

Modal Based Rotating Disc Model for Disc Brake Squeal

Yongchang Du and Yujian Wang
Tsinghua University

Pu Gao and Yingping Lv

ABSTRACT

Modelling of disc in brake squeal analysis is complicated because of the rotation of disc and the sliding contact between disc and pads. Many analytical or analytical numerical combined modeling methods have been developed considering the disc brake vibration and squeal as a moving load problem. Yet in the most common used complex eigenvalue analysis method, the moving load nature normally has been ignored. In this paper, a new modelling method for rotating disc from the point of view of modal is presented. First finite element model of stationary disc is built and modal parameters are calculated. Then the dynamic response of rotating disc which is excited and observed at spatial fixed positions is studied. The frequency response function is derived through space and time transformations. The equivalent modal parameter is extracted and expressed as the function of rotation speed and original stationary status modal parameters. Because of rotating, each mode of the disc split to two complex modes. Finally, response of rotating disc under fixed constant force is studied using the presented equivalent modal parameters. It shows that the response consists of a static response and two travelling waves with frequencies differ from the stationary modal frequency.

CITATION: Du, Y., Wang, Y., Gao, P., and Lv, Y., "Modal Based Rotating Disc Model for Disc Brake Squeal," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(1):2015, doi:10.4271/2015-01-0665.

INTRODUCTION

Brake squeal has been one of the most elusive problems for automotive industry since the early 20th century. Squeal is a sustained, high frequency ($>1000\text{Hz}$) vibration of brake system components during a braking action resulting in noise audible to vehicle occupants or passers-by. A number of theories have been formulated to explain the mechanisms of brake squeal, and numerous studies have tried with varied success to apply them to the dynamics of disc brakes [1].

Many models, including experimental, analytical and numerical models, have been presented for analyzing brakes squeal. Numerical methods for simulation and analysis of brake squeal using FE (finite element) model can be categorized into two types, namely, complex eigenvalue analysis and dynamic transient analysis [2]. Currently the complex eigenvalue method is preferred and widely used in predicting the squeal propensity of the brake system including damping and contact. For disc brake squeals, Liles [3] was the first person who utilized complex eigenvalue analysis using FE method to predict disk squeal propensity. Guan [4] constructed FE models for all the brake components, built closed-loop coupling disc brake model and developed a series of analysis methods such as substructure (component) modal participation analysis, substructure modal parameter sensitivity analysis and feed-in energy analysis [4, 5, 6, 7, 8]. Similar approaches were adopted by other researchers, i.e. Nack [9], Yuan [10], Kung [11], Goto [12] et al.

Modelling of disc brake for brake squeal analysis is complicated by the fact that the disc is rotating and the pads and caliper are fixed and the nature of the sliding contact between pads and disc. Since there is relative rotation between the disc and the pads, this is a typical moving load problem. For simple moving load problems, analytical methods may be used to obtain closed-form solutions. Complex moving load problems, on the other hand, have to be dealt with by numerical methods such as the finite element method [13]. In this case, since relative moving exists in the system, a finite element analysis will have to be carried out at each time step. To make matter worse, the length of the time step has to be very small as high frequencies of the whole system are often involved. As a result, a large amount of computation is needed [14].

Ouyang, Cao, etc [15, 16] proposed a method to treat the disc vibration and squeal as a moving load problem and put forward an analytical-numerical combined method. In their works, the disc brake was conceptually divided into two parts: the rotating disc, which was approximated as an annular, thin plate and solved by an analytic method, and nonrotating components, which were solved by FE method. Then a nonlinear eigenvalue problem was constructed and solved by a search approach. By varying each system parameter of interest, the instability of the whole brake system about a specific parameter was found by examining the real parts of the eigenvalues. Through numerical simulation they concluded that the original double eigenvalues of the disc split into pairs of distinct eigenvalues which are in conjugate pairs. The eigenvalues with greater frequencies had positive real parts. The differences between the two

distinct frequencies for each pair of eigenvalues were larger for increasing rotating speed of the disc in the low speed range. The instability of the system becomes more severe with increasing rotating speed.

Historically, in most brake models for complex eigenvalue analysis, the moving load nature normally has been ignored. For example, in closed-loop coupling model [4, 5], the dynamic behavior of disc is represented by its stationary modal parameters. The effect of rotating is only embodied in the relentlessly one-direction friction forces on pads / disc interfaces. So these models are not suitable for those cases when squeal properties under different rotation speed are the main concern.

This paper aims at the modal solution of disc rotor concerning rotation. Modal parameters of stationary disc are calculated from FE model. The rotating disc is supposed to be excited and corresponding responses to be observed at spatial fixed points which are not rotating with the disc. The characteristics of equivalent modal parameters, which represent the dynamic properties of rotating disc suffering moving loads, will be studied. Response of the rotating disc under fixed constant force is studied using the presented equivalent modal parameters. It can be seen that the response consists of static response and two travelling waves which frequencies differ from the stationary modal frequency.

FE MODEL AND MODAL PARAMETERS OF THE STATIONERY DISC

The target disc is from a passenger car's front brake. FE model of the disc is constructed, as illustrated in [Figure 1](#). There are about 8500 elements in the model. Some very small features such as tiny chamfers are not included in the model. By adjusting the material's density, the masses of FE model and the real disc reach identical.

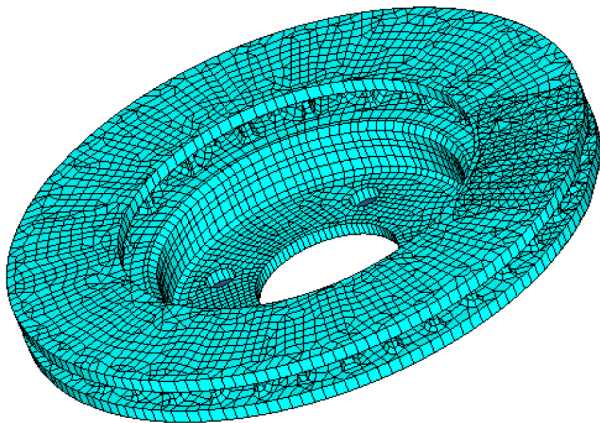


Figure 1. FE model of the disc

Experimental modal analysis and FE modal analysis of the disc under free-free boundary condition are carried out to validate the developed model. Natural frequencies and mode shapes obtained in the experiments are used to compare with the finite element results. The disc has various mode shapes. Only OPD (out-of-plane diametric) type modes are used for comparison because the responses of these modes in axial direction are remarkably large and easy to measure in

the test. Through a tuning process of the material properties of the model, such as Young's modulus, predicted natural frequencies are brought closer to the experimental data, as shown in [Table 1](#).

Table 1. Modal results of the disc at free-free boundary condition

Modal shape	FE (Hz)	Test (Hz)	Error (%)
2 nd OPD	1215.1	1210.6	0.372
3 rd OPD	2791.2	2760.3	1.119
4 th OPD	4469.4	4407.1	1.414
5 th OPD	6182.2	6128.4	0.878
6 th OPD	7896.4	7848.8	0.606
7 th OPD	9604.0	9675.3	-0.737

After the validation, proper constraints are applied to FE model and an FEA modal analysis is performed to obtain modal parameters to be used for squeal analysis. The analysis frequency range is set up to 20 kHz. In this frequency range, 32 distinct modes are found. These modes include out-of-plane (OP) modes and in-plane (IP) modes. Listed in [Table 2](#) are three types of OP modes. They are the diametric OPD, or (0, n), modes; the disc twisting OPT, or (1, n), modes; and the umbrella OPU, or (n, 0), modes. Listed in [Table 3](#) are two types of IP modes. They are the circumferential IPC and the radial IPR modes.

Table 2. Disc OP modes

Mode	Name	Freq (Hz)	Mode	Name	Freq (Hz)
(0, 0)	0th OPD	1762	(1, 1)	1st OPT	7477
(0, 1)	1st OPD	1300	(1, 2)	2nd OPT	8438
(0, 2)	2nd OPD	1471	(1, 3)	3rd OPT	9842
(0, 3)	3rd OPD	2835	(1, 4)	4th OPT	11617
(0, 4)	4th OPD	4491	(1, 5)	5th OPT	13569
(0, 5)	5th OPD	6193	(1, 6)	6th OPT	15726
(0, 6)	6th OPD	7903	(1, 7)	7th OPT	17941
(0, 7)	7th OPD	9649			
(0, 8)	8th OPD	11411			
(0, 9)	9th OPD	13241			
(0, 10)	10th OPD	15092			
(0, 11)	11th OPD	16967			
(0, 12)	12th OPD	18977	(1, 0)	1st OPU	7554

Table 3. Disc IP modes

Mode	Name	Freq (Hz)	Mode	Name	Freq (Hz)
1	0th IPR	6689	1	0th IPC	3469
2	1st IPR	4110	2	1st IPC	8813
3	2nd IPR	5165	3	2nd IPC	13047
4	3rd IPR	7529	4	3rd IPC	18011
5	4th IPR	10786			
6	5th IPR	14488			
7	6th IPR	18266			

Carefully examining the graphical patterns of modes listed in [Table 2](#) and [Table 3](#), we find that if the differences of vibrating directions are ignored, the IPR and IPC modes are similar to OPD modes. The OPU mode can be considered as a special case of OPT. Two types of graphical patterns are shown in [Figure 2](#).

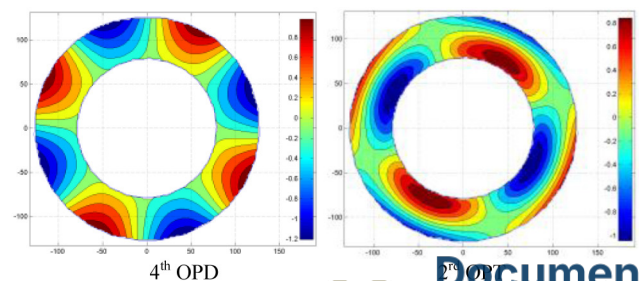


Figure 2. Typical graphical patterns of mode shapes of the disc