GRANT journal
ISSN 1805-062X, 1805-0638 (online), ETTN 072-11-00002-09-4

EUROPEAN GRANT PROJECTS | RESULTS | RESEARCH & DEVELOPMENT | SCIENCE

Analysis stability of brake related to squeal by finite element method

Huynh Le Hong Thai[1]
Phan Thanh Nhan[2]

[1]Technical University of Liberec; Studentská 1402/2, Liberec, Czech Republic; huynhlehongthai@yahoo.com
[2]Technical University of Liberec; Studentská 1402/2, Liberec, Czech Republic; nhancohoc@yahoo.com

Grant: TUL c.2826/115
Name of grant: Student grant competition.
Category: Machinery and Tools

Abstract Brake squeal is defined as a phenomenon of dynamic instability that occurs at one or more of the natural frequencies of the brake system. When brake is unstable lead to vibration of structure and occurs noise. The main of noise caused by friction material, the couple of pads and rotor, beside have more components of brake system like caliper, anchor bracket, knuckle and suspension. Pads and rotor coupling has major impact on mid to high frequency. This paper shows the the a method to identify brake squeal by finite element menthod (FEM), It base on solve complex eigenvalue and find out the real part and imagrine part. The purpose of the analysis presented here is to demonstrate the methodology by identifying any potentially unstable modes in a particular disc brake system.

Keywords Brake squeal, complex eigenvalue, stablity, disc brake

1. INTRODUCTION

Brake squeal has been studied since 1930s until now but unfortunately, the large body of research into brake squeal has failed to provide a complete understanding the ability to predict its occurrence [1]. With the advances in the computer technology recent years, more complicated and complete FE models can be easily built as well as quick turn-around in simulation time. Advancement in the contact formulation and algorithm much help engineers and researchers to obtain more reliable and accurate representation of precdiction stability of brake squeal. This parameter is significantly important either for the complex eigenvalue analysis or the dynamic transient analysis. In recent years, the complex eigenvalue becomes the most preferred method in the brake research community to study brake squeal than the transient analysis [2]. A simple mathematical model of a disc brake is described to form a basic formula of kinetic friction coefficient. The complex eigenvalue analysis is performed for different friction characteristics including friction damping for the real contact interface model and the perfect contact interface model of the friction material.

An essential stage in studying disc brake squeal using finite element (FE) method is the development and validation of the disc brake FE model. It has been thought that an accurate representation of a disc brake (geometry and material properties) and validated FE model would later produce reliable and accurate results. In current practice in developing an FE model most researchers make simplifications and assume that the pad and disc interface is a perfect plane surface [3]. However, it is already known that most contact interfaces have rough and irregular surfaces. Although this simplification may ease the model development and reduce modelling time, it perhaps can reduce the accuracy of predicted results, particularly in the contact analysis and subsequently complex eigenvalue analysis. This section focuses on prediction of unstable frequencies using the complex eigenvalue analysis and describes disc brake vibration characteristics at a system level. It is important that validations of the prediction results are not only based on the unstable frequency alone but also in terms of its unstable mode shapes.

2. COMPLEX EIGENVALUE ANALYSIS

There are many important areas of structural analysis in which it is essential to be able to extract the eigenvalues of the system and, hence, obtain its natural frequencies of vibration or investigate possible bifurcations that may be associated with kinematic instabilities.

The mathematical eigenvalue problem is a classical field of study, and much work has been devoted to providing eigenvalue extraction methods. For many important cases the matrices are symmetric. The eigenvalue problem for natural modes of small vibration of a finite element model is

$$M\ddot{x} + C\dot{x} + Kx = 0 \tag{1}$$

The governing equation can be rewritten in classical matrix notation as

$$\left(\mu^2[M] + \mu[C] + [K]\right)\{\phi\} = 0 \tag{2}$$

Where [M] is the mass matrix, [C] is the damping matrix, which includes friction induced contributions, and [K] is the stiffness matrix, which is unsymmetric due to friction which μ is eigenvalue, Φ is eigenvector. The eigensystem (Eq.2) in general will have complex eigenvalues and eigenvectors. This system can be symmetrized by assuming that is symmetric and by neglecting [C] during eigenvalue extraction. The symmetrized system has real squared eigenvalues, μ^2, and real eigenvectors only.

Typically, for symmetric eigenproblems we will also assume that [K] is positive semidefinite. In this case μ becomes an imaginary eigenvalue, μ=iω, where is the circular frequency, and the eigenvalue problem can be written as

$$\left(-\omega^2[M]+[K]\right)\{\phi\}=0 \tag{3}$$

To solve eigenvalue extraction for symmetric systems (Eq.3) we can use subspace iteration, Lanczos or Householder with quarter rotation methods. In this paper presents complex eigenvalue extraction by subspace projection method.

In the subspace projection method the original eigensystem (Eq. 2) is projected onto a subspace spanned by the eigenvectors of the undamped, symmetric system (Eq. 3). Thus, the symmetrized eigenproblem must be solved prior to the complex eigenvalue extraction procedure to create the subspace onto which the original system will be projected. Next, the original mass, damping, and stiffness matrices are projected onto the subspace of N eigenvectors [4]:

$$\begin{aligned}\left[M^*\right]&=[\phi_1,...,\phi_N]^T[M][\phi_1,...,\phi_N]\\ \left[C^*\right]&=[\phi_1,...,\phi_N]^T[C][\phi_1,...,\phi_N]\\ \left[K^*\right]&=[\phi_1,...,\phi_N]^T[K][\phi_1,...,\phi_N]\end{aligned} \tag{4}$$

Then, the projected complex eigenproblem become

$$\left(\mu^2\left[M^*\right]+\mu\left[C^*\right]+\left[K^*\right]\right)\{\phi^*\}=0 \tag{5}$$

Finally the complex eigenvectors of original system can be obtained by

$$\{\phi\}=[\phi_1,...,\phi_1]^T\{\phi^*\} \tag{6}$$

Complex eigenvalue μ as expressed as $\mu = \alpha \pm i\omega$ where α is real part of μ, Re(μ) indicating the stability of the system, and ω is the imaginary part of μ, Im(μ) indicating the mode frequency. The generalized displacement of the disc system, x, can then be expressed as

$$x = Ae^{\mu t} = e^{\alpha t}(A_1\cos\omega t + A_2\sin\omega t) \tag{7}$$

This analysis determines the stability of the system. When the system is unstable, α becomes positive and squeal noise occurs. An extra term, damping ratio, is defined as $-\alpha/(\pi|\omega|)$. If the damping ratio is negative, the system becomes unstable, and vice versa. The main aim of this analysis is to reduce the damping ratio of the dominant unstable modes [5].

3. RESULT AND CONCLUSION

In this study, only consider disc brake system that rotates about the axis of a wheel, a caliper piston assembly where the piston slides inside the caliper that is mounted to the vehicle suspension system, and a pair of brake pads. When hydraulic pressure is applied, the piston is pushed forward to press the inner pad against the disc, and simultaneously the outer pad is pressed by the caliper against the disc [6]. Figure 1 shows model disc brake and meshed element model of the car front brake under consideration.

Fig 1. Disc brake system model (left) and meshed (right)

Setting condition boundary will be based on the disc brake and pad configuration. Identification of correct boundary conditions determines the accuracy of the FE analysis results. Hence more emphasis is given to this part of the analysis. The disc is completely fixed at the ten counter-bolt holes and the ears of the pads are constrained to allow only move in the direction of the force apply on pads. The model was build and calculated on Abaqus software. When the real part of a complex eigenvalue becomes positive for a given set of operational parameters, the corresponding mode becomes unstable and has the potential to radiate squeal noise. An unstable mode is, thus, referred to as a squeal mode like figure 2 and figure 3.

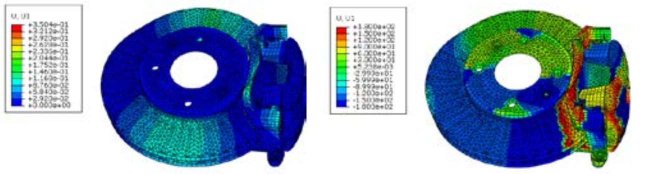

Fig 2: Mode shape (left) and phase angle (right) of the unstable modes 37th at 3,39kHz

The accuracy of the complex mode calculation depends strongly on the accuracy of the surface pressure distribution between the pad and the disc. There are 106 modes in frequences range 12KHz.

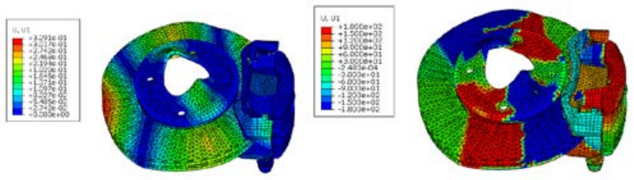

Fig 3. Mode shape (left) and phase angle (right) of the unstable modes 86th at 7,54kHz

The complex modes analysis determines the complex eigenvalues based on the real frequency spectrum calculated in Eq 6. Figure 4, 5 are expressed eigenfrequence and eigenvalue, real part and stabilization dissipation for whole model.

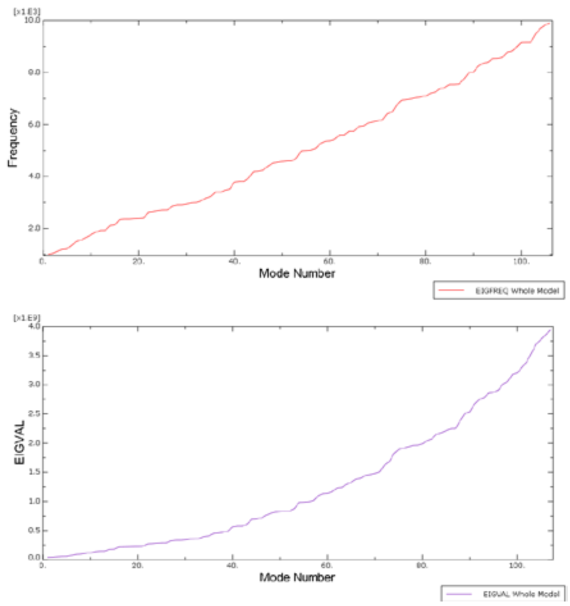

Fig 4. Egienfrequence (above) and eigenvalue (below) for whole model

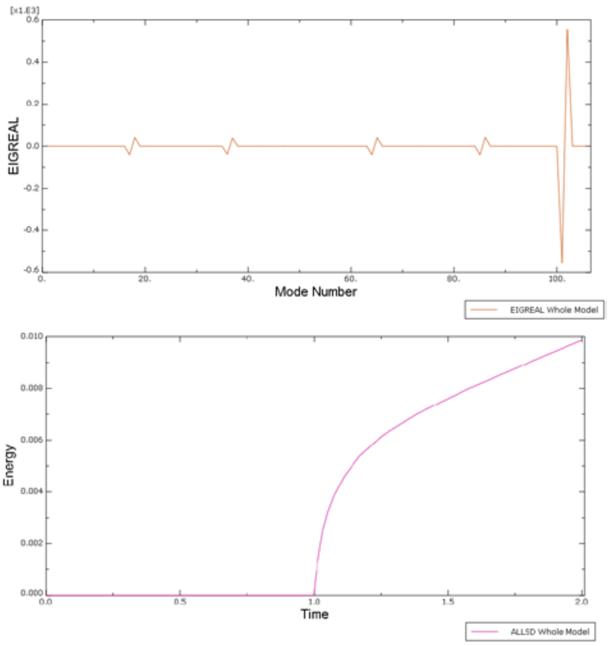

Fig 5. Real part values (above) and static dissipation values (below) for whole model

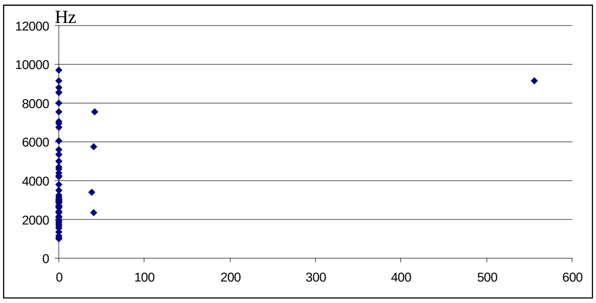

Fig 6. Real part values in different frequencies.

From the simulation, it shows that at frequency that positive real parts (figure 6), disc brake becomes unstable and indicate tendency of squeal to occur at frequencies as 2362, 3396, 5742, 7540, 9154 (Hz). It is also suggested that most unstable frequencies and its mode shapes of the disc are quite close to natural frequencies and normal mode shapes of disc in free-free boundary condition.

Advantage of the complex eigenvalue analysis was utilized to predict unstable frequencies and modes. The result shown stress occurs on disc surface and contact pressure also modal vibration of disc brake.

Sources

1. Papinniemi, A., et al., *Brake squeal: a literature review.* Applied Acoustics, 2002. 63(4): p. 391-400.
2. Thai, H.L.H., *Methods analysis vibration and noise of car disc brake*, in ICoVP. 2011: PRAGUE, Czech Republic.
3. Bakar, A.R.A., *Modelling and Simulation of Disc Brake Contact Analysis and Squeal,* in Mechanical Engineering. 2005, University of Liverpool: Liverpool, United Kingdom.
4. Abaqus, Abaqus/CAE User's manual, version 10. V10.
5. Liu, P., et al., *Analysis of disc brake squeal using the complex eigenvalue method.* Applied Acoustics, 2007. 68(6): p. 603-615.
6. Thai, H.L.H., N. PAVEL, and P.T. NHAN, *Identify source vibration and noise of disc brake*, IRMES. 2011: Zlatibor, Republic of Serbia.
7. Himanshu Misra, Wayne Nack, Tom Kowalski, Louis Komzsik, Erwin Johnson: *Brake Analysis and NVH Optimization Using MSC.NASTRAN*.
8. James Walker Jr: *High-Performance Brake Systems*: Design, Selection, and Installation (S-A Design), 2007.
9. A . R. Abu Bakar, H. Ouyang and D. Titeica: *Modelling and simulation of disc brake contact analysis and squeal,* 2005, Seminar on Advances Malaysian Noise, Vibration and Comfort.