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Sensitivity analysis of brake squeal tendency to substructures' modal parameters

Dihua Guan*, Xindong Su, Fang Zhang

State Key Laboratory of Automotive Safety and Energy Conservation, Tsinghua University, Beijing, PR China

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Abstract

Sensitivity analysis methods are explored to determine the dominant modal parameters of substructures of a brake system for brake squeal suppression analysis. The related formulae of sensitivities of the positive real part of the eigenvalue of squeal mode (RES) to substructures' modal parameters (SMP) are derived. The sensitivity analysis method can determine the dominant modal parameters influencing the squeal occurrence in a regular way, and the dominant modal parameters will be set to be the target of structural modification attempted to eliminate the squeal mode. Sensitivity analysis of a typical squealing disc brake is performed. The analysis results show that a modified rotor or a modified bracket can be used to eliminate the squealing and that the latter is taken in practice and verified experimentally.

1. Introduction

Brake squealing noise is a key element of city environmental pollution and reduced ride comfort. Brake squeal has been an intractable problem in automotive industry for a long time. Replacement of asbestos with the newer materials which have low damping, high thermal resistance for the friction lining and the need for lightweight vehicles are making the problem more severe than ever [1]. So it is necessary as well as profitable to eliminate it.

*Corresponding author. Department of Automotive Engineering, Tsinghua University, Beijing 100084, PR China. Fax: +86 10 62785708.

E-mail address: guandh@mail.tsinghua.edu.cn (D. Guan).

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In recent years, many researchers [2–5] through linear model analysis, consider that the brake system whose characteristic equation has complex eigenvalues with positive real parts has the unstable mode, and will vibrate divergently under an unavoidable initial random disturb, while the noise occurs. Furthermore, Ref. [5] puts forward the idea that the stability of the brake system is determined by the match of the dynamic characteristics of the brake components.

A lot of successful instances indicate that structural modification of the brake components is an effective approach to suppress the brake squeal [3,6]. The approach of Ref. [6], mainly based on experimental investigation, although effective in resolving some practical cases with brake squeal problems, needs a lot of tests in general, and leads to high cost in both time and finance. While, Ref. [3], mainly based on theoretical model analysis, directly derives an effective scheme for suppressing brake squeal, which has been verified by experiment. In the paper, the analysis method of coefficient of substructure's modal contribution is used to determine the dominant substructures' modal parameters (SMP) influencing occurrence of the unstable mode. Furthermore, aiming at the flaw of the method of substructure's modal contribution analysis, Ref. [7] introduces the feed-in energy analysis method, through which the assumption that feed-in energy, as well as the real part of the eigenvalue of squeal mode (RES) can indicate the squeal tendency is verified. Comparing to the method in Ref. [3], the feed-in energy analysis method in Ref. [7] can be used for brake squeal analysis much more precisely.

In this paper, as a different approach to determine the dominant SMP (including the eigenvalue and eigenvector), the sensitivity analysis of the positive RES to the SMP is put forward. Since the positive RES can be used to indicate the tendency of brake squeal occurrence, the sensitivities of the RES to the SMP can disclose the influence of the SMP on tendency of squeal occurrence. Based on this, the formulae of sensitivities of the RES to the SMP are derived. The sensitivity analysis method is then applied to a squealing problem of a typical disc brake, which shows that the sensitivity analysis method can be used to analyze the influences of the SMP on the squeal tendency in a regular, comprehensive way and can be easily integrated into CAE design for brake squeal suppression.

2. The closed-loop coupling model for brake squeal analysis

The characteristic equation of the brake closed-loop coupling model is [3]

$$[M]\{\ddot{u}\} + ([K] - [K_f])\{u\} = 0, \tag{1}$$

where $\{u\}$ denotes the displacement vector of all nodes of the brake FE model, $[M]$ and $[K]$ are the mass and stiffness matrices respectively, $[K_f]$ is the unsymmetric friction coupling stiffness matrix which represents the frictional and elastic coupling relationships between substructure's interface nodes.

Convert the displacement vector $\{u\}$ into the substructure's modal coordinates $\{q\}$

$$\{u\} = [\Phi]\{q\}, \tag{2}$$

where $[\Phi]$ is the matrix composed of respective mass normalized modal shape matrix of substructures.