Acoustics'08 Paris
June 29-July 4, 2008
www.acoustics08-paris.org

Numerical investigations into the squeal propensity of a railway disc brake

Olivier Chiello[a] and Xavier Lorang[b]

[a]INRETS, Transport and Environment Lab., 25 avenue François Mitterrand, 69675 Bron cedex, France
[b]SNCF, Innovative & Research Dept., 45 rue de Londres, 75379 Paris, France
olivier.chiello@inrets.fr

This paper falls within the scope of a research program concerned with the reduction of the squeal noise generated by high power railway disc brakes. It focuses on the numerical results provided by a finite element model of the brake including unilateral contact and Coulomb friction at the disc/pad interfaces. In particular, the dynamic stability of the sliding equilibrium is investigated by performing a complex eigenvalue analysis of the linearized equations verified by the structural displacement fields. Complex eigenvalues and complex modes are used to estimate the squeal propensity of the brake in a given frequency range. The effect of various mechanical and geometrical parameters is studied in order to better understand the mechanism leading to the system's instability.

1 Introduction

The noise generated by vehicles' brakes is one of the most difficult problems with brake systems. Research for predicting and removing brake noise has been regularly performed for many years [1]. Different kinds of noise are generated by disc brakes: low frequency noise (100-1000 Hz), like groans or moans, and medium or high-frequency noise (1-15 kHz), often called squeal, characterized by very annoying spectra with one or several pure frequencies. Despite great progress in the modelling of disc brake squeal, most of the noise abatement approaches are still individual and empirical.

Nowadays, it seems that the understanding of the physical phenomena arising in the squeal generation may lead to the development of technological solutions. In this paper, the modelling investigations performed within the framework of the French research program CRISFIS[1] are presented. In the first parts, the brake system and the main experimental results are briefly exposed. The modelling approach is then discussed. Finally, the results provided by the stability analysis of a simplified FE model of the brake are presented.

2 Description of the brake

The disc-brake system of TGV trains is mounted on a bogie as shown on figure 1. There are four discs on each axle. A disc brake is composed of two symmetric plates of lining with cylindrical pads which compress the disc. The discs are fixed on the axle by a hub (e.g. a very thin structure clamped on the axle).

Figure 1: The brake system (discs on the left, linings on the right)

3 Main experimental results

Measurements were carried out on rolling stock in controlled conditions. Different linings and discs were fitted with a microphone in the close field, a laser vibrometer aiming at the disc surface and some thermocouples and accelerometers on the linings. As seen on figure 2, the main squealing frequencies are above 6000 Hz. In this frequency range, it was found that the squeal noise is mainly radiated by the disc in axial vibrations (flexion, out-of-plane) .

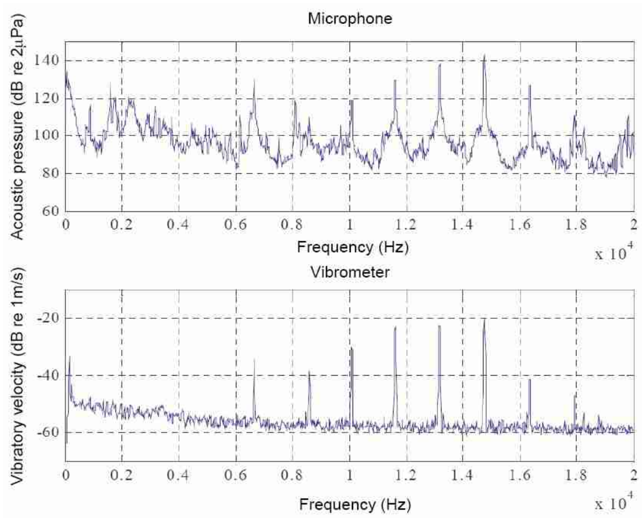

Figure 2: Acoustic and vibratory spectra of an instrumented disc

A test bench was used to characterize the vibrations field on the disc by laser measurements during braking in the laboratory. At frequencies above 6000 Hz, the measurements confirmed that the disc vibrates mainly axially. The measured vibrations fields are flexion waves, which may be interpreted as flexion modes rotating along the circumference of the disc. From 7000 to 18000 Hz, the identified modes have no nodal circles and nodal diameters varying from 9 to 15 (A0-9 to A0-15). At 6650 Hz, the mode is rather a circumferential in-plane mode with 2 nodal diameters (C0-2) and no nodal circle but with important axial deformations, due to the depth of the disc and Poisson's effect.

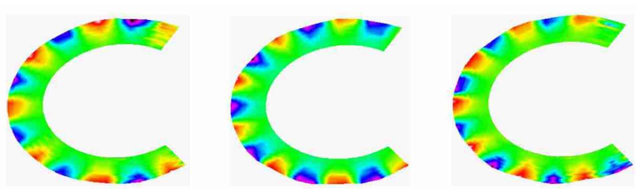

Figure 3: Vibration fields measured during squealing (modes A0-9, A0-10, A0-11)

[1]Squealing and Cracking of High Power Railway Disc Brakes

4 Modelling approach

The modelling approach is based on the main following assumptions:

- it is supposed that the squeal noise comes from the sound radiation of the structural components of the brake system, in particular the disc and the linings, in self-sustained vibrations,
- these self-sustained vibrations are supposed to be linked to the instability of the sliding equilibrium of the system,
- this instability is supposed to be the result of the non symmetric Coulomb friction law.

Compared with the braking duration, a short period of time is also considered. During this period, the rotating speed, the braking pressure and the friction coefficient are supposed to be constant and the wear and thermo-mechanical phenomena are neglected. Under these assumptions, the general problem of two elastic structures in unilateral frictional contact is considered. The formulation proposed by Moirot is used [2]. An overview of the approach is presented Figure 4.

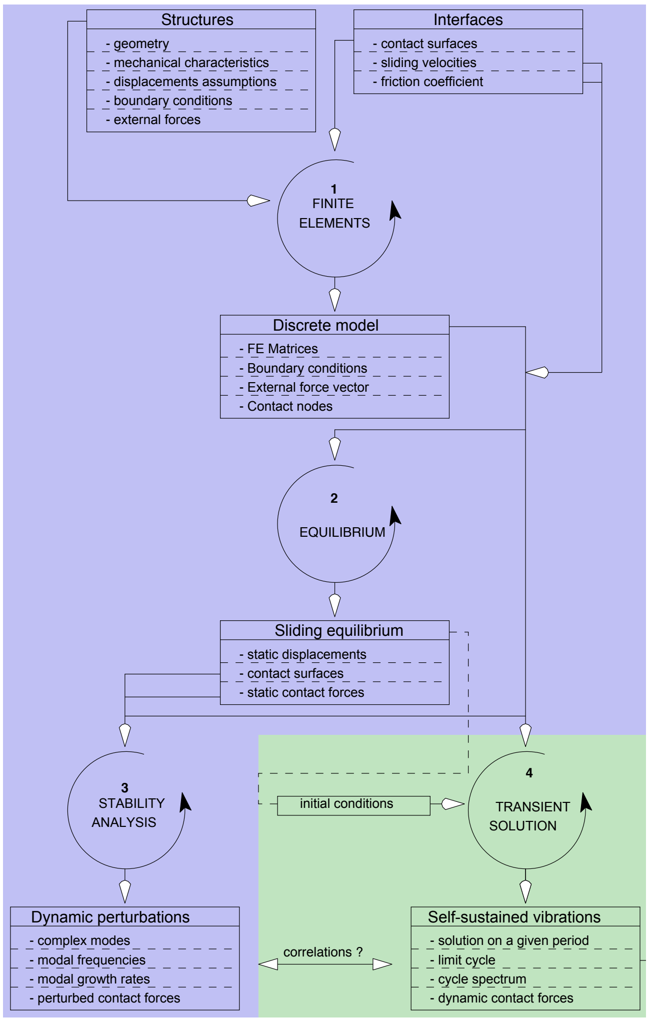

Figure 4: Overview of the modelling approach

The structural components are modelled by finite elements with compatible meshes at the interface. First, the sliding equilibrium is determined. This consists in finding a deformed state of the structure which satisfies the contact conditions. An iterative algorithm is used to solve this non linear problem. The stability of the equilibrium is then studied. This consists in finding the vibratory perturbations of the equilibrium, e.g. the solutions of the linearized equations of the problem. Practically, the complex modes and eigenvalues of a non symmetric linear system have to be calculated. The time evolution of the envelope of a complex mode is then linked 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 grows exponentially and is likely to generate self-sustained vibrations: the mode is unstable.

A common interpretation of the stability analysis is to suppose that the frequencies and the mode shapes of the main unstable complex modes correspond to the self-sustained vibrations. However, this hypothesis is uncertain and, moreover, does not give any useful information about the amplitude of these vibrations. So the resolution of the non linear dynamic problem is a necessary step is the unstable cases. This consists in finding the limit cycles to which the transient solution of the problem converges. In this paper, only the results provided by the stability analysis are presented (see [3, 4] for the non linear results).

5 Numerical results

5.1 FE model of the brake

Figure 3 shows the model of the brake in which only the disc, the hub and the cylindrical pads are taken into account. The elasticity of the pads support is neglected. This simplified model is sufficient to obtain instabilities well correlated with experimental results in the [6000–14000 Hz] frequency range. Viscous damping (Rayleigh) is also considered. The results of experimental modal analysis of the disc-hub couple and the pads were used to optimize the mechanical parameters (stiffness and damping) of the components (see table 1).

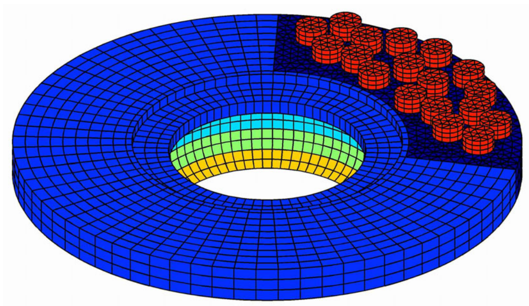

Figure 5: FE model of the disc/pads system

Disc/hub	
Young's modulus (Pa)	2.1×10^{11}
Poisson's ratio	0.28
Density (kg/m^{-3})	7850
Pads	
Young's modulus (Pa)	3.5×10^{9}
Poisson's ratio	0.3
Density (kg/m^{-3})	6250
Braking configuration	
Braking force (N)	4000
Rotation speed (rad s^{-1})	4
Friction coefficient	0.4

Table 1: Mechanical parameters of the system

5.2 Unstable complex modes

A stability analysis of the system without damping was performed. More than 800 modes were calculated to reach an upper limit frequency of 14 kHz. The growth factors of the unstable complex modes are represented figure 6. Two kinds of modes may be distinguished:

- the pad unstable modes (frequencies close to 4000 and 12000 Hz) for which the disc vibrations are very small (see figure 7),
- the disc unstable modes (frequencies between 6000 and 14000 Hz) for which the vibrations of the disc are large (see figure 8).

The corresponding mode shapes and frequencies of the disc unstable modes are close to the modes of the disc in free conditions but rotate along the disc (waves). Most of these modes are axial modes without nodal circles and with one nodal circle. Another mode (close to 6650 Hz) is the in-plane mode C0-2 for which axial components are high, however. Except the axial modes with one nodal circle, all these unstable modes are close to the experimental squeal frequencies. Non linear effects such as modes competition might explain this difference [3, 4].

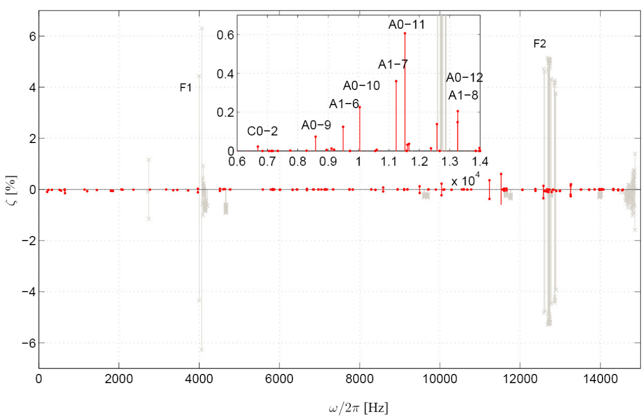

Figure 6: Growth rates of complex modes (pad modes in gray , disc modes in red)

It may be noted that the growth rates of the unstable disc modes increase when their frequencies get closer to the frequencies of the unstable pad flexion modes (F2 on figure 6).

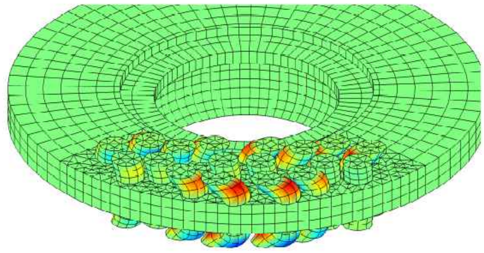

Figure 7: Unstable pad complex mode F1

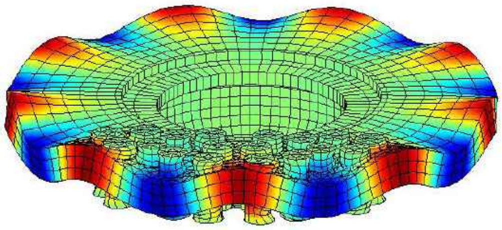

Figure 8: Unstable disc complex mode A0-9

5.3 Effect of the pad parameters

The effects of various pad parameters were investigated in order to give some practical recommendations to reduce squeal noise. The main results of this parameter study are presented in this section. The influence of the contact surface (pads' wear states), the pad Young's modulus and the pad depths are illustrated respectively on figures 9, 10 and 11. These results were obtained by using the parameters of table 1, except for the reference pad for which a Young's modulus of 2000 MPa and a density of 5000 kg/m^{-3} was chosen. On these figures, only growth rates corresponding to unstable disc modes are represented. In addition, figure 12 shows the stability diagram when some damping is added in the pads. This result is obtained for a pad Young's modulus of 3500 MPa and a pad density of 6250 kg/m^{-3}.

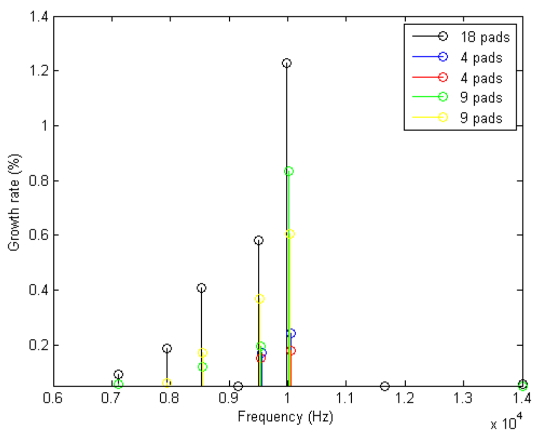

Figure 9: Effect of the number of pads in contact on stability

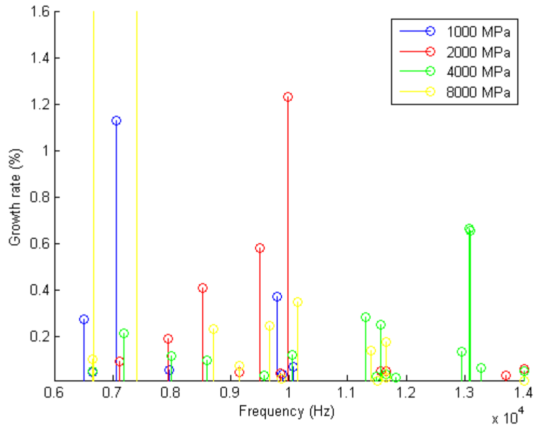

Figure 10: Effect of the pad Young's modulus on stability

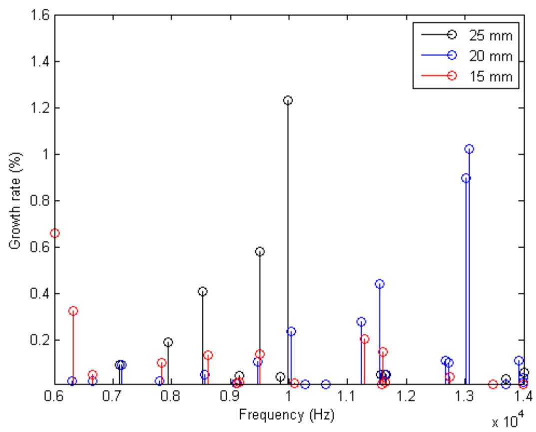

Figure 11: Effect of the pad depth on stability

Several conclusions may be drawn from these results:

- the growth rates of the unstable disc modes strongly decrease when the contact area decreases,
- the growth rates of the unstable disc modes increase when their frequencies get closer to the frequencies of the pad flexion modes (F2 on figure 6) so the influence of the pad Young's modulus and the pad depth is mainly explained by the shift of these pad flexion modes,
- when the Young modulus is equal to 8000 MPa or the depth is equal to 15 mm, a maximum of the growth rates may be observed between 6000 and 7000 Hz, which is probably due to the proximity with other pad modes (F1 on figure 6)),
- the pad damping stabilizes the disc unstable modes excepted when their frequencies are very close to the pad flexion modes.

This last remark suggests that there is two kind of instability, depending on the proximity of the frequencies of the disc modes and the pad modes (see [3] for more details).

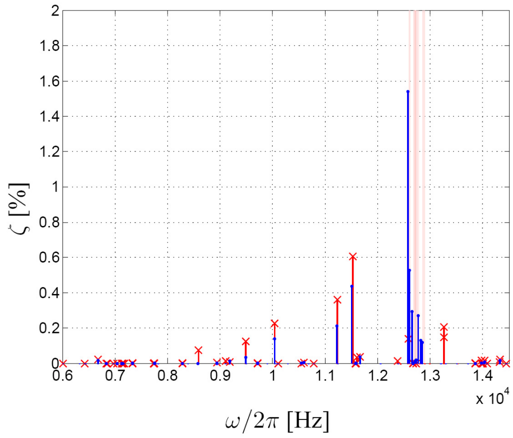

Figure 12: Effect of the pad damping on stability (without damping in red, with damping in blue, pad modes in light red)

6 Conclusion

In this paper, a numerical model of the squeal noise generated by railway disc brakes was presented. It was shown how the frictional coupling between the disc and the numerous cylindrical pads of the linings may lead to instabilities. Moreover, the unstable disc modes provided by the stability analysis of the system were found to be very close to the measured vibrations in terms of frequencies and mode shapes. A parameter study was also performed to determine the effect of the pads Young's modulus, the pads depth and the contact area on stability. These results showed the destabilizing effect of the pad flexion modes in the frequency range of interest. The effect of the pad damping was also analyzed and it was found how an increase of the damping could destabilize the system at some frequencies.

Acknowledgements

This work is part of the French PREDIT project CRISFIS concerned with the "Squealing and cracking of high power railway disc brakes". The authors wish to thank the ADEME for its financial support and all the other partners for their implication in the project.

References

[1] N. M. Kinkaid, O. M. O'Reilly, and P. Papadopoulos. Automotive disc brake squeal. *Journal of Sound and Vibration*, 267:105–166, 2003.

[2] F. Moirot. *Etude de la stabilité d'un équilibre de Coulomb - application au crissement des freins à disque*. Thèse de doctorat, Ecole Polytechnique, France, 1998.

[3] X. Lorang. *Instabilité vibratoire des structures en contact frottant : application au crissement des freins de TGV*. Thèse de doctorat, Ecole Polytechnique, France, 2007.

[4] X. Lorang and O. Chiello. Stability and transient analysis in the modelling of railway disc brake

squeal. In *Noise and vibration mitigation for rail transportation systems*, Notes on Numerical Fluid Mechanics and Multidisciplinary Design (NNFM) 99:447–453, Springer Berlin/Heidelberg, 2008.