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THE EFFECT OF MODAL DAMPING ON BRAKE SQUEAL INSTABILITY

Francesco Massi*[1], Aldo Sestieri[1], and Laurent Baillet[2]

[1] Department of Mechanics and Aeronautics, University of Rome "La Sapienza",
via Eudossiana 18, 00184, Rome, Italy
[2] LGIT, Laboratory of Geophysics and Tectonophysics
University Joseph Fourier, 1381, rue de la Piscine - Domaine Universitaire - 38400 Saint-Martin-D'Hères, Grenoble, France
francesco.massi@uniroma1.it

Abstract

"Brake squeal" groups a large set of high frequency sound emissions from brake systems, generated during the braking phase and characterized by a harmonic spectrum. The generation of squeal is due to an unstable behaviour of the system during the braking phase. Many researchers associate the squeal phenomena to the coalescence of two eigenfrequencies of the system: when two modes, characterized by large deformation at the contact surface, couple at the same frequency, one of them becomes unstable leading to increasing vibration.

The present experimental analysis is focused on correlating the propensity to instability of a system mode with its modal damping. Because of the complexity of real brake systems, a simplified experimental set-up is proposed. The set-up is a valid tool that permits to control easily and modify its dynamics. A clear distinction is obtained between the dynamics of the friction pad and the caliper. Different conditions of unstable coupling between a mode of the brake disc and a mode of the caliper or the pad are analyzed.

Two opposite roles of the modal damping are detected. A large modal damping can either prevent the rise of squeal instabilities or enlarge the squeal propensity of the brake apparatus. Moreover, the add of damping, aimed to prevent a specified squeal condition, can cause the rise of different squeal events.

INTRODUCTION

Squeal noise is a difficult subject partly because of its strong dependence on many parameters and partly because the mechanical interactions in the brake system are very complicated, including nonlinear contact problems at the friction interface. Considerable efforts e.g. [1-2] were directed to investigate brake squeal and different approaches to the problem were proposed; among them, four main possible mechanisms of friction-inducing noise in disc brake systems are reported in the literature. This are the stick-slip, the sprag-slip [3], the negative friction-velocity slope [4] and the modal coupling mechanisms [5]. The modal coupling, in particular, is actually the most accepted: the squeal instability is predicted by the "lock-in" [6] between two system modes that start to have the same frequency until one of them becomes unstable. The complex eigenvalues analysis is a popular numerical tool for the prediction of squeal instability [2]. In [7] the squeal events, reproduced with the experimental set-up that is utilized in this work, are clearly identified as modal instabilities and they are obtained when the coincidence in frequency between a couple of suitable modes exists. This coincidence is referred below as frequency "tune-in"

This paper is addressed to describe the effect of the modal damping of the unstable modes on the squeal propensity. Modal damping is also introduced to prevent specific squeal frequencies. The analysis is performed on a simplified brake set-up that allows a simple control and modification of its dynamics.

EXPERIMENTAL SET-UP

A simplified experimental set-up is preferred to a real brake because the last is characterized by a geometry and dynamics that can be hardly controlled and understood. The set-up consists in a rotating disc (the brake disc) and a small friction pad pushed against the disc by weights positioned on the pad support that is used to represent the brake caliper (figure 1).

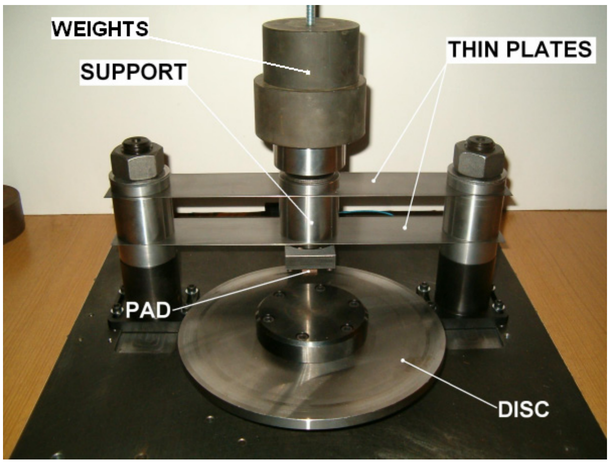

Figure 1- Experimental set-up

The disc is made of steel (internal diameter 100 mm, external diameter 240 mm, thickness 10 mm) and is assembled to the shaft by two hubs of large thickness that insure a rigid behaviour of the connection. An electric motor rotates the disc. The disc velocity can be adjusted between 5 and 100 rpm. Brake pads are made of commercial friction material, obtained by machining standard brake pads. Reduced pad dimensions are adopted to simplify the dynamics of the pad that can be adjusted by varying its dimensions. The support (central cylindrical body in the figure) is also made of steel and its shape is chosen to have for it a simple dynamics. The normal force between pad and disc (braking pressure) can be varied, by adding weights on the top of the support, between 25 and 225 N. The support itself weighs 25 N. Two thin plates hold the pad support in the tangential direction. This solution permits to have a low (zero in non deformed vertical condition) stiffness in the normal direction and high stiffness in the tangential direction, necessary to oppose the friction force. To add damping into the system, rubber layers can be introduced between the body of the support and the thin plates.

SET-UP DYNAMICS

Because of its key role in the squeal occurrence, the dynamics of the set-up is identified and followed during the whole experimental campaign. A preliminary EMA (Experimental Modal Analysis) is performed on the assembled system, i.e. when the pad is in contact with the disc. The small contact surface allows to have a small coupling between the tree main substructures: disc, support and pad. The dynamics of the assembled can be considered as the sum of the dynamics of the substructures. In particular, the attention is focused on three sets of system modes that are considered to be the cause [8] of the squeal instability: the bending modes of the disc (normal direction with respect to the friction surface), the bending modes of the support (tangential direction with respect to the friction surface) and the tangential modes of the friction pad.

The bending modes of the disc are characterized by nodal diameters and nodal circumferences: the (n,m) mode of the disc is characterized by n nodal circumferences and m nodal diameters. The disc is characterized by an axial symmetry: therefore the modes of the disc are generally double modes. When put in contact with the pads, the disc looses its axial symmetry, so that the modes of the disc are no longer double modes and they split [9].

The following notation is used to name the splitted modes:

- mode (n,m-) a nodal diameter is coincident with the contact point;
- mode (n,m+) an antinode is coincident with the contact point.

The support modes are analyzed by a SIMO (Single-Input-Multi-Output) analysis, exciting the support in the tangential direction, close to the contact area. In the frequency range of interest two rigid and three bending tangential modes of the support are recognized. This analysis is performed on the coupled system, when the disc does not rotate. However, same peaks in frequency and same deformed shapes are found during brake simulations. The second and the third mode of the support are characterized by the largest tangential displacement at the contact area, and because of this they are the only modes of the support involved in the squeal [7]. The second mode is a rigid rotational mode of the support, while the third mode is the first bending mode of the support.

Table 1 lists the natural frequencies of the system obtained with the EMA when a normal load of 25 N is applied and a friction pad with 10X10 mm contact surface is mounted.

Table 1 – System natural frequencies and modal damping.

MODE	FREQUENCY [Hz]	HYSTERTICAL DAMPING %	MODE	FREQUENCY [Hz]	HYSTERTICAL DAMPING %
I support	489	7,27	(0,5+)	5589	0,32
(0,1+)	925	5,36	(1,1)	7217	2,35
(0,2-)	1425	1,67	V support	7717	0,75
(0,2+)	1625	3,69	(0,6)	7725	0,26
II support	2091	0,72	(1,2)	8025	3,06
(0,3-)	2317	1,18	(0,7+)	10088	0,37
(0,3+)	2458	2,02	(0,7-)	10141	0,38
III support	2912	3,99	(1,4)	12367	1,07
(1,0)	3058	2,11	(0,8+)	12725	0,27
(0,4-)	3750	1,31	(0,8-)	12825	0,17
(0,4+)	3808	0,67	(1,5)	15283	0,58
IV support	5146	2,09	(0,9+)	15517	0,54
(0,5-)	5575	0,49	(0,9-)	15708	0,13

The third substructure to investigate is the friction pad. Also its dynamic is easily recognizable in the assembled dynamics. Figure 2 shows the PSD (Power Spectral Density) of the pad acceleration in the tangential direction during brake tests without squeal. The first three peaks in frequency correspond to three support modes. The others two peaks at 4 and 11.1 kHz correspond to modes of the pad. A further FEM modal analysis developed with Ansys allowed to identify two pad modes characterized by deformation in the tangential (friction) direction, at the same frequencies obtained experimentally. The tangential modes of the pad are characterized by a large modal damping, due to the large material damping of the friction material.

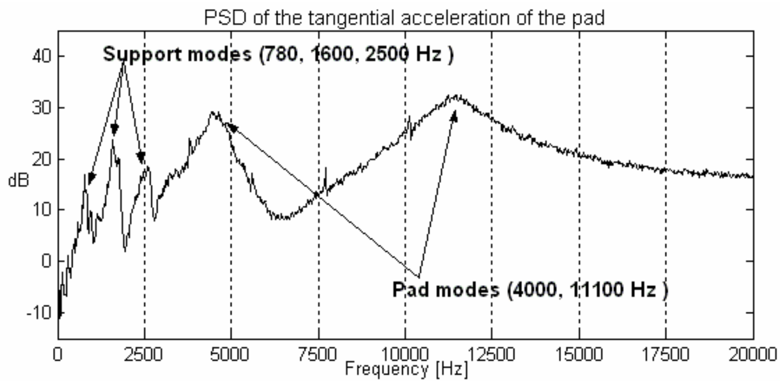

Figure 2 - PSD of the tangential acceleration of the pad during brake phase without squeal.

MODAL DAMPING AND SQUEAL

In [7] the different squeal frequencies, reproduced on the brake set-up, are reported and described. In particular two different coupling conditions, that lead to squeal, are identified: coupling between a bending mode of the disc with a bending mode of the support, and coupling between a bending mode of the disc and a tangential mode of the pad. The necessary conditions for the unstable coupling is the "tune-in" of the natural frequencies and a large displacement (normal for the disc, and tangential for the pad or support) at the contact area [9].

Squeal propensity and modal damping

An important distinction between the "disc-pad" and the "disc-support" modal coupling is a larger modal damping of the modes of the pad with respect to the modes of the support. Five different squeal frequencies are found: 1566 Hz, 2467 Hz, 3767 Hz, 7850 Hz and 10150 Hz. The squeals at 1566 and 2467 Hz are obtained when a mode of the support couples with a mode of the disc; squeals at 3767 Hz, 7850 Hz and 10150 Hz are obtained when a mode of the pad couples with a mode of the disc.
Unstable coupling between a mode of the support, i.e. characterized by low modal damping, and a mode of the disc happens only when an exact tune-in between the two natural frequencies occurs. Figure 3 shows the PSD of the tangential acceleration of the pad when the parameters are settled to have exact tune-in between the natural frequency of the third mode of the support and the natural frequency of the (0,3+) mode of the disc (black line). In this case squeal occurs. The dashed line shows the PSD when the dynamics of the system is modified, by displacing the weights at the top of the support, and the two mentioned natural frequencies differ for about 40 Hz. In this case squeal does not occur.

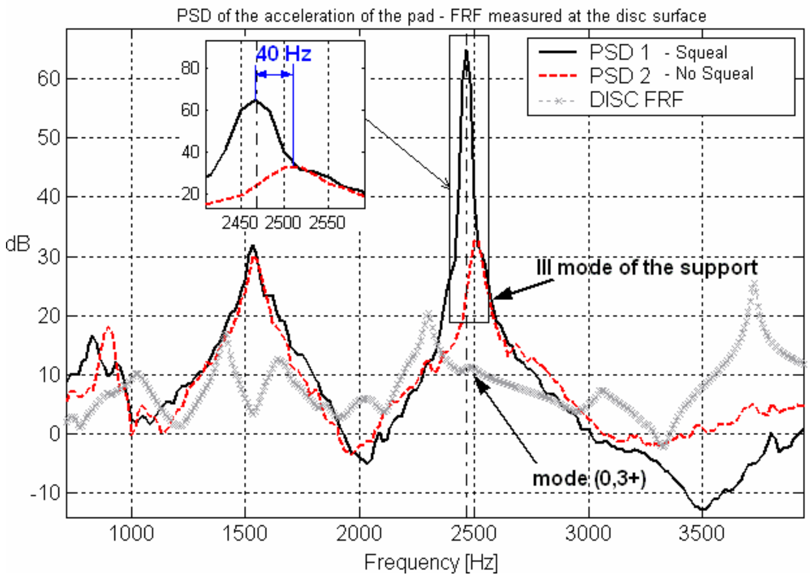

Figure 3 – Squeal due to a "support-disc modal coupling" occurs only for exact tune-in of the natural frequencies.

On the contrary, a coupling between a mode of the pad, i.e. characterized by large modal damping, and a mode of the disc can occur also when the two natural frequencies do not coincide, but are close enough. Figure 4-a and 4-b show the behavior of the system when the parameters are settled to have tune-in (3767 Hz) between the first mode of the pad and the (0,4+) mode of the disc. The time behavior of the acceleration of the pad (figure 4-a) shows that squeal starts as soon as the contact between pad and disc happens (≈2s). The natural frequency of the pad coincides with the natural frequency of the disc, i.e. the squeal frequency (figure 4-b). When the normal load is changed from the tune-in value, from 45 N to 48 N, the brake test starts in silent condition (without squeal), but the instability can be still excited by an impulse. In fact, when a hammer impulse (≈3s) is applied to the disc surface, squeal starts and does not stop (figure 4-c). For this value of the normal load the PSD of the pad acceleration (measured during braking in silent condition) shows that the natural frequency of the pad differs for about 40 Hz from the one of the disc (figure 4-d).
When the normal load increases up to 70 N squeal starts after the impulses and stops after few seconds (figure 4-e). In this condition the two natural frequencies differ for about 300 Hz (figure 4-f).
By increasing the normal load up to 120 N, squeal is not excited even if the impulse is applied to the system (figure4-g). With these values of the parameters the two natural frequencies differ more than 400 Hz (figure 4-h).
It is worth to notice that when squeal occurs and there is not exact tune-in between the two natural frequencies, it develops at the natural frequency of the mode with lowest modal damping (the mode of the disc in the present case).

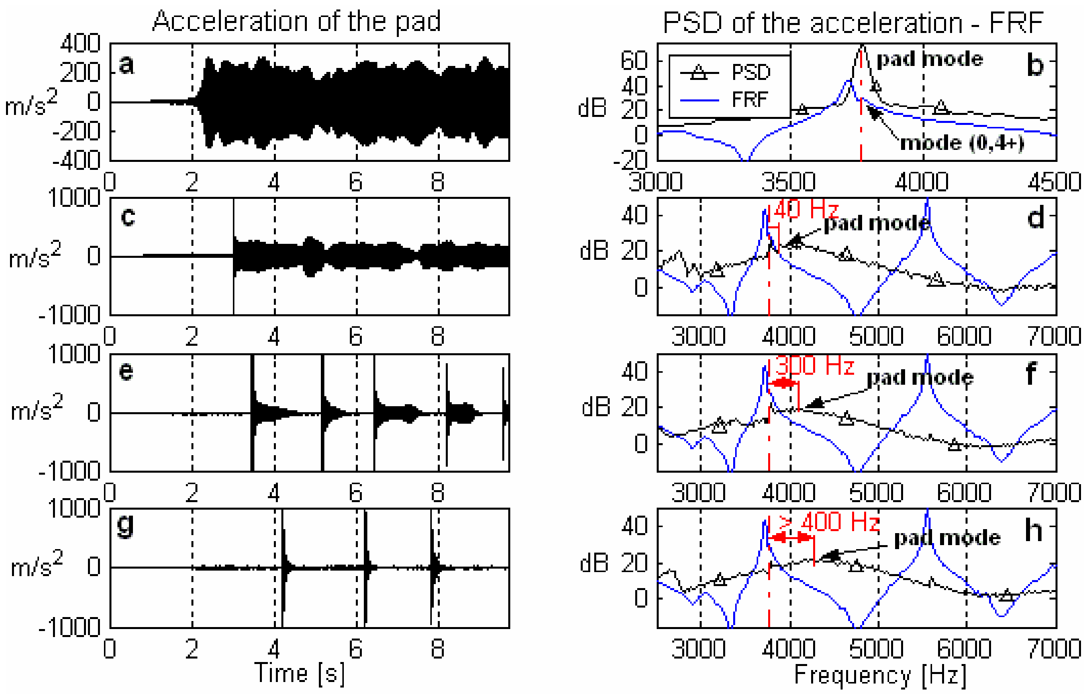

*Figure 4- Time and frequency behavior of the system during brake tests with normal load equal to: (**a** – **b**) 45 N;(**c** – **d**) 48 N; (**e** – **f**) 70 N; (**g** – **h**) 120 N.*

In conclusion a larger modal damping enlarges the range of frequencies where the two modes couple to give squeal instability. In fact, the unstable coupling is obtained not only because of the largest response of the system at its resonance frequencies, but, specially, because of the phase of the response of the mode when it is excited close to its natural frequency. A larger modal damping enlarges the frequency band where the phase changes. Moreover, squeal develops at the natural frequency of the disc because it is characterized by the smallest range where the phase shift is present (smallest modal damping).

Squeal suppression by modal damping

The previous section describes how a large modal damping increases the squeal propensity of the brake system. Therefore, the introduction of a really high modal damping allows to reduce vibration at resonance, and hence, to prevent the rise of squeal at the frequency of the damped mode.
In order to stop or prevent squeal events, thin layers of rubber are introduced between the support and the thin-plates. The rubber introduces a high modal damping to the third mode of the support (2.7 kHz) that is characterized by its bending deformation. The response of the mode is extremely reduced (from 28 dB to 8 dB). The introduction of modal damping prevents the rise of squeal obtained at the same frequency. In fact, once a damping layer is introduced with the same values of the parameters (105 N, 10x15 mm, 0.5 mm) that lead to the unstable coupling between the third (bending) mode of the support and the (0,3+) mode of the disc, no squeal is obtained (figure 5-a). It is useless to modify the parameters to search for a coupling between the two modes. On the contrary, when the values of the parameters are settled to have coupling between the first mode of the pad and the (0,4+) mode of the disc, the introduction of the rubber layers does not affect the rise of squeal (3767 Hz), that occurs as easily as without rubber (figure 5-b). In fact the rubber layers do not introduce modal damping either into the modes of the pad or into the modes of the disc.

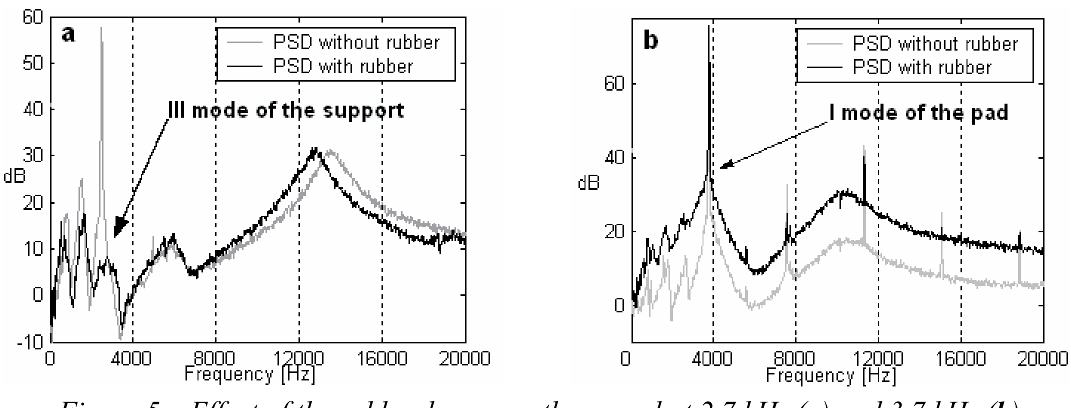

*Figure 5 – Effect of the rubber layers on the squeal at 2.7 kHz (**a**) and 3.7 kHz (**b**)*

A further effect related to the introduction of rubber layers is the shift of the modes of the pad toward lower frequencies, due to the decrease of the tangential stiffness at the connection between the pad and the support. This effect shifts the natural frequency of the second mode of the pad from the tune-in frequency with the (0,7+) mode of the disc (squeal at 10150 Hz) into the tune-in frequency with the (0,6+) mode of the disc (squeal at 8850 Hz). Thus, the introduction of the rubber layers prevents a specific squeal frequency, but has no effect on the others. Moreover it creates conditions to have a more unstable modal coupling.

CONCLUSIONS

This paper presents different squeal conditions obtained with a simplified brake apparatus. The dynamics of the system is related to the squeal occurrence. In particular two different squeal conditions, involving modes of the pad and mode of the support, are analyzed.

Two different roles of the modal damping are described:

- the introduction of a really high drastic modal damping permits to prevent the squeal instability that involves the damped mode. However this expedient does not affect the squeal instabilities that involve other modes of the system.
- high modal damping enlarges the range of frequencies where the high damped mode can couple with other modes of the system. This can be explained by the phase function of the system response that is smoother for higher modal dampings.

In conclusion, particular attention should be placed when introducing damping into the system to prevent squeal. A not correct intervention can either cause the tune-in between more modes of the system by changing its dynamics, or enlarge the range of frequencies where the damped mode can couple with other modes of the system.

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