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A simplified approach for the calculation of acoustic emission in the case of friction-induced noise and vibration

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ABSTRACT

The acoustic response associated with squeal noise radiations is a hard issue due to the need to consider non-linearities of contact and friction, to solve the associated nonlinear dynamic problem and to calculate the noise emissions due to self-excited vibrations. In this work, the focus is on the calculation of the sound pressure in free space generated during squeal events.

The calculation of the sound pressure can be performed by the Boundary Element Method (BEM). The inputs of this method are a boundary element model, a field of normal velocity characterized by a unique frequency. However, the field of velocity associated with friction-induced vibrations is composed of several harmonic components. So, the BEM equation has to be solved for each frequency and in most cases, the number of harmonic components is significant. Therefore, the computation time can be prohibitive.

The reduction of the number of harmonic component is a key point for the quick estimation of the squeal noise. The proposed approach is based on the detection and the selection of the predominant harmonic components in the mean square velocity. It is applied on two cases of squeal and allows us to consider only few frequencies.

In this study, a new method will be proposed in order to quickly well estimate the noise emission in free space. This approach will be based on an approximated acoustic power of brake system which is assumed to be a punctual source, an interpolated directivity and the decrease of the acoustic power levels.

This method is applied on two classical cases of squeal with one and two unstable modes. It allows us to well reconstruct the acoustic power levels map. Several error estimators are introduced and show that the reconstructed field is close to the reference calculated with a complete BEM.

1. Introduction

The brake squeal phenomenon is characterized by high frequency noise emissions due to friction-induced self-excited vibrations. The prediction and the calculation of squeal noise are complex tasks which are composed of several steps [1]. The first one deals with the nonlinear modeling with the definition of the mechanical system geometry, the introduction of nonlinear laws corresponding to the contact and friction phenomenon over the interface. Then, a stability analysis is

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performed with respect to one or several parameters. This analysis allows us to detect the unstable equilibrium configurations that may lead to squeal and provides the fundamental frequencies of the unstable modes. The next step consists of calculating the time responses and it has been shown in the literature that the stationary regime associated with the squeal has a spectrum composed of fundamental frequencies (that should be different from the frequencies of the unstable modes [2]), their harmonic components and also their linear combinations. Then, the calculation of the noise radiations during squeal events can be performed with the Boundary Element Method [3,4]. This is the classical way to well estimate the sound pressure. This numerical method is based on the resolution of the Kirchhoff–Helmholtz equation which depends on the wave frequency. Therefore, the sound pressure calculation has to be carried out for each harmonic component of the velocity field.

In this aim, the multi-frequency acoustic calculation method has previously been developed [1]. The method is based on the decomposition of the velocity into Fourier series. Therefore, the global wave is decomposed into elementary waves with a unique frequency. The BEM is then applied for each wave and the global sound pressure field is calculated by superposition. The application of the BEM is composed of three steps. Firstly the boundary element model is built and it is composed of the system skin. Secondly, the surface sound pressure is calculated and finally, the sound pressure in the free space is evaluated by using the surface pressure. So, the latter is calculated for each frequency and the free field pressure is calculated for each frequency and each field plane. However, the number of frequencies can be significant making the computation time prohibitive.

In a previous work [1], it has been shown that only few harmonic components are predominant in the acoustic response. So it can be useful to determine these frequencies before the application of the multi-frequency acoustic calculation method. The first aim of the current paper is to develop a criterion based on the mean square velocity convergence (i.e. the dynamic response) which allows us to detect the predominant frequencies.

The second objective of this work is to propose a new method which allows us to quickly well estimate the sound pressure in the free field. In order to calculate the radiations, the directivity and a propagation model which describe the level decrease are required. The main idea is to determine the directivity with the BEM and to determine an analytical function with an interpolation. Finally, for each frequency, the sound pressure can be evaluated at every field points without the BEM.

The present paper is organized as follows. Firstly, the vibroacoustic of a squealing disc brake system is presented. The brake system under study is detailed and the dynamic and acoustic responses for two cases with one and two unstable modes are given. Secondly, the criterion which allows us to detect the predominant frequencies in the dynamic response is presented and applied for the two cases under study. Thirdly, the acoustic approximation method is presented and several sound pressure error estimators are presented. Finally, the acoustic method is applied and validated on the two cases.

2. Vibroacoustic of a squealing brake system

In this section, the simplified disc brake model under study is presented. The finite element model, the modeling of the frictional interface and the stability results are detailed. Next, the focus will be on the time responses associated with two classical cases of squeal with one and two unstable modes. Finally, the multi-frequency acoustic calculation method is applied to calculate the surface sound pressure and the sound pressure radiated in free space.

2.1. Brake system modeling

The two main components involved in the squeal are the disc and the pad which share a frictional interface. This allows us to focus on a simplified disc brake system composed of a circular disc and a pad as illustrated in Fig. 1. The inner radius of the disc is clamped due to the shaft connection and the outline of the upper surface of the pad can only translate along the normal **z**-direction to represent the caliper connection. In this work, automotive brake dimensions have been used.

The contact and friction phenomenon occurs over the common frictional interface which is modeled with nine uniformly spaced contact points as illustrated in Fig. 1(a). These contact elements are arbitrarily chosen and these number and positions strongly influence the stability analysis. However, increasing the number provides a better description of the interface but the calculation performances can be much reduced. In this study, nine elements seem to be a good compromise [2].

Between the previous selected points, nonlinear contact and friction forces are introduced. The nonlinear contact force vector is described with linear and nonlinear stiffnesses and contact/loss of contact configurations is considered. The cubic contact law associated with nonlinear contact elements takes the following expression:

$$F^{\mathrm{d}}_{\mathrm{contact},z} = \begin{cases} k_{\mathrm{L}}\delta + k_{\mathrm{NL}}\delta^3 & \text{if } \delta < 0 \\ 0 & \text{otherwise} \end{cases} \tag{1}$$

where $\delta = X_{\mathrm{p}} - X_{\mathrm{d}}$ is the relative displacement, X_{p} and X_{d} denote the normal displacements of the pad and the disc respectively. k_{L} and k_{NL} are the linear and cubic stiffnesses, $F^{\mathrm{p}}_{\mathrm{contact},z}$ and $F^{\mathrm{d}}_{\mathrm{contact},z}$ are the components of the normal contact force vector applied to the pad and the disc respectively. It can be noted that $F^{\mathrm{p}}_{\mathrm{contact},z} = -F^{\mathrm{d}}_{\mathrm{contact},z}$. This contact force expression has been chosen to fit experiments as explained in [5]. The main limitation of this formulation is that it allows