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Automotive brake squeal analysis with rotating finite elements of asymmetric disc in time

Jaeyoung Kang

Division of Mechanical and Automotive Engineering, College of Engineering, Kongju National University, Cheonan-Si, Republic of Korea

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ABSTRACT

The new finite element brake squeal model is proposed where the finite elements of a real brake disc rotate in time. Contact nodal forces between the rotating disc and stationary pads are allocated to the moving contact area at every time step. When the proposed model is applied to an asymmetric automotive brake disc, it becomes the periodic time-varying brake system. The stability boundary of the discrete time-varying system is numerically calculated by the Floquet theory. Also, the quasi-static linearized eigenvalue analysis is conducted to show that the unstable modes repeatedly appear at the short interval of the disc rotation angle. The results are consistent with the angle-dependent local phenomenon of squeal termed squeal periodicity in the squeal experiment. In the nonlinear time-domain analysis, the squeal vibration increases and then decays in time for the rotating mode shape functions. It demonstrates that the rotation of an asymmetric disc can change the nonlinear squeal behavior as well as the linear stability character drastically.

1. Introduction

Brake squeal is one of the major noise and vibration problems in automotive industry and the related literature has been reviewed in [1–3]. Several mechanisms such as mode-coupling, negative friction-slope and modal damping and gyroscopic effects have been suggested as the cause of squeal in [4–6]. Since a real disc brake system has complex geometry, the disc brake system has been modeled using finite elements (FE). A friction contact model on the contact nodes was developed as the 1-dimensional and 2-dimensional friction model in the FE brake squeal model [7–10]. In automotive industry and academy, the commercial FE software such as ABAQUS is usually implemented for brake squeal simulation [11].

For efficient eigenvalue sensitivity analysis, the brake squeal models mixed with FE geometry and analytical contact kinematics were developed. Huang et al. [12] constructed the FE model of drum brake squeal by the assumed modes method where the normal contact and circumferential friction nodal forces are connected between the stationary drum and shoe. Kang et al. [13] applied this FE modeling approach to the disc brake squeal. The disc rotation in these FE models has been omitted by assuming that rotation effects are negligible for the low speed of the squealing disc.

As mentioned by Ouyang et al. [14–16], brake squeal can be simulated as the moving load problem in which the contact forces between the stationary pad and rotating disc are discretized into finite elements. When the disc rotates, the contact area of a disc and pad instantaneously slides over. On the relative rotation, the disc should rotate past the pads and, thus, different surface area are mating with the pad contact area. The series of Ouyang's articles [14–16] constructed the moving

E-mail address: jkang@kongju.ac.kr

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load model with an analytical thin plate as a rotating disc and the FE model for the stationary parts such as a pad and caliper. In their approach, the disc plate is set to be stationary while the pads rotate around the disc. In the series of Kang's articles [17–19], the velocity vector in the rotating disc is described in the global frame by using the coordinate transformation where the 2-dimensional friction nodal force between the disc and pad is expressed in the global frame through the finite elements of the regular mesh method and mode discretization.

Regarding the solution of the moving load models, the previous models assume the axis-symmetry of the disc. As a result, the system eigenvalues do not change with the disc rotation angle. Although the rotation of the mode shape functions for the disc produces the angle-dependent system matrices, the system eigenvalues as the dynamic characteristics should not be angle-dependent but time-invariant due to the axis-symmetry. For example, the coordinate transformation matrix can be explicitly found for transforming the time-varying equation of motion (8) in [16] into the time-invariant one (20) in [16]. Another example is that the rotating mode shape function can be transformed into the stationary one by the coordinate transformation Eqs. (17)–(19) in [18] without any loss of generality. Therefore, the equations of motion for the axis-symmetric rotating disc can be described to be time-invariant. In conclusion, the system eigenvalues of the axis-symmetric brake squeal model are unique and invariable with respect to the rotation angle regardless of the rotating motion of the disc.

For the perfectly axis-symmetric disc, any pair of doublet modes should have an identical frequency. However, the frequency split between any pair of disc modes is always observed in the automotive brake disc [16,20,21]. It reveals that any real brake disc is not the perfectly axis-symmetric. It was mentioned in [1] that bolt assembly or geometric imperfections destroy the rotational symmetry of the disc, leading to the frequency splitting. For the asymmetric disc, the mode shape functions become distorted in shape and spatially modulated from the axis-symmetric mode shapes [22]. Accordingly, the eigenvalues of the brake system with the asymmetric rotating disc may not be invariable for the variation of the rotation angle. Moreover, the nonlinear dynamic behavior after the onset of squeal may be modulated due to this asymmetric characteristics.

Strictly to say, therefore, the perfectly-symmetric assumption is not valid for the automotive brake disc, and the coordinate transformation does not work under these strict circumstances. Instead, the finite elements of the automotive disc should actually rotate in time. As a result, the stiffness and damping matrices in the automotive brake squeal model should be time-varying and they cannot be transformed into the time-invariant due to the spatially-distorted mode shapes. In the previous literature, the time-varying brake system with the rotation of the asymmetric FE disc has never been studied because it requires to rotate the numerous finite elements of the disc and their modal data in time.

In the brake squeal test [23,24], the modal vibrations of squeal repeatedly occurred at a short interval of rotation angle over one disc revolution, which was termed squeal periodicity. It implies that the instability of squeal modes does not continue over the long-term revolutions of the disc, but arises within a short interval of rotation angle. Therefore, the squeal periodicity is the angle-dependent and short-term phenomenon. However, the invariable eigenvalues of the rotating axis-symmetric disc case as well as the fixed eigenvalues of the stationary disc case seem to contradict with the short-term phenomenon of brake squeal periodicity.

The aim of this study is to develop the FE brake squeal model with the real rotation of the asymmetric brake disc in time, and then, demonstrate the squeal periodicity in the linear stability and nonlinear time-domain analysis. In the present work, the angle-dependent eigenvalues of the rotating asymmetric disc are suggested as one possible cause of squeal periodicity. For this, the FE mode shape vectors of the disc rotate in time at a constant rotating speed and they are periodic per one disc revolution. Consequently, the rotating brake squeal model is described as the periodic time-varying system in the discretized finite element manner. Then, the stability analysis is conducted by means of both the Floquet stability and complex eigenvalue analysis. From the stability results, the interesting new findings about the angle-dependent squeal phenomenon of a rotating asymmetric disc are provided. Nonlinear squeal response in time is also investigated by the time-domain analysis.

2. Derivation of equations of motion

In this study, an actual automotive brake system is modeled in the finite element manner. The FE brake disc is asymmetric and the finite elements of the disc rotate at a constant speed Ω. In turn, the finite element mode shape vectors (or mode shape functions) of the disc attached to the rotating disc also rotate as shown in Fig. 1. From the reference observer fixed to the XYZ frame, the displacement vector of the rotating disc at a globally fixed position $\mathbf{X}$ can be discretized in the modal expansion form such that:

$$\mathbf{u}(\mathbf{X},t) = u(\mathbf{X},t)\mathbf{e}_r + v(\mathbf{X},t)\mathbf{e}_\theta + w(\mathbf{X},t)\mathbf{e}_z \tag{1}$$

$$u(\mathbf{X},t) \cong \sum_{n=1}^{\mathrm{N_d}} \phi_{r,n}(\mathbf{X},t)\cdot q_n(t) \tag{2}$$

$$v(\mathbf{X},t) \cong \sum_{n=1}^{\mathrm{N_d}} \phi_{\theta,n}(\mathbf{X},t)\cdot q_n(t) \tag{3}$$

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