BP}(u_{DFz} - u_{FDE}) \end{Bmatrix} \end{aligned} \quad (3)$$

where N represents the normal contact force between the substructures (their subscripts have same meaning as nodal displacement u), μ is friction coefficient. K_{BP} represents the equivalent spring coupling effect between piston head and the bottom of piston housing of caliper to simulate the elastic of hydraulic pipeline and oil.