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## Structural Acoustics and Vibration
## Session 3pSAb: Applications in Structural Acoustics and Vibration I

# 3pSAb1. Squeal noise generated by railway disc brakes: Experiments and stability computations on large industrial models

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The squeal noise generated by railway disk brakes is an everyday source of discomfort for the passengers both inside and outside the trains in stations. The development of silent brake components is needed and requires a better characterisation and understanding of the phenomenon. This is the aim of the experimental and numerical investigations performed in the framework of the French AcouFren project and presented in this paper. The first part deals with the analysis of experimental data coming from bench tests in a lot of braking configurations including different brake pads. In the second part, the measurements are compared with the results of a large FE model of the brake taking into account the mechanical complexity of each component, especially the brake pads. Components models have been previously updated using experimental modal analysis but the whole model is a direct assembling of it, without updating. The assumption of unilateral contact and Coulomb friction at the pad/disc interface is sufficient to destabilize the sliding equilibrium of the brake and lead to self-sustained vibrations. Complex vibrating modes are computed in order to describe and understand the dynamic instabilities.

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## INTRODUCTION

The squeal noise generated by vehicles brakes is a difficult problem in the automotive and aeronautic industry but also for railways. When trains arrive in stations, squeal noise levels sometimes reach more than 100 dB(A) on the platform. Considering that the squeal spectrum is most of the time composed of one or several pure frequencies, it is a very annoying noise for both passengers and residents. Research for predict and remove or at least reduce squeal noise has been regularly performed for many years. In the last decades, great progress have been done in the modeling of friction-induced instabilities [1] and brake squeal [2, 3]. This is the case in particular for the TGV disc-brake system for which a refined mechanical modeling of the phenomenon has been carried out in order to understand the mechanism of squeal generation [4, 5, 6]. It seems that predictive industrial models can now be developed and be used to specify and design technological solutions.

One of the aim of the French research program AcouFren is to develop and valid such a squeal model for the TGV brake. Although it is known that the design of the brake system is crucial for the squeal generation, the project model is expected to allow the design of new pads without any change on the system itself : the challenge is thus to reproduce the vibratory and acoustic experimental behavior of the brake for several kinds of pads. The strategy is first to develop detailed updated models of the structural brake components (disc, caliper and pads) and then to assemble them (without updating of the whole structure) by using simple structural laws at the contact interface.

The first part of the paper describes the brake system. The second part deals with the mechanical characterization and the modeling of the brake components including 3 “pilot” pads. In the third part, the various experiments performed on the brake system are presented. Typical squeal spectra corresponding to the pads are discussed and compared. The last part is dedicated to the model. The main assumptions and numerical methods are briefly exposed and the results of the stability analysis are compared with squeal measurements using modal frequencies, divergence rates but also disc energy contributions.

## DESCRIPTION OF THE BRAKE SYSTEM

The disc-brake system of TGV trains is composed of a steel disc clamped to the bogie axle through a thin hub, a cast-iron "caliper-type" structure suspended to the bogie and controlled by a pneumatic system, and two symmetric pads fixed into the caliper on each side of the disc (cf. Figure 1).

**FIGURE 1:** The TGV disc-brake system : "caliper-type" structure (left), pads fixed into the caliper (center), brake assembly on the test bench (right)

During braking, symmetric axial forces are transmitted from the piston to the pads, compressing the rotating disc. Friction forces at the disc-pads interface generate the expected braking couple. On a train there are four discs on each axle whereas in the present study, the disc is directly clamped to the transmission tree of the test braking bench as shown on Figure 1. Each pad is made of one or several pieces of friction materials (friction pins) fixed to a supporting backplate. The main differences between the pads lie in the the geometry of the friction pins, the mechanical characteristics of the friction material, the connection between the pins and the backplate and the mechanical characteristics of the backplate. Three pads (called G1, G2 and G3) have been thoroughly tested and modeled. Pad G1 is composed of 9 circular friction pins rigidly clamped to a simple steel backplate. Pad G2 is composed of 9 circular friction pins rigidly clamped to a more robust backplate composed of 2 riveted steel plates. Pad G3 is composed of 10 polygonal friction pins fixed in pairs to a simple steel backplate through an original mechanism bringing flexibility and damping to the connection. The friction materials of the three pads are different.

## MECHANICAL CHARACTERIZATION AND MODELING OF BRAKE COMPONENTS

### Disc and Caliper

In a first step, experimental modal analysis have been performed separately on the disc and on the caliper without contact between the two components. For this purpose, the disc has been excited near its periphery in its three main directions (axial, radial and tangential) whereas for the caliper, only one axial excitation located on the pads supporting structure has been used. The caliper has been pressed directly (i.e. without pads) on a soft material instead of the disc in order to simulate free contact conditions. More than 70 disc modes have been identified between 0 and 15 kHz. Most of them are double modes, because of the axi-symmetry of the disc. Excepted the first very first modes, measured modal factors are rather low, particularly for the axial modes without nodal diameters and the tangential modes. High resonance peaks can be clearly identified in the Frequency Responses Function given on Figure 2. Conversely, caliper modes have been identified only up to 5 kHz and they are highly damped.

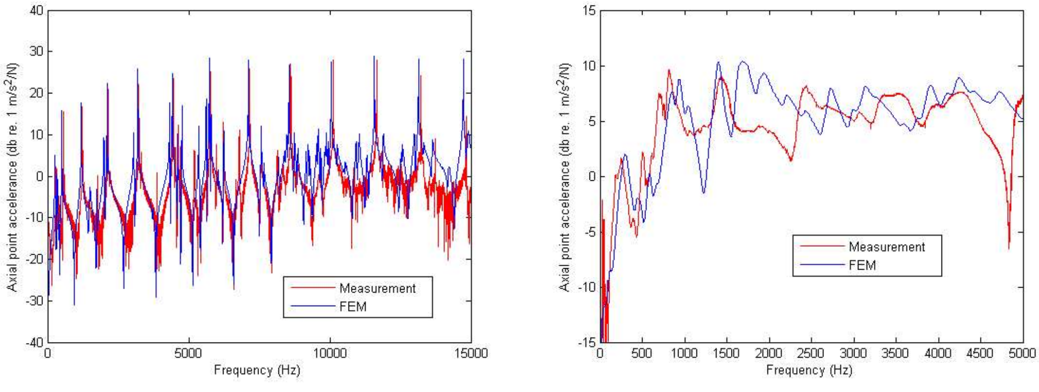


**FIGURE 2:** Experimental and numerical axial point accelerances of the disc (left) and the caliper (right)

The two structures have been modeled using finite elements, linear elasticity theory and small displacements assumption. Mechanical parameters have been updated using frequency and shape correlations between numerical and experimental modes. Figure 2 show that good results have been obtained for the disc whereas more difficulties have been encountered with the caliper. This is not surprising since this a complex structure with many connections leading to non linear local effects like friction and contact.

### Pads

In order to characterize the pads, three experiments have been performed. The two first tests concern the friction material itself. Specific experimental set-ups have been developped to measure both the complex dynamic compression and shear stiffness of the friction pins between 0 and 1000 Hz. No significant frequency dependance has been noticed in this frequency range. Assuming an orthotropic elastic behavior, elastic parameters of the friction material have been deduced from this test by updating FE models of the pins. The third test is an experimental modal analysis of the whole pad in free conditions. The first 10 modes of the pad have been identified below a frequency varying from 700 to 2300 Hz, according to the pad stiffness. Table 1 summarizes the main founded characteristics for the three pads. It is clear that pad G2 is the stiffest structure, with regards to both the friction pins and the backplate. Backplate of pad G2 is also more damped. Pads G1 and G3 have rather similar pins and backplate characteristics but one should not forget that, unlike G1, pad G3 has a flexible pins/backplate connection.

**TABLE 1:** Characteristics of the pads

| | **G1** | **G2** | **G3** |
|---|---|---|---|
| Friction pins | | | |
| Geometry | circular | circular | polygonal |
| Compression stiffness | 50 MN/m | 98 MN/m | 52 MN/m |
| Shear stiffness | 122 MN/m | 413 MN/m | 77 MN/m |
| Damping factor | 3.0 % | 4.3 % | 4.2 % |
| Backplate | | | |
| Frequency of the 1st mode | 267 Hz | 617 Hz | 231 Hz |
| Damping factor | 0.4 % | 1.9 % | 0.4 % |
| Pins/backplate connection | rigid | rigid | flexible |

These three pads have been finely modeled by finite elements, keeping the main geometrical and structural complexities in the model. Elastic and damping parameters of the backplate and of the pins/backplate connections have been updated using frequency and shape correlations between numerical and experimental free modes.

## SQUEAL MEASUREMENTS

### Experimental Set-up

The measurement of the vibro-acoustic behavior of the TGV brake system under working conditions has been performed with the help of dynamic tests on bench that is located at SNCF Agence d'Essai Ferroviaire. The structure has been fully instrumented with accelerometers on the caliper and a laser vibrometer aiming at the disc surface in order to measure its normal velocity as shown on Figure 1. A microphone has also been placed in front of the brake at 1 m from the disc surface. Operational parameters such as the normal load, the rotation speed of the disc, the resulting braking couple and the ambient temperature has been measured and controlled during experiments. By using the normal load, the braking couple and a mean braking radius, an apparent mean friction coefficient has also been calculated. In addition, two kind of tests have been conducted. Transient braking tests (with decreasing speed) correspond to real braking tests for which the mass to brake is constant. Stationary braking tests (with constant speed) are performed by controlling virtually the braking couple transmitted to the disc. The second test allows us to justify one of the main assumptions of the model presented in the next section (i.e. the rotation speed of the disc is supposed to be constant). The repeatability of the tests has been investigated and, as a whole, good results have been found.

## Experimental Results

Firstly, transient and stationary braking tests have been compared. Whereas transient braking tests show an evolution of the vibro-acoustic behavior during braking, stationary braking tests generally lead to stationary squeal. By restraining the observation of the transient braking tests in a time window in which the rotation speed is close to the speed of the stationary tests, the comparison is rather convincing: global vibratory and acoustic amplitudes are the same order of magnitude and the frequency contents appear to be very similar. However, some differences remain in the relative amplitudes of the various frequency peaks.

Secondly, the effects of the operational parameters have been investigated. The most influential of them are the normal load and the rotation direction. The normal load most often lead to richer spectra and higher amplitudes. The rotation direction may have significant effects (it may radically change the squeal spectrum) depending on the pad. The influence of the rotation speed seems to be correlated with the variation of the resulting apparent mean friction coefficient. At low rotation speeds, the friction coefficient often increases, leading to higher squeal levels but the variation of the rotation speed itself (i.e. without variation of the friction coefficient) has little effects.

The third but most important investigation concerns the role of the pad in the squeal generation. Each tested pad leads to a specific kind of squeal with very different frequency content and global levels as shown on Figure 3. With pad G1, the brake generates an intense squeal noise at low/middle frequencies (0-5000 Hz) with higher peaks around 1200, 1900 and 2500 Hz. The frequency content of the disc normal velocity measured by the vibrometer is very similar suggesting that an important part of the noise is radiated by the flexural vibrations of the disc. The squeal noise generated by the brake equipped with pad G2 contains predominant energies in the middle/high frequency range (3300, 6600 and 9900 Hz) and the pressure levels are very high at this frequencies (110 dB at 6600 Hz). Like for pad G2, by observing the vibrometer results, it is no doubt that the disc vibration has an important contribution in the acoustic emission. Comparing with pads G1 and G2, pad G3 is almost "silent". The noise levels are significantly lower with only some small peaks around 2500 and 5000 Hz. In addition, comparing the acoustic and vibrating spectra, it seems that the role of the disc is only partial in the noise emission. To summarize, it is evident that the characteristics of the pads are crucial in the squeal generation of the brake. However, although these results suggest that the flexibility of the pins/backplate connection is benefic to reduce the noise, the role of the other parameters like the stiffness of the friction pins or the stiffness of the backplate is not yet clear. In order to attempt to understand these effects, the development of a model is necessary.

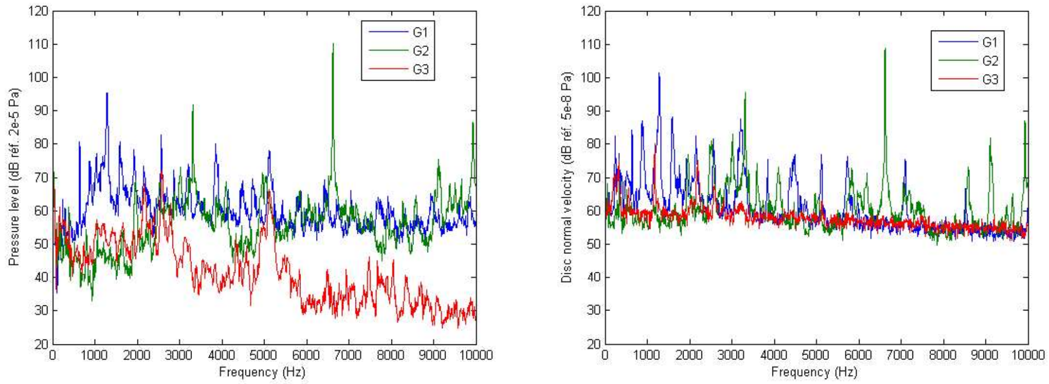


**FIGURE 3:** Typical squeal spectra corresponding to the 3 pads: microphone pressure level (left) and disc normal velocity measured by the vibrometer (right)

## SQUEAL MODELING AND COMPARISONS WITH MEASUREMENTS

### Main assumptions

The proposed approach, detailed for instance in [6] is based on the following classical assumptions. The squeal noise is supposed to be the result of the sound radiation of the structural components of the brake system in self-sustained vibrations. The self-sustained vibrations are supposed to be due to the dynamic instability of the sliding equilibrium of the system. The instability is supposed to be a *structural mode-coupling instability* induced by the non symmetry of the friction law at the pads/disc contact interface. In addition, compared with the braking duration, a short period of time is also considered. During this period, the rotating speed, the normal load and the friction coefficient are supposed to be constant and the wear and thermo-mechanical phenomena is neglected. The structural components are modeled by finite elements as detailed in section 2, but with compatible meshes at the interface. For the sake of simplicity, *Coulomb friction laws with constant friction coefficient and unilateral (Signorini) contact laws* are chosen: these laws are theoretically sufficient to highlight the main characteristics of the instabilities responsible for the squeal noise, according to the above assumptions.

### Overview of the approach

Three steps are needed to study the dynamics of the system. First, the sliding equilibrium is determined. It consist of finding a deformed quasi-static state of the structure which satisfies the sliding unilateral contact conditions. An iterative status algorithm is used to solve the corresponding non symmetric non linear problem. The stability of this equilibrium is then studied. This consists of finding the future of some sliding vibratory perturbations of the equilibrium by calculating the natural solutions of the linearized equations of the problem. Practically, the complex modes $\Phi_\mathbf{i}$ and eigenvalues $\lambda_i$ of a second-order non symmetric linear system is calculated :

$$\begin{gathered}(\lambda^2\mathbf{M}+\lambda\mathbf{C}+\mathbf{K})\mathbf{U}=\mathbf{P_n}^T\mathbf{R_n}+\mathbf{P_t}^T\mathbf{R_t}\\ \mathbf{P_n}\mathbf{U}=\mathbf{0}\\ \mathbf{R_t}=\mu\mathbf{R_n}\end{gathered} \tag{1}$$

where $\mathbf{R_n}$ and $\mathbf{R_t}$ denote respectively the normal and tangential generalized reactions at frictional contact interface, $\mathbf{P_n}$ and $\mathbf{P_t}$ are the corresponding observation matrices (e.g pointing respectively to the normal and tangential contact relative displacements) and $\mu$ is the friction coefficient. Frictional constraints are taken into account by adding frictional matrices to the classical FE matrices whereas normal constraints are verified by classical elimination [6]. The time evolution of a mode envelope relates to the sign of the real part of the eigenvalue. If this real part is negative, the perturbation vanishes and the equilibrium is found again: the mode is stable. If this real part is positive, the perturbation growths exponentially and is likely to generate self-sustained vibrations: the mode is unstable. A classical indicator is the modal divergence rate given by :

$$\zeta_i=\frac{\Re(\lambda_i)}{\omega_i} \tag{2}$$

where $\omega_i=\Im(\lambda_i)$ is the natural pulsation of the mode. Divergence rates are physically equivalent to negative modal damping factors. One must emphasize that the non symmetric eigenvalue problem is solved by using a reduced basis combining real undamped frictionless modes and a first-order static approximation of the responses due to the friction forces, as proposed in [6]. This is an essential tool to avoid the high computational costs due to the size of the model presented in the next subsection.

A common interpretation of the stability analysis is that the frequencies and the mode shapes of the main unstable complex modes are close to the self-sustained vibrations. This is a very approximative rule since non linear effects may change both the frequencies and the shapes of the modes. Moreover, non linear effects may also add significant contributions around the harmonics of the modal frequencies and even annihilate the contributions of some very unstable modes. Finally, stability analysis does not give any useful information about the amplitude of the self-sustained vibrations. So the resolution of the non linear dynamic problem is a necessary step in the unstable cases. This consists in finding the limit cycles to which the transient solution of the problem converges. This objective is part of the AcouFren project but it is not tackled in this paper. Nevertheless, some preliminary transient non linear results on simplified brake models have shown that unstable modes with high mechanical energy contributions of the disc are more likely to appear in the final limit cycle than the others. Moreover, according to the experimental results, an important part of the squeal noise seems to be radiated by the disc. This is why, in the next paragraph, stability results are interpreted using divergence rates but also disc contributions.

## Stability results

After coupling the disc, the caliper and the pads, the FE model of the whole structure is very large (cf. Figure 4). It must be noticed that no updating has been performed on this assembly. Depending on the pad, the total number of degrees of freedom (dof) lies between 600000 and 800000 and the number of contact degrees of freedom between 2500 and 5000. For each pad, 700 complex modes have been calculated allowing the determination of complex modes below a limit frequency of 8 kHz for the softer pad (G3) and 14 kHz for the stiffer pad (G2). The effects of the operational parameters and the friction coefficient have been examined. In this paper, only results corresponding to the typical squeal noises highlighted in the previous section are discussed. Simulations corresponding to a friction coefficient $\mu = 0.5$ are presented, considering that the measured apparent mean friction coefficient may vary between 0.44 and 0.56 depending on the pad.

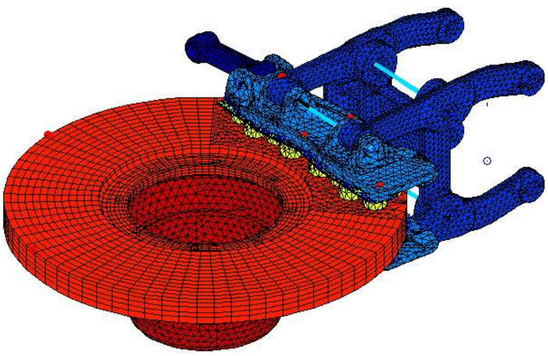

**FIGURE 4:** Model of the whole brake system including the disc, the caliper and the pads

Figures 5 show the divergence rates and the disc energy contributions of the significantly unstable modes (modes for which $\zeta_i > 0.1\%$) for the three pads. Many differences between the pads are found. With pads G1 and G3, many unstable modes are found. Most of them are “pads” modes with frequencies comprised in a narrow frequency range depending essentially on the stiffness of the friction pins (between 3 and 4 kHz with pad G1 / between 4 and 5 kHz with pad G2). With pad G1, only three modes have their disc energy contributions higher than 10 %: these are 3 modes of the assembly near 1000, 1900 and 2500 Hz. With pad G2, the modes having the highest disc contribution are two modes of the assembly near 3000 Hz and one “disc” mode at 6600 Hz. Divergence rates corresponding to pad G2 are also higher than those corresponding to pad G1. The results obtained with pad G3 are more simple: only one pad/caliper mode is significantly unstable with a frequency near 2500 Hz. Beyond all the reserves about the difficulties to interpret the stability analysis, all these results seems to be in agreement with the measurements. The global level of stability induced by the different pads is well reproduced and the frequencies of the unstable modes can be related with the frequency content of the acoustic and vibratory squeal spectra, considering that some measured peaks may be due to the harmonics induced by non linear effects.

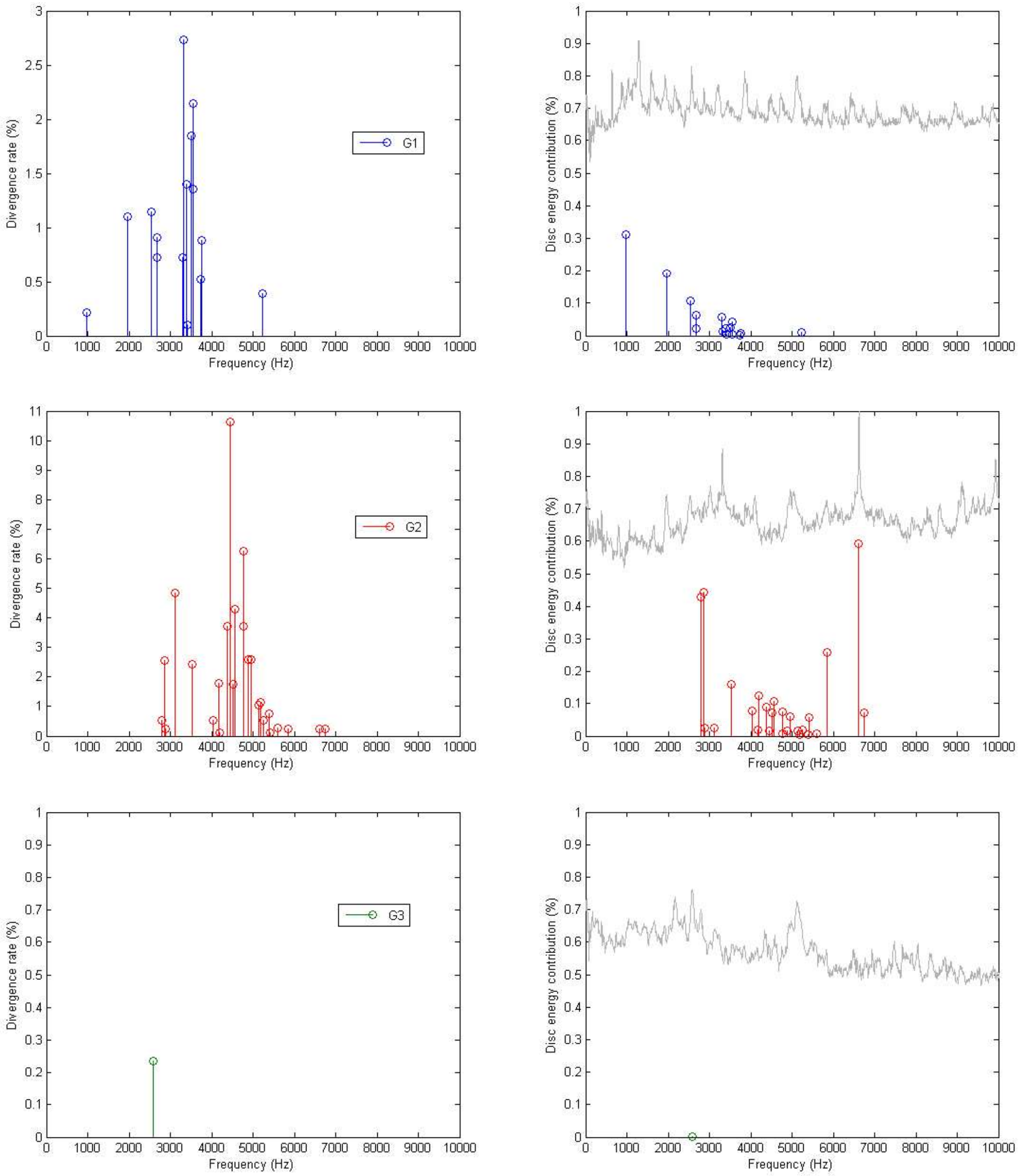


**FIGURE 5:** Divergence rates (left) and disc energy contributions (right) of unstable modes of the brake with pad G1 (top), G2 (center), G3 (bottom). Corresponding unscaled acoustic spectra (in dB) are superimposed on disc contributions (in gray)

## CONCLUSIONS

Considering the progress done in the modeling of friction-induced vibrations during the last years, the development of predictive squeal models seems to be a realistic challenge. The aim of the presented works is to show how simulations and measurements can be correlated for industrial brakes like TGV disc brakes. In particular, the influence of the pad on the stability of the whole brake system has been investigated. Results are encouraging: the FE model reproduces the differences between the pads without any updating on the assemblies. This is a first stage since only stability results are discussed but works on non linear transient simulations are in progress.

## ACKNOWLEDGMENTS

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## REFERENCES

[1] R. Ibrahim, "Friction-induced vibration", ASME Applied Mechanics Review **47(7)**, 209–226 (1994).

[2] N. Kinkaid, O. O'Reilly, and P. Papadopoulos, "Automotive disc brake squeal", Journal of Sound and Vibration **267**, 105–166 (2003).

[3] H. Ouyang, N. Nack, Y. Yuan, and F. Chen, "Numerical analysis of automotive disc brake squeal: a review", International Journal of Vehicle Noise and Vibration **1**, 207–231 (2005).

[4] X. Lorang, F. Foy-Margiocchi, Q. Nguyen, and P. Gautier, "Tgv disc brake squeal", Journal of Sound and Vibration **293**, 735–746 (2006).

[5] X. Lorang and O. Chiello, "Stability and transient analysis in the modelling of railway disc brake squeal", Notes on Numerical Fluid mechnaics and Multidisciplinary Design **99**, 447–453 (2008).

[6] D. Brizard, O. Chiello, J.-J. Sinou, and X. Lorang, "Performances of some reduced bases for the stability analysis of a disc/pads system in sliding contact", Journal of Sound and Vibration **330**, 703–720 (2011).