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Jean-Jacques Sinou, Olivier Chiello, Andréa Loyer, Xavier Lorang, Franck
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Jean-Jacques Sinou, Olivier Chiello, Andréa Loyer, Xavier Lorang, Franck Cocheteux, et al.. Approches expérimentales et numériques pour l'étude du comportement vibratoire de freins à disque ferroviaires vis-à-vis du crissement. CFM 2013 - 21ème Congrès Français de Mécanique, Aug 2013, Bordeaux, France. hal-03439046

HAL Id: hal-03439046

<https://hal.science/hal-03439046>

Submitted on 22 Nov 2021

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Experiments and simulations for squeal prediction on industrial railway brakes

J.-J. SINOÛ^a, O. CHIELLO^{b and c}, A. LOYER^{a,b and d}, X. LORANG^d, F. COCHETEUX^e AND S. BELLAJ^e

a. Laboratoire de Tribologie et Dynamique des Systèmes UMR 5513, Ecole Centrale Lyon, Ecully, France.

b. IFSTTAR, LAE, F-69675 Bron, France

c. Centre Lyonnais d'Acoustique, Université de Lyon, F-69622 Lyon, France

d. SNCF Innovative and Research Department, Physics of Railway System and Passenger, Paris, France.

e. SNCF Agence d'Essai Ferroviaire, Vitry-Sur-Seine, France.

Résumé :

Cet article présente quelques études expérimentales et numériques récentes sur la prédiction de crissement pour les freins ferroviaires. Des essais spécifiques sous différentes conditions de fonctionnement sont réalisés afin de mieux comprendre les phénomènes vibratoires et mécanismes générés lors de l'apparition du crissement. Ensuite, une stratégie numérique est proposée. Cette dernière considère non seulement la modélisation du système de freinage TGV et une analyse de la stabilité, mais également le processus transitoire non linéaire sur base réduite. Cette étude s'insère plus globalement dans le projet de recherche AcouFren, subventionné par l'ADEME, dont l'objectif est de proposer de développer des outils d'aide à la spécification et à la conception de freins à disque ferroviaires optimisés vis-à-vis du crissement.

Abstract :

This paper presents some recent experimental and numerical investigations on industrial railway brakes for squeal prediction. Specific experimental set-ups based on various braking tests are designed for investigating dynamic behavior of TGV squeal noise and its squeal characterization. Then, a numerical strategy is proposed, including not only the modeling of the TGV brake system and the stability analysis, but also the transient nonlinear dynamic and computational process based on efficient reduced basis. This study comes within the scope of a research program AcouFren that is supported by ADEME (Agence De l'Environnement et de la Maîtrise de l'Energie) concerning the reduction of the squeal noise generated by high power railway disc brakes.

Mots clefs : dynamique non-linéaire, crissement, expériences et simulations

1 Introduction

Friction-induced vibration and noise emanating from railway disc brakes is a source of considerable discomfort and leads to dissatisfaction for the passengers both inside and outside the trains in stations. Solving potential vibration problems requires experimental and theoretical approaches to obtain a better understanding of the phenomenon [1-2]. Although it has been the subject of many investigations over recent decades [3-4], friction-induced instabilities are still an active field of research in dynamics. The goal of this study is to present experimental and numerical analyses of the squeal vibration and prediction on industrial railway brakes. The first part of the paper gives a brief description of the TGV disc brake system and analysis of experimental data coming from tests on bench in laboratory SNCF. Secondly, the paper focuses on the numerical simulation: a stability analysis (i.e. complex eigenvalue problem) and a complete dynamic transient analysis (i.e. nonlinear self-excited vibration due to instabilities) are undertaken. More particularly, comparisons with experimental results will be performed in order to judge the relevance of the mechanical modelling strategy for squeal prediction on industrial railway brakes.

2 Experimental approach

2.1 Description of the TGV brake system

The disc-brake system is composed of four discs on each wheels axle and sliding bodies that are constituted of two symmetric lining plates with cylindrical pads (18 pad for each side), as illustrated in Figures 1. The brakes are activated by the pneumatic system pressure and slow down rotation of the wheels by the friction caused by pressing brake pads against brake discs.

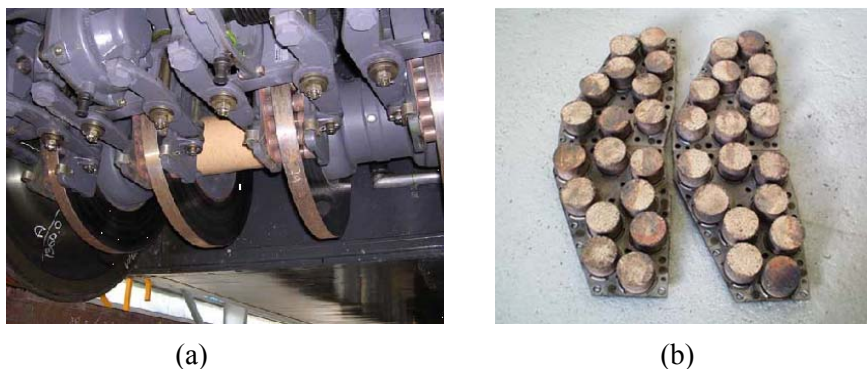


FIG. 1 – TGV brake system - (a) TGV bogie, (b) part of a brake pad

2.2 Experiments

The evaluation of the squeal prediction and the dynamical behaviour of the TGV brake system under working conditions are performed with the help of dynamic tests on bench that is located at SNCF Agence d'Essai Ferroviaire. The TGV disc is brought up to speed, and then pressure is introduced to activate the brake. The spectrum of brake squeal and transient vibrations are obtained via the experimental measurement. For this, the TGV brake system is fully instrumented with accelerometers on the stationary part. Vibration measurements for the rotating part (disc) are performed by using a vibrometer. Moreover, microphones are mounted near the disc. In the present study, experimental tests are divided into two main categories:

- experiments with an evolution of the rotational speed of the disc: these tests are called “transient braking tests” and correspond to real braking tests,
- experiments with a controlled steady rotational speed (i.e. dynamic fluctuations in rotational speed are not significant): these tests are called “controlled braking tests”.

The ultimate goal is to validate, via comparisons of experimental tests with various operating conditions (more particularly by taking into account the two possibilities of variable or constant rotating speed of the disc), the possibility of simulating squeal events via a numerical analysis which takes into account only a constant rotating speed of the disc.

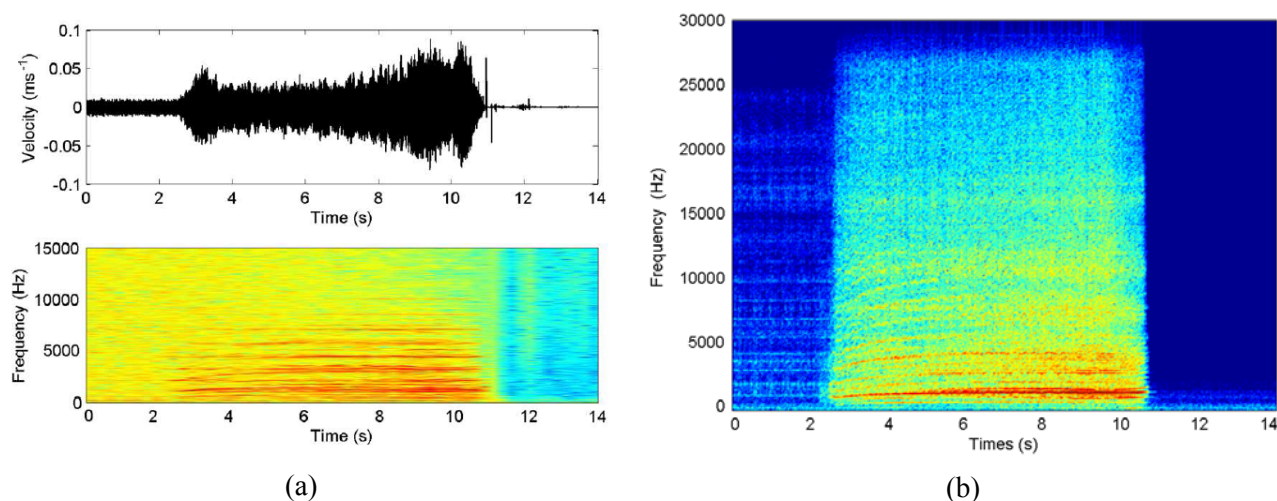


FIG. 2 – Experimental data for vibrometer measurement (a) controlled braking test- Time plot and wavelet power spectrum (b) Transient braking test Continuous Wavelet Transform

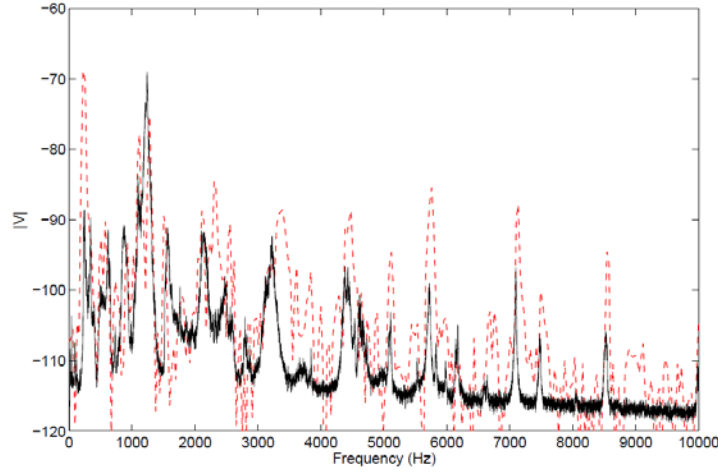


FIG. 3 – Comparison of controlled braking test (15kN-14km/h) and transient braking test (15kN-25km/h) at $t = 7$ s for an equivalent rotating speed of 14km/h (in dB) (red lines = transient braking tests; dashed black lines = controlled braking tests)

Figures 2(a) show the transient non-linear responses (velocity of a normal point on the disc) for a controlled braking tests and the wavelet power spectrum for vibrometer's measurement. The velocity response appears to be very complicated. First of all, it can be seen that the velocity response increases between $t = [2.5; 3.2]$ s. Secondly, a small decrease is observed between $t = [3.2; 3.8]$ s. Finally, a small increase of the velocity signal is shown for $t = [3.8; 10]$ s. Showing the associated wavelet power spectrum, several frequency contributions are observed during all the transient oscillations. Then, experimental results (vibrometer measurement) for a transient braking test are given in Figure 2(b). By comparing the transient braking tests and controlled braking tests (see Figure 3), the squeal characterization appears to be very similar: TGV squeal noise appears at low/middle frequencies in the 0–10000 Hz range with a finite number of frequency peaks. The most repeatable frequency contributions are around 1700Hz, 2150Hz, 2500Hz, 3400Hz, 3900Hz and 4600Hz.

3 Formulation of the problem

The TGV brake model is composed of one disc, outer and inner pads (18 pins applied on either sides of the disc are taken into account) modelled using the finite element method. Then, the backplate and support are considered by adding the flexibility of these pads' supporting structures. We assume a Coulomb law with a constant friction coefficient μ is used. This formulation can be summarized as follow:

$$\|\mathbf{r}_t\| \leq -\mu r_n; \quad \|\mathbf{r}_t\| = -\mu r_n \Rightarrow \exists \xi \in \mathbb{R}^+, \dot{\mathbf{u}}_t - \mathbf{v}_g = -\xi \mathbf{r}_t; \quad \|\mathbf{r}_t\| < -\mu r_n \Rightarrow \dot{\mathbf{u}}_t - \mathbf{v}_g = \mathbf{0} \quad (1)$$

where \mathbf{r} is the contact reaction, \mathbf{u} is the displacement field, \mathbf{v}_g is the Eulerian sliding speed, n and t subscripts stand for the normal and tangential projections of a field on the contact interface respectively. Moreover, to deal with the unilateral contact, a non-regularized Signorini law is chosen:

$$u_n - g \leq 0 \quad ; \quad r_n \leq 0 \quad ; \quad (u_n - g) r_n = 0 \quad (2)$$

where g is the initial gap at the contact interface. The main advantage of the Signorini law results in the fact that it does not require the introduction of a coefficient such as contact stiffness that would require measurement and should be difficult to estimate. By using classical finite element discretization of the problem with linear elements on the potential contact zone leads, the nonlinear dynamics problem may be written in a discrete form as follows:

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{C}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{f} + \mathbf{r}_c \quad (3)$$

where \mathbf{M} , \mathbf{K} and \mathbf{C} are the classical mass, stiffness and damping matrices of the system. \mathbf{f} and \mathbf{r}_c define the generalized force and contact reaction respectively. First of all, validation of the finite element model versus experiments is performed by applying a classical modal analysis. The contact reaction \mathbf{r}_c , the displacement \mathbf{u} and the velocity $\dot{\mathbf{u}}$ verify the contact and friction laws defined in Equation (1) and (2) at each mesh node. Classically, a reformulation of these contact and friction laws can be rewritten in terms of

projections on the negative real set ($proj_{\mathbb{R}^-}$) and on the Coulomb cone ($proj_{K_\mu}$). This formulation is used to facilitate the numerical implementation in the treatment of the contact state [5]

$$r_n = proj_{\mathbb{R}^-} \left(r_n - \rho_n'' (u_n - g) \right), \forall \rho_n'' \text{ where } proj_{\mathbb{R}^-} (x) = \min(x, 0) \quad (4)$$

$$\mathbf{r}_t = proj_{K_\mu} \left(\mathbf{r} - \rho_t (\dot{\mathbf{u}}_t - \mathbf{v}_g) \right), \forall \rho_t > 0 \text{ with } proj_{K_\mu} (\mathbf{x}) = \begin{cases} \mathbf{x}_t & \text{if } \|\mathbf{x}_t\| / |x_n| \leq \mu \\ \mu \frac{|x_n|}{\|\mathbf{x}_t\|} \mathbf{x}_t & \text{otherwise} \end{cases} \quad (5)$$

where ρ_n'' and ρ_t are two arbitrary positive scalars called normal displacement augmentation parameter and tangential augmentation parameter respectively [6].

4 Numerical simulation

4.1 Stability analysis

First of all, a classical stability analysis can be performed. The first step is the static problem: the steady-state operating point for the full set of non-linear equations is obtained by solving them for the equilibrium point. The steady sliding equilibrium of the TGV brake system is given on Figure 4(a) for a friction coefficient of 0.35 and a given net brake pressure: it is observed that this sliding position is not trivial with a more or less pronounced compression of pads. Moreover, the effects of the rotational direction of the disc can also be seen. Then, one obtains the linearized equations of motion by introducing small perturbations about the equilibrium point into the non-linear equations [6]. Stability consists on computing the complex modes and the complex eigenvalues associated to the linearized problem in the frequency range of interest. Solving this problem is achieved by using the Residual Iteration Method [7]. The stability analysis of this equilibrium point is given on Figure 4(b) (for a friction coefficient of 0.35). Nine unstable modes (with positive divergence rate) are detected.

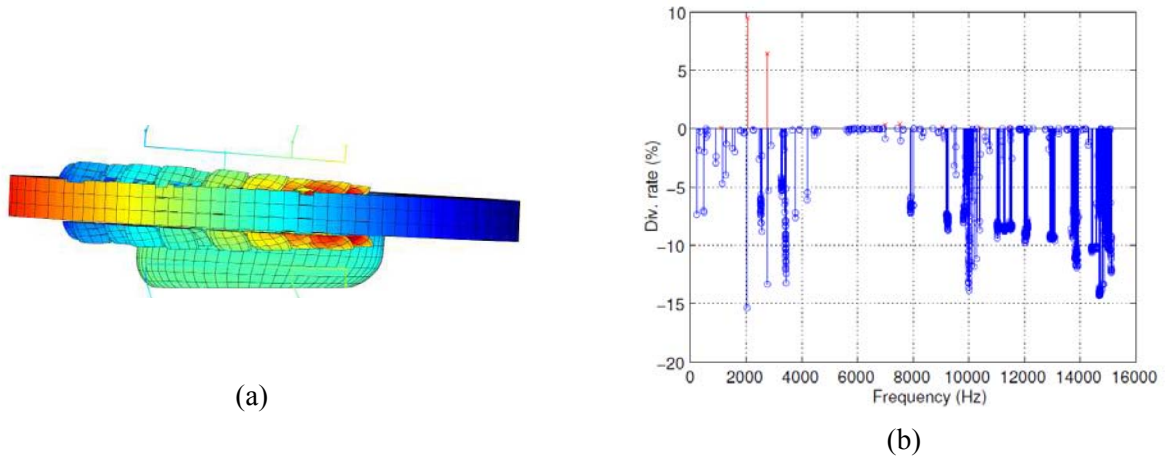


FIG. 4 – Stability analysis of the TGV brake system (a) Sliding equilibrium (b) Eigenvalues in the complex (red=unstable modes; blue=stable modes)

4.2 Transient nonlinear dynamic and self-excited vibration

As previously explained in [8], the stability analysis may lead to an underestimation or an over-estimation of the unstable modes observed in the non-linear time simulation due to the fact that linear conditions are not valid during transient oscillations. Therefore, a numerical resolution of the complete nonlinear system has to be performed in addition to the stability analysis to estimate the nonlinear behaviour of the solution far from the sliding equilibrium. Since the instability of the sliding equilibrium may lead to strongly nonlinear events with contact and no-contact states at the different frictional interfaces between each pad and the disc, a first-order θ -method time integration scheme [6] is developed for the computation of the transient solution.

Moreover, due to the size of the system and the extensive time of the nonlinear solving process, an efficient spatial model reduction is performed to estimate the nonlinear behavior of the TGV brake system. The

proposed reduction basis has been previously discussed and validated for the case of an elastic layer with a frictional interface [6]. The chosen basis is a classical modal truncated basis built from the real and imaginary parts of the complex stability modes (dynamic modes) with addition of constrained boundary modes at the contact interface (static modes). In the paper of Loyer et al. [6], this type of basis has been referred to as F_n^s where n is the number of modes included and s corresponds to the inclusion of static modes in the reduction basis. In the present study, 1000 dynamic modes are kept in the reduced basis ($n = 1000$, F_{1000}^s): this represents almost all the stability modes in the $[0 - 15]$ kHz frequency range. This is a large number of modes but the frequency range of interest is also large ($[0 - 10]$ kHz) and the modal density of the pads structure is rather high due to the number of pins. However, considering the initial number of degrees of freedom (72685), the reduced basis is in any case advantageous.

Figure 5(a) illustrates the non-linear transient solution of the velocity in the case of $\mu = 0.35$. A succession of two phases is shown: the first one (for $t < 0.01$ s) corresponds to a classical increase of the solution. The second one (for $t = [0.01; 0.11]$ s) is characterized by a global saturation with oscillations. Comparing the time-history computation and experiments that have been previously presented in Section 2 (see Figures 2), the level of stationary amplitudes is well reproduced. In order to better analyze the evolution of the nonlinear transient behavior of the TGV brake system, Figure 5(b) expands the numerical transient solution on the basis of the complex modes. For the interested reader, the complex modal projection and formulation are explained in details by Loyer et al. [6]. Following the evolution of the modal participations allow us to evaluate the energy contribution of each mode during the transient response of the TGV brake system. Showing the complex modal projection (see Figure 5(b)), the contribution of each unstable mode (that have been previously computed through the stability analysis) during the nonlinear transient and stationary solutions appears to be very clear. Two main zones are detected: for $t = [0; 0.01]$ s, a transition phase is observed with a fast increase of the contribution of modes 1, 2 and 3 (denoted M1, M2 and M3 respectively in Figure 5(b)). For $t > 0.02$ s the modal participation of all unstable modes fluctuates a little with a global signal suggesting that we have reached an approximately steady state solution. So it clearly appears that the modal participations of modes 4, 5, 6, 7, 8 and 9 remain a low level compared to the modal participations of modes 1, 2 and 3. In conclusion, the non-linear response is mainly composed of the evolution of three principal modal participations.

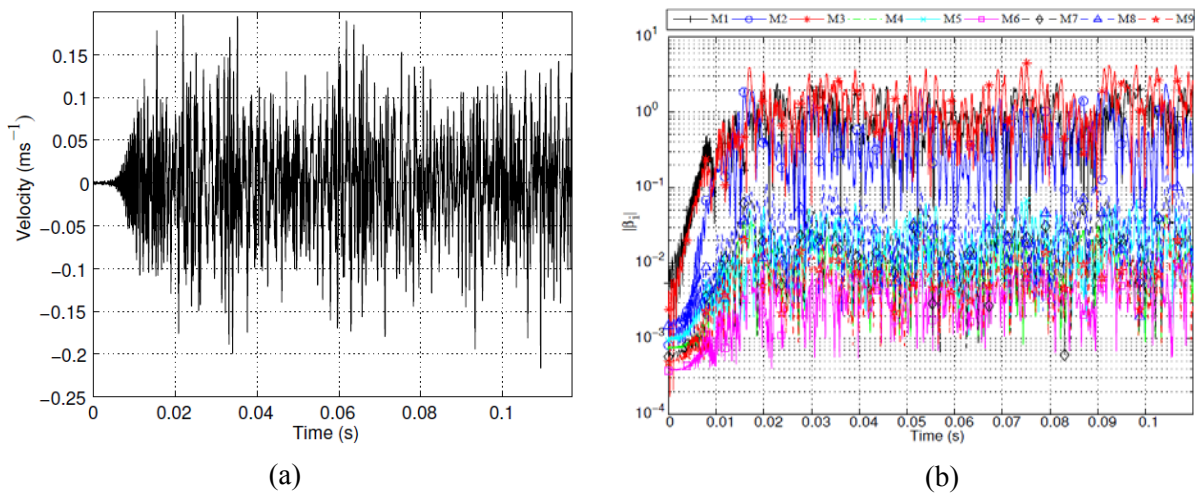


FIG. 5 – Transient analysis of the TGV brake system with reduced basis (a) Normal velocity and (b) complex modal projection where “Mi” defines the i th unstable mode

5 Comparison with experiments

A typical brake squeal spectrum obtained via numerical simulations with the reduced basis is presented in Figure 6 and compared with measurements that have been discussed in Section 2 (see also Figure 4). While some features are recognizable there are also clearly some significant differences. First of all, some differences are observed in the frequency range $[0; 2500]$ Hz: even if it is conceivable to correlate some experimental and numerical frequency peaks, it is clearly shown that the experimental frequency peaks are more distinct. Some similarities are also visible: for example, it can be distinguished two peaks around 3200Hz and 4600Hz for both experiments and numerical simulations. However, comparisons of the peak

amplitudes are not quite the same order of magnitude. Then, the global magnitudes and the frequency content and peak magnitudes appear to be more coherent between [500; 10000]Hz. Even if a deviation of the frequency peaks are visible (around 5700Hz, 6200Hz, 7100Hz, 7500Hz and 8500Hz), an agreement between experiment and numerical tests is found. So it may be concluded that comparisons show some similarities but also significant differences.

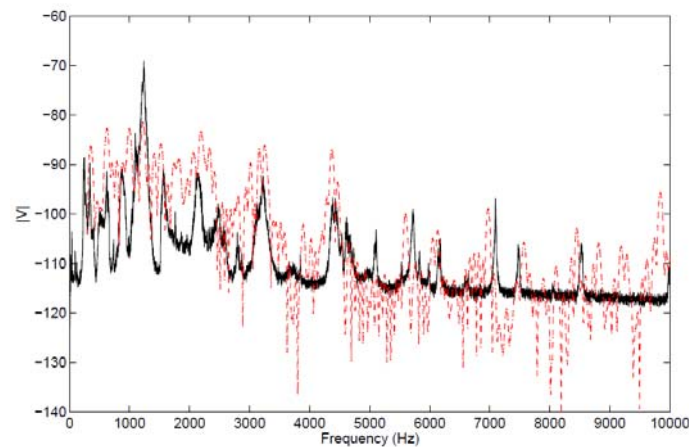


FIG. 6 – Comparisons of brake squeal spectrum via experimental and numerical approaches (in dB) (black lines=experiments; dashed red lines=reduced basis)

6 Conclusion

First of all, this paper presents experimental analysis to understand the mechanism of TGV brake squeal. Even if the phenomenon of squeal can be complex, experiments show that squeal can be clearly identified as the emergence of a finite number of frequencies regardless the operating conditions. Secondly, a complete finite element model of TGV brake system has been developed to model vibration instabilities at the origin of disc brake squeal. Then, numerical methods dedicated to stability analysis and transient computations for industrial models have been proposed. Numerical results are in agreement with the experimental tests for the prediction of brake squeal.

Acknowledgments

The authors gratefully acknowledge ADEME (Agence De l'Environnement et de la Maîtrise de l'Energie) who has partially funded this project AcouFren.

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