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ETUDES EXPERIMENTALES ET SIMULATION NUMERIQUE DU PHENOMENE DE CRISSEMENT SUR TGV

Experiments and numerical simulation of TGV disc brake squeal

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Keywords : *squeal, nonlinear dynamic, experiments, numerical simulation, friction.*

Résumé

Le crissement est un bruit strident fréquemment produit par les systèmes de freinage. Dans le milieu ferroviaire, des relevés de niveau acoustique ont montré que le crissement dû à l'arrivée en gare de certains trains pouvait atteindre 110 dB à un mètre du bord du quai. Ainsi, la problématique liée au crissement de freins à disque ferroviaires vise à traiter la gêne occasionnée par le crissement, principalement pour les passagers présents sur le quai lors de l'arrivée d'un train en gare, mais aussi pour les riverains et le personnel présent dans les gares. Cette étude vise donc à mieux comprendre les phénomènes vibratoires et mécanismes générés lors de l'apparition du crissement des freins à disque ferroviaires. Pour ce faire, des essais expérimentaux variés, ainsi que des confrontations avec des modèles éléments finis et simulations numériques complexes sont proposés. 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.

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]. In the automotive and aeronautic industries the phenomenon of brake squeal is well known

because of the noise and vibration produced. Although it has been the subject of many investigations over recent decades [3-8], 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: first of all, validation of the finite element model is performed by applying a classical modal analysis. Then, 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.

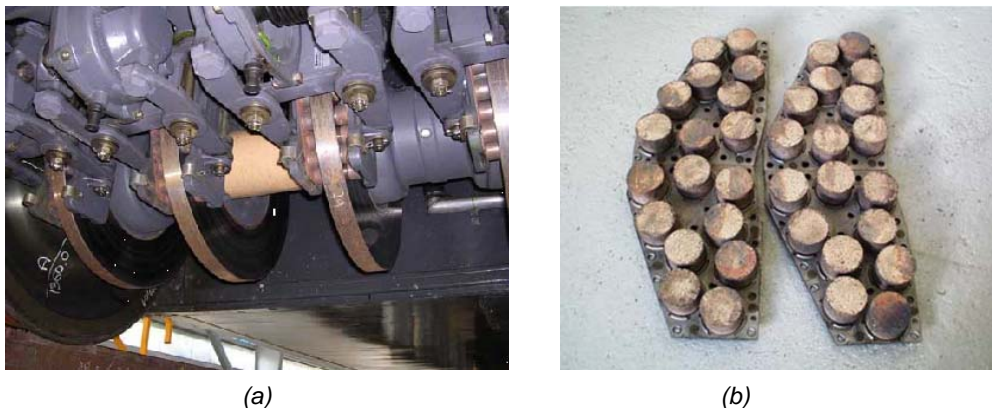


Fig. 1. TGV brake system - (a) TGV bogie, (b) part of a brake pad.

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 test ends when the TGV disc stops. 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, as indicated in (Fig. 2). Vibration measurements for the rotating part (disc) are performed by using a vibrometer. Moreover, microphones are mounted near the disc.

To have a more precise estimation of the range and variability of vibration instabilities at the origin of disc brake squeal, a series of tests with the fully instrumented TGV brake system are performed for different operating conditions. Effects of the variation of the speed before braking system (25 km/h and 60km/h), rotational direction of the disc (positive and negative rotations defined by Rot+ and Rot-, respectively) and compression force (8kN and 15kN) are more particularly undertaken.

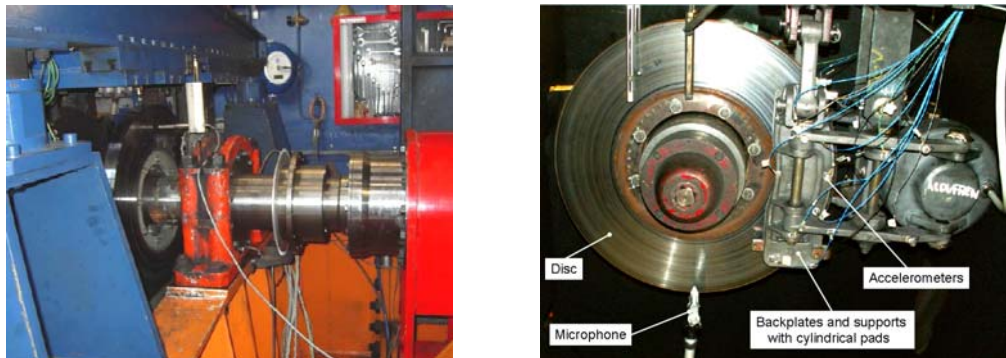


Fig. 2. Dynamic tests for TGV brake squeal

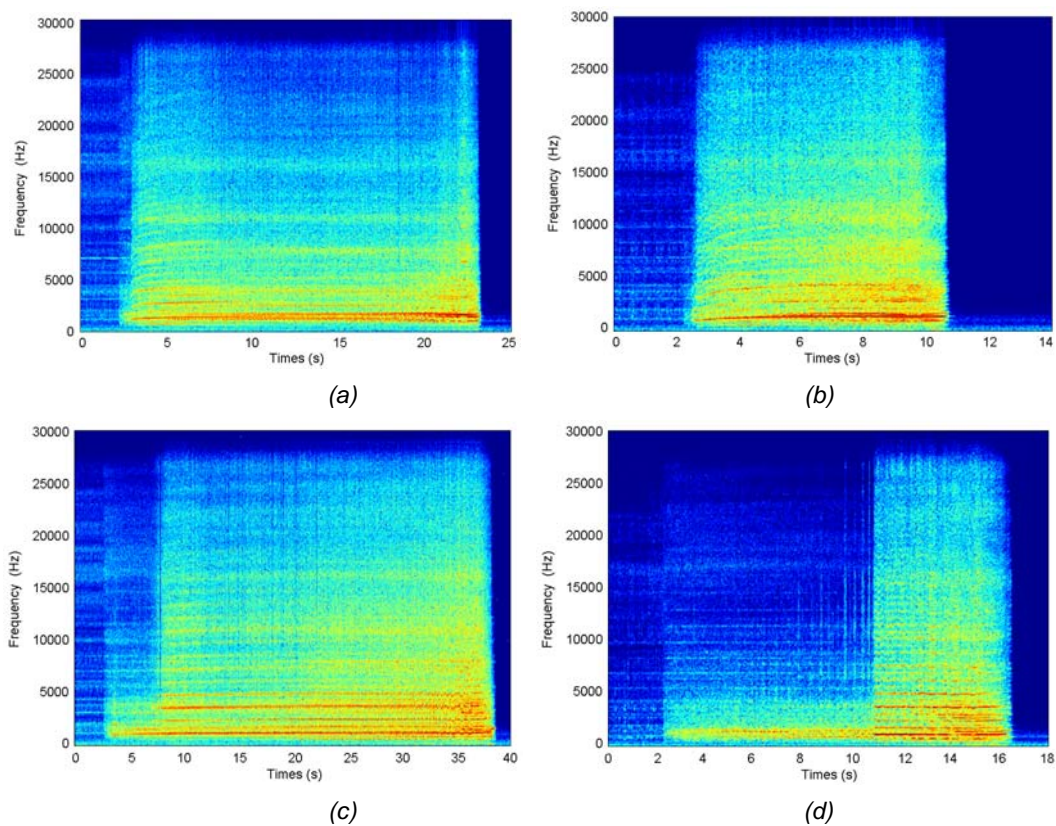


Fig. 3. Continuous Wavelet Transform of experimental data (a) 15kN- 60km/h (b) 15kN- 25km/h (c) 8kN- 60km/h (d) 8kN- 25km/h

The continuous wavelet transform (CWT) based on the Morlet mother wavelet is used to study the time-history responses. Experimental results (vibrometer measurement) for four operating conditions (15kN-60km/h, 15kN-25km/h, 8kN-60km/h and 8kN-25km/h with a positive rotation Rot+) are given in (Fig. 3). Moreover the repeatability of experiments is investigated by performing three identical tests for each deterministic operating condition. As shown in (Tab. 1), it appears that the response of the TGV brake system and the associated frequency content do not differ between three tests when a series of deterministic tests is performed in the same operating conditions.

Tests allow identifying two main complex nonlinear phenomena for TGV brake squeal due to the variation of the compression force. The first identified behavior is illustrated in (Fig. 3a) and (Fig.3b) (for 15kN- 60km/h and 15kN- 25km/h). The second one is given in

(Fig. 3c) and (Fig.3d) (for 8kN-60km/h and 8kN-25km/h). Even if the nonlinear transient response and the associated CWT are not identical, the frequency content of TGV brake squeal appears to be globally the same for the four operating conditions, as indicated in (Tab. 1): TGV brake squeal appears at low/middle frequency in the 0–10000 Hz range (with a predominant frequency content in the 0–5000 Hz range).

For the first identified behaviour of TGV squeal (15kN-60km/h and 15kN-25km/h, see (Fig. 3a) and (Fig. 3b)), only one characteristic dynamic behavior is identified. At the beginning of transient vibrations, an evolution and increase of the squeal frequencies is observed (see (Fig.3a) and (Fig.3b) between $t=[3 ; 4]$ s for 15kN-60km/h, and $t=[2.5 ; 5]$ s for 15kN-25km/h). Moreover, it clearly appears that all the transient non-linear oscillations can become complex with the contribution of several frequencies.

For the second identified behaviour of TGV squeal (8kN-60km/h and 8kN-25km/h, see (Fig. 3c) and (Fig. 3d)), two dynamic behaviours are observed: firstly, a "simple" behaviour of the transient oscillations with only two main frequency resonances around 1000-2000Hz (see (Fig.3c) and (Fig.3d) between $t=[2 ; 7]$ s for 8kN-60km/h, and $t=[2 ; 11]$ s for 8kN-25km/h); secondly, a "complex" non-linear transient behaviour with the appearances of new contributions in the 2000-10000Hz range (see (Fig.3c) and (Fig.3d) between $t=[7 ; 39]$ s for 8kN-60km/h, and $t=[11 ; 17]$ s for 8kN-25km/h). As explained in [9], evolutions of the transient vibrations and the frequency content of the TGV brake squeal are governed by the modification of the sliding non-linear equilibrium point (i.e. initial static position due to the compression force) during self-excited vibration. This may lead to new instabilities in the TGV brake system and induces a transition from one to the other behaviour.

Tests	N°	Frequency (Hz)										
		1200 ±100	1700 ±100	2150 ±75	2500 ±50	3400 ±100	3900 ±50	4600 ±100	5800	7150 ±25	8500	10700
8kN 25km/h Rot +	1	X	X	X	X	X	X					
	2	X	X	X	X	X	X					
	3	X	X	X	X	X		X				
8kN 25km/h Rot -	1	X	X	X	X	X	X	X	X			
	2	X	X	X	X	X	X		X	X		
	3	X	X	X		X	X					
8kN 60km/h Rot +	1	X	X	X	X	X	X	X	X			X
	2	X	X	X	X	X	X		X	X		
	3	X	X	X	X	X						
8kN 60km/h Rot -	1	X	X	X		X	X					
	2	X	X			X				X	X	
	3	X	X	X		X					X	
15kN 25km/h Rot +	1	X	X			X		X				
	2	X	X									
	3	X	X		X	X		X				
15kN 25km/h Rot -	1	X	X	X		X		X		X		
	2	X	X	X		X		X		X		
	3	X	X	X		X		X		X		
15kN 60km/h Rot +	1	X	X	X		X						
	2	X	X	X		X						
	3	X	X			X						
15kN 60km/h Rot -	1	X	X	X								
	2	X	X	X	X	X		X		X		
	3	X	X	X	X	X		X				
Occurrence %		100	100	79	54	92	33	38	8	21	8	4

Tab. 1. Experimental analysis of TGV brake system with squeal frequencies identification

3 Numerical simulation and comparison with experiments

3.1 TGV braking system and 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, as illustrated in (Fig.4). Then, the backplate and support are considered by adding the flexibility of these pads' supporting structures.

Considering the description of the nonlinear interface, a Coulomb law with a constant friction coefficient μ is used. This formulation can be summarized as follow:

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

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:

$$u_n - g \leq 0 \quad ; \quad r_n \leq 0 \quad ; \quad (u_n - g)r_n = 0 \quad (\text{eq. 2})$$

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 (see [10] for details):

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

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. The contact reaction \mathbf{r}_c , the displacement \mathbf{u} and the velocity $\dot{\mathbf{u}}$ verify the contact and friction laws defined in (eq. 1) and (eq. 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 ($proj_{K_\mu}$) is used to facilitate the numerical implementation in the treatment of the contact state [11]

$$\begin{aligned} r_n &= proj_{\mathbb{R}^-} (r_n - \rho_n^u (u_n - g)), \forall \rho_n^u \quad \text{where} \quad proj_{\mathbb{R}^-} (x) = \min(x, 0) \quad (\text{eq. 4}) \\ \mathbf{r}_t &= proj_{K_\mu} (\mathbf{r} - \rho_t (\dot{\mathbf{u}}_t - \mathbf{v}_g)), \forall \rho_t > 0 \quad \text{with} \quad proj_{K_\mu} (\mathbf{x}) = \begin{cases} \mathbf{x}_t & \text{if } \|\mathbf{x}_t\| / |x_n| \leq \mu \\ \mu \frac{|x_n|}{\|\mathbf{x}_t\|} \mathbf{x}_t & \text{otherwise} \end{cases} \quad (\text{eq. 5}) \end{aligned}$$

where ρ_n^u and ρ_t are two arbitrary positive scalars called normal displacement augmentation parameter and tangential augmentation parameter respectively [9].

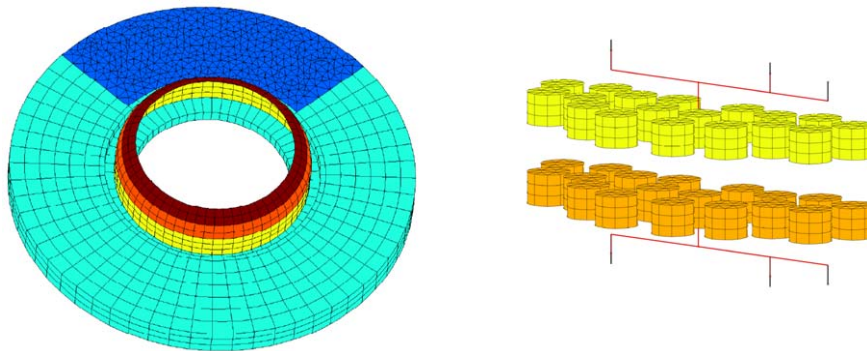


Fig. 4. Finite element model of TGV brake system

3.2 Stability analysis

In order to predict the occurrence of self-excited vibrations, a classical stability analysis can be performed. This approach can be divided into two parts. 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. This equilibrium point is obtained by solving the nonlinear static equations for a given net brake pressure. Then, one obtains the linearized equations of motion by introducing small perturbations about the equilibrium point into the non-linear equations [2, 8]. 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 [12]. The complex eigenvalues $\lambda = a + i\omega$ provide information about the local stability of the equilibrium point. The TGV brake system is stable if all the real part a of the eigenvalues are negative, and unstable if there exist one or more eigenvalues having a positive real part. The imaginary part of these eigenvalues represents frequencies of unstable complex modes that correspond to squeal frequencies.

The stability of the system is given on (Fig. 5). 9 unstable modes (with positive divergence) are detected. (Fig. 6) shows the mode shapes of these 9 unstable modes. We can see that the most unstable modes appear from pads modes (near 2050 Hz and 2760 Hz with a growth rate of 9.47% and 6.44% respectively).

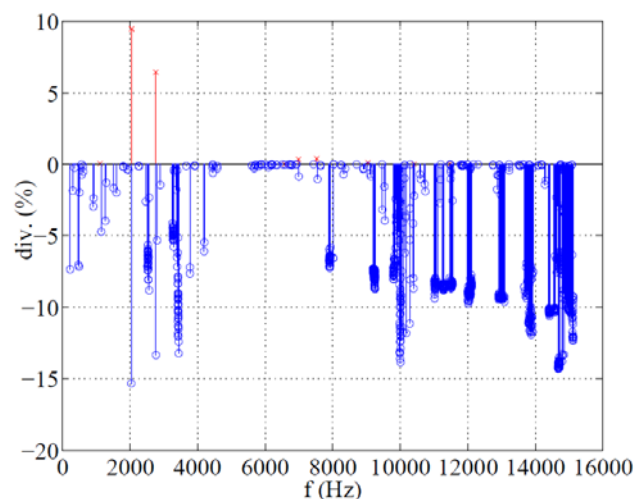


Fig. 5. Stability of TGV brake system (red: unstable modes, blue: stable modes)

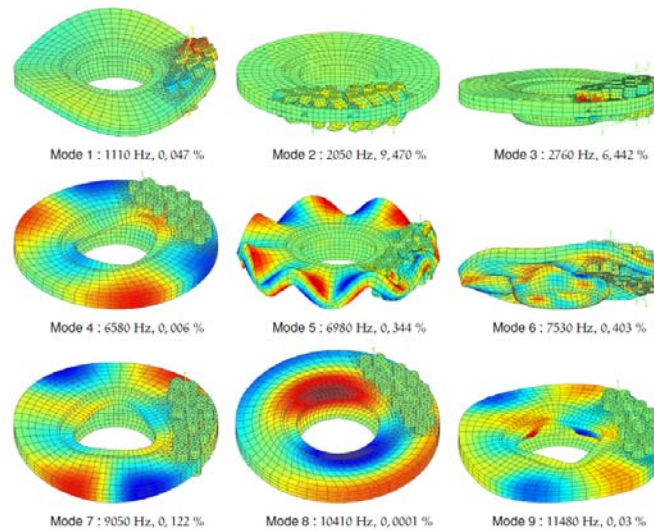


Fig. 6. Mode shapes of the unstable modes (with the frequency and growth rate)

3.3 Nonlinear self-excited vibration and comparison with experiments

As previously explained in [9] and [13], the stability analysis may lead to an under-estimation or an over-estimation of the unstable modes observed in the non-linear time simulation due to the fact that linear conditions (i.e. the linearized stability around an initial equilibrium point) are not valid during transient oscillations. So the non-linear transient self-excited vibrations can become very complex and include more or less unstable modes due to the non-linear contact and loss of contact interactions at the frictional interface. 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 [10] is developed for the computation of the transient solution.

A typical brake squeal spectrum obtained via numerical simulation is presented and compared with measurements in (Fig. 7). There is a good agreement, although slight differences may be noticed.

Moreover, non-linear squeal vibrations can become complex with appearances of new frequency peaks in the signals (in comparison with a stability analysis). For example, a new resonance peak is predicted near 4000Hz. It may correspond to the second harmonic component of the unstable mode at 2050Hz. This demonstrated that squeal is composed of not only fundamental frequencies of unstable modes (i.e. eigenvalues via stability analysis) but also harmonic components and new contributions due to the coexistence of several fundamental frequencies

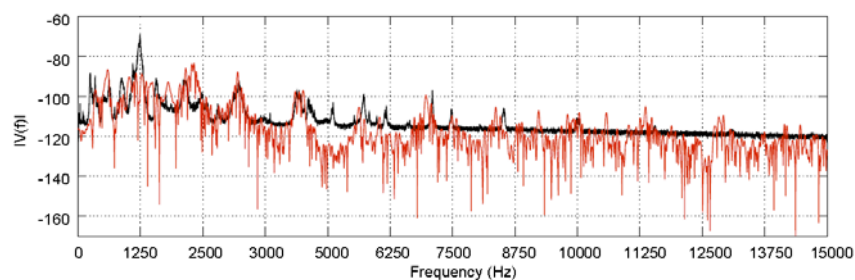


Fig. 7. Comparison between numerical simulation (black line) and experiments (red lines)

4 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.

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