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On the influence of multiple contact conditions on brake squeal

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Abstract

This study focuses on squeal noise prediction for an automotive brake system. For this purpose, a stability study of a finite element model of the brake system is carried out. For the determination of the squeal propensity of a brake system via finite element models, the commonly used approach consists in considering only a friction coefficient at the pad-disc interface. However, numerous other contacts exist in a brake system. In the present study, the influence of several contacts between the caliper, the bracket, the pad and the piston is studied. It turns out that the consideration of these numerous contacts has a real impact on the stability results and can not therefore be neglected. Indeed, a high dispersion of results for the system's eigenvalues indicating strong modifications of the stability behavior and thus of the squeal propensity is observed when different contact conditions are considered. This study insights the necessity to take into account of all contact conditions during the design process of brake systems.

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Keywords: Brake squeal; Finite element model; Multi-contact; Complex Eigenvalue Analysis; Stability

1. Introduction

Even if the squeal noise has no effect on the efficiency of braking, the resulting self-excited vibrations are undesirable since they can be the source of noise nuisance for passengers and so represent an important cost to the customer service. In this context, many studies have been dedicated to the squeal noise prediction [1–10]. In an industrial context, the stability analysis and the well-known Complex Eigenvalues Analysis (CEA) are mostly used to detect the onset of instability and to establish the squeal propensity [11].

Until now, the majority of numerical simulations shows the predominant influence of the pad-disc interface on the brake system stability. However, many contact interfaces may be present in an automotive brake system. Nowadays, the influence of the state of these contacts on the stability behaviour of the brake system and thus on the squeal propensity is not yet well understood. Thereby, the main contribution of the present study aims to take into account these different interfaces in the process of the prediction of squeal noise and to investigate the sensitivity of the squeal propensity to the various contact interfaces. From a finite element model of an industrial automotive brake, effects of friction conditions at each contact interfaces are analyzed via CEA. For that purpose, nine interfaces characterizing

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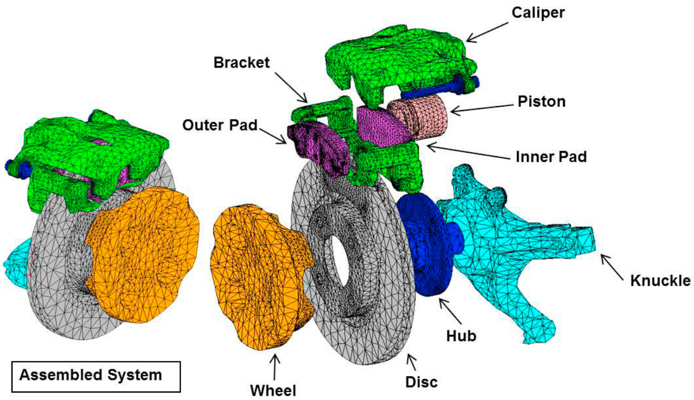

Fig. 1: Finite element model under study - Assembled view (left) and exploded view (right)

contacts between pads and the other elements of the brake system are retained in addition to the classical pad-disc interface.

The paper is organized as follows. Firstly, the brake system under study and the classical CEA are briefly described. Secondly, numerical results considering the effect of the friction coefficient between the pad and the disc as well as other potential frictional interfaces are presented.

2. Automotive braking system under study

2.1. Finite element model

The system under study is an automotive brake system using a floating caliper technology. The caliper that maintains the two pads can slide in parallel to the rotation axis of the disc. During braking, an hydraulic pressure is applied on the piston which pushes the inner pad until contact with the disc. Then, reaction forces push the body of the caliper and the outer pad against the opposite face of the disc. The finite element model of the brake system under study is shown in Fig. 1. For more details see [12]. This model was developed using Abaqus software. The pads are free of movement in the caliper and can be in interaction with different components of the brake system through one or several contact interfaces. In the present study, three contact interactions are taken into account: piston/pad, bracket/pad and caliper/pad. The state of each contact interface is directly linked to the position of pads. Showing the finite element model under study, nine possible interfaces are considered, as illustrated in Fig. 2. The associated parameters are given in Table 1. If a state of contact is modified (in particular if the contact is considered as sliding or not), a new static equilibrium of the brake system may appear which then leads to a new potential state of the brake system with respect to its stability. All frictional interfaces are characterized by a classical Coulomb's law with a constant friction coefficient for each interface. The detachment (i.e. switching from a contact state to a non-contact state at the selected interfaces) is allowed.

The general non-linear dynamic equation of the brake system can be written as:

$$\mathbf{M}\ddot{\mathbf{X}} + \mathbf{C}\dot{\mathbf{X}} + \mathbf{K}\mathbf{X} + \mathbf{F}_{nl}(\mathbf{X}) = \mathbf{F}_{ext} \tag{1}$$

where $\mathbf{X}$, $\dot{\mathbf{X}}$ and $\ddot{\mathbf{X}}$ are the displacement, velocity and acceleration vectors, respectively. $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are the mass, damping and stiffness matrices of the system, respectively. $\mathbf{F}_{ext}$ represents the external forces (i.e. the pressure applied on the piston and the caliper). $\mathbf{F}_{nl}$ is related to the contributions from both the contact nonlinear forces and frictional forces of each interface.

2.2. Stability analysis

In order to determine the squeal propensity of the brake system, the classical CEA is performed on the linearized system around its equilibrium position. The origin of the instabilities of the brake system is due to the presence of

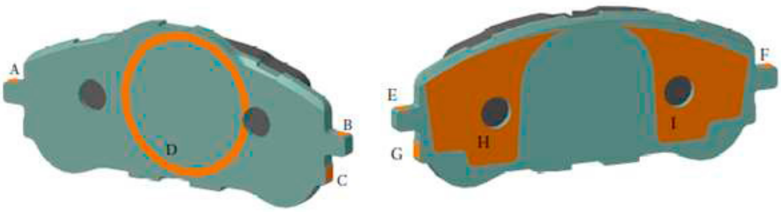

Fig. 2: Contact Interfaces of internal (left) and external (right) pads

Table 1: Contact parameters

Interface letter	Type of contact	Parameter
A	Inner Top Radial Contact	ITR
B	Inner Bottom Radial Contact	IBR
C	Inner Bracket-Pad Contact	IBP
D	Pad-Piston	PP
E	Outer Bottom Radial Contact	OBR
F	Outer Top Radial Contact	OTR
G	Outer Bracket-Pad Contact	OBP
H	Bottom Caliper-Pad Contact	BP
I	Top Caliper-Pad Contact	TP

the friction forces that can add non-symmetrical terms in matrices of the system. The different steps of the stability analysis are presented below.

The first step consists in the determination of the non-linear static equilibrium $\mathbf{U}_S$ by solving:

$$\mathbf{K}\mathbf{U}_S + \mathbf{F}_{nl}(\mathbf{U}_S) = \mathbf{F}_{ext} \tag{2}$$

Then the system is linearized around its non-linear static equilibrium. It can be written in the following way:

$$\mathbf{M}\ddot{\overline{\mathbf{X}}} + \mathbf{C}\dot{\overline{\mathbf{X}}} + (\mathbf{K} + \mathbf{J}_{nl})\,\overline{\mathbf{X}} = \mathbf{0} \tag{3}$$

where $\overline{X}$ defines the perturbation around the equilibrium point (i.e. $\mathbf{X} = \mathbf{U}_S + \overline{\mathbf{X}}$). $\mathbf{J}_{nl}$ is the linearized expression of the non-linear forces $\mathbf{F}_{nl}$ around the static equilibrium $\mathbf{U}_S$. The associated eigenvalue problem is solved with the equation:

$$(\lambda^2\mathbf{M} + \lambda\mathbf{C} + (\mathbf{K} + \mathbf{J}_{nl}))\mathbf{\Phi} = 0 \tag{4}$$

As previously explained the contribution $\mathbf{J}_{nl}$ is unsymmetric due to friction. So eigenvalues λ_j and eigenvectors $\mathbf{\Phi}_j$ are complex. Each λ_j can be written as $\lambda_j = a_j + i\omega_j$ where ω_j is the pulsation associated to the mode $\mathbf{\Phi}_j$ and a_j is the real part. The system is considered as unstable if at least one eigenvalue has a positive real part.

3. Numerical results

3.1. Reference case: influence of the friction coefficient at the pad/disc interface

First of all, a reference case is investigated: CEA is computed for different values of the friction coefficient $\mu_{pad-disc}$ at the pad-disc interface whereas all other contact states are considered to be identical (contact with a friction coefficient equal to 0.15). The friction coefficient $\mu_{pad-disc}$ varies from 0.05 to 0.95 with a step of 0.1. The results on the stability analysis for the industrial representative brake system under study are given in Fig. 3

When the friction coefficient $\mu_{pad-disc}$ increases, a mode coupling phenomenon may appear. The real part of an eigenvalue becomes positive and the associated mode is unstable. In this present case, six unstable modes are observed

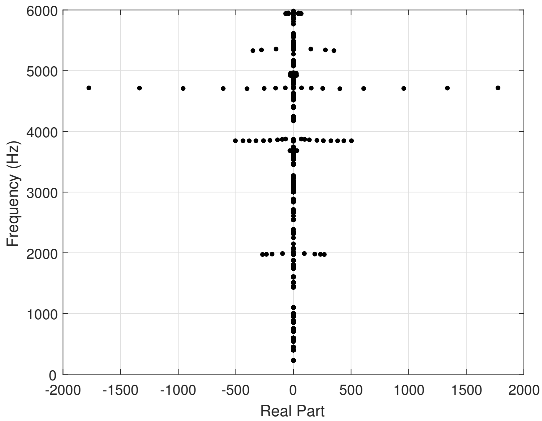

Fig. 3: Evolution of eigenvalues of the brake system assuming a variation of the friction coefficient at the interface pad-disc

Table 2: Characteristics of unstable modes

Number of the unstable mode	Frequency (Hz)	$\mu_{0,pad-disc}$
1	316, 4	0.65
2	616.6	0.15
3	750.7	0.25
4	789.4	0.25
5	852.6	0.75
6	947.2	0.55

in the frequency range of interest and in the friction range of study. Table 2 gives the value of the coefficient at the Hopf bifurcation point $\mu_{0,pad-disc}$ for each unstable mode (i.e. the value of $\mu_{pad-disc}$ for which the real part of the eigenvalue becomes positive). In conclusion, the influence and the major role of the coefficient of friction $\mu_{pad-disc}$ on the stability of the system is highly visible.

3.2. Influence of the friction coefficient at different interfaces

In order to identify the influence of the friction coefficient at the different contact interfaces, a second study is performed in order to observe the evolution of the system stability versus the friction coefficient. For that purpose, the friction coefficient of the pad-disc interface $\mu_{pad-disc}$ is fixed. All other contacts presented in Fig. 2 and Table 1 are characterized by a single friction coefficient μ that varies from 0 to 1. For each value of μ, a CEA is performed. Results obtained for three different values of $\mu_{pad-disc}$ are displayed in Fig. 4. The influence of the friction coefficient μ is obvious. Two groups are distinguishable: the first one corresponds to the sliding contact conditions (red points in Fig. 4) whereas the second group is composed of the frictional contacts conditions.

3.3. Influence of contact conditions on the stability

This last part of the study is focused on the influence of pads position on the squeal propensity, namely the influence of contact conditions on the system stability. For that purpose, two contact conditions are considered for the nine considered interfaces: the first case concerns a friction coefficient equal 0 (i.e. to a sliding case with $\mu = 0$), while the second case is about a frictional contact with $\mu = 0.15$. It is worth noting that contacts with a non zero frictional coefficient are not necessarily in a sticking state after static analysis. However, they as considered as such during the linearization procedure in the computation of the eigenvalues. The friction coefficient $\mu_{pad-disc}$ at the pad-disc interface is fixed to 0.5. This represents a total of 512 configurations. For each configuration, a stability analysis is realized. The superimposition of all complex eigenvalues computed is displayed in Fig. 5. If some groups of unstable

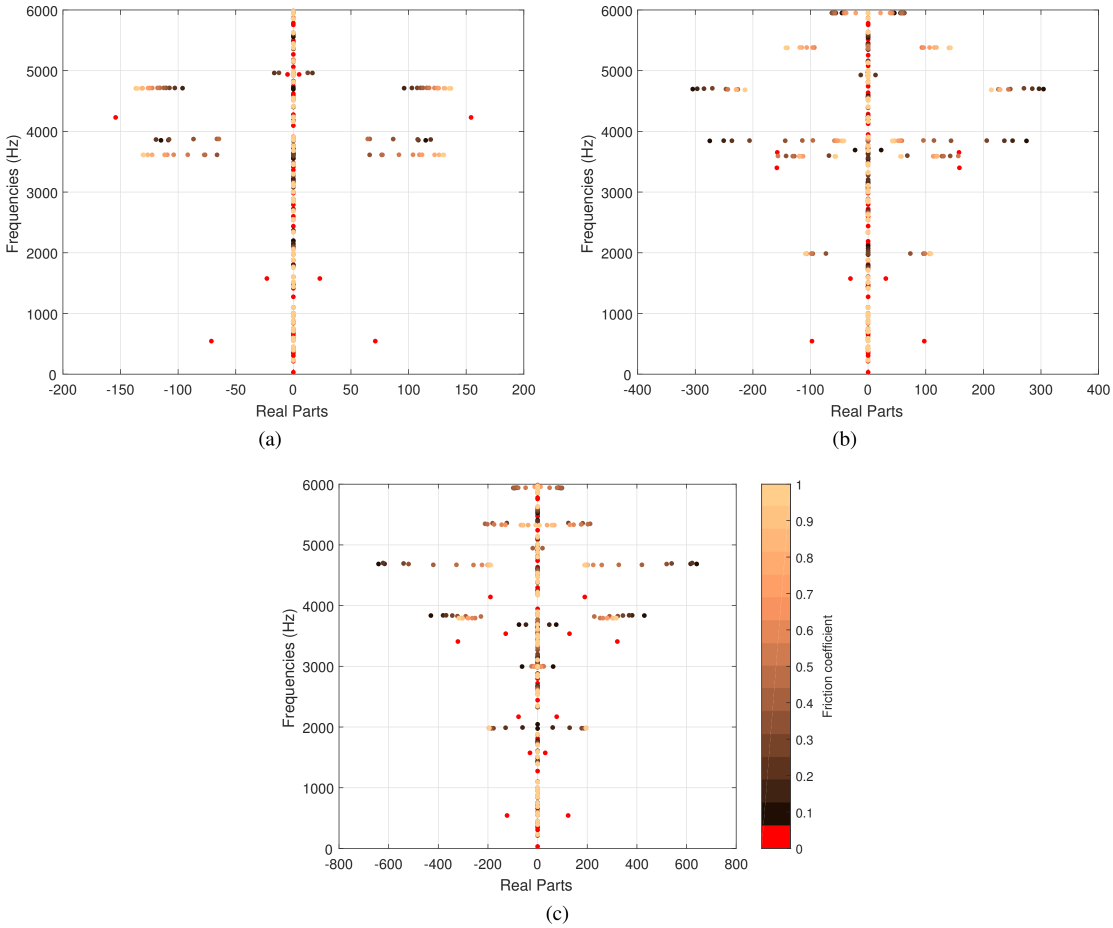

Fig. 4: Evolution of complex eigenvalues in the complex plan assuming a variation of the friction coefficient at the interfaces for three different values of $\mu_{pad-disc}$: (a) $\mu_{pad-disc} = 0.3$, (b) $\mu_{pad-disc} = 0.5$ (c) $\mu_{pad-disc} = 0.7$

eigenvalues (i.e. evolution of a given unstable mode versus the various configurations) can be distinguished (for example around [1850 – 2000] Hz or between [1080 – 1120] Hz), a strong dispersion of the frequencies of unstable modes is observed for frequencies higher than 3300 Hz. So the stability of the brake system is strongly influenced by the contact conditions. The squeal propensity of the brake system is also highly dependent of these contact conditions. These results point out the necessity of taking into account all the contacts that potentially exist in a brake system for the prediction of the squeal propensity. This even if the friction coefficient at the pad/disc interface $\mu_{pad-disc}$ is one of the most important parameter versus the stability analysis of a brake system. It is not sufficient to have an effective representation of the stability behavior of the braking system: considering only this frictional interface (and so neglect other friction interfaces) may lead to bad design of a brake system.

4. Conclusion

Stability analysis of an industrial brake system is conducted by considering not only the classical pad/disc frictional interface but also three potential contact interactions between piston/pad, bracket/pad and caliper/pad. Numerical results highlight that considering only the friction coefficient of the pad-disc interface is not sufficient for a good prediction of the squeal propensity. Numerous other contacts play an important role on the stability behavior of the system. When several contacts are considered, a high dispersion on the frequencies of unstable modes is observed.

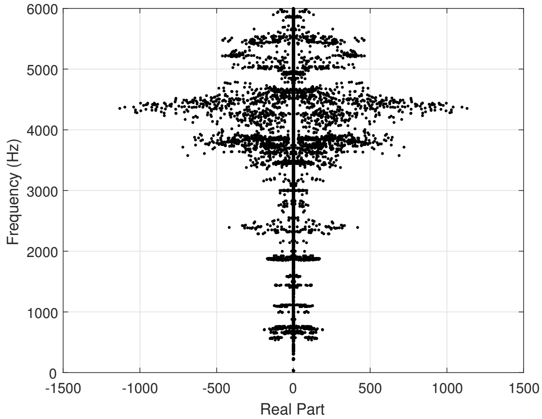

Fig. 5: Complex eigenvalues for all configurations considered

This study shows the difficulty to predict all instabilities with a unique configuration and highlights the necessity to consider various configurations in order to improve the prediction of squeal noise.

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