Experimental analysis on squeal modal instability

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Abstract

In this paper, an experimental analysis performed on a simplified brake apparatus is presented. Brake squeal is a major concern in braking design. During past years a common approach for squeal prediction was the complex eigenvalues analysis. Squeal phenomenon is treated like a dynamic instability. When two modes of the brake system couple at the same frequency, one of them becomes unstable leading to increasing vibration. The presented experimental analysis is focused on correlating squeal characteristics with the dynamic behavior of the system. The experimental modal identification of the set-up is performed and different squeal conditions and frequencies are reproduced and analyzed. Particular attention is addressed to the system dynamics in function of the driving parameters on squeal occurrence. Squeal events are correlated with the modal behavior of the system in function of the main parameters, like contact pressure, friction material properties and system geometry. The robustness of the obtained squeal events permits a further analysis on the triggering of the squeal instability during braking, including the values of parameters that bring to instability. The obtained results agree with the modal coupling approach for squeal prediction, and confirm the characterization of squeal as dynamic instability.

Introduction

Disc brake noise continues to be object of investigation for automotive manufacturers and researchers. Because of the complexity of the problem and the need of estimating the squeal tendency during brake design, many analytical and numerical [1] approaches have been proposed. The modal coupling approach (mode lock-in) between two system modes, proposed by Akay & all [2], is one of the most accepted. The complex eigenvalues analysis is a popular numerical tool for squeal instability prediction [3-4]: squeal propensity is quantified by the dynamic instability of certain system modes. This paper shows an experimental analysis aimed to validate this approach for squeal prediction and to identify squeal phenomenon as a modal instability of the system. Brake squeal is a strongly non-linear phenomenon, characterized by friction material and contact non linearities. The modal approach allows to predict the rise of squeal when it is still in linear conditions. The paper uses a simplified set-up that is particularly appropriate to identify and eventually modify its dynamic behaviour.

A description of a modal analysis of the set-up is first presented. The dynamics of the system is studied by considering separately three main sub-structures of the brake(calliper, disc and pad). Different ways to shift the mode frequencies of the system are used to find different instability conditions. All squeal conditions are related to an appropriate dynamic configurations of the system with particular values of the parameters. Some comments and remarks are finally reported.

Experimental rig

Since a real brake apparatus is characterized by geometry and dynamics that can be hardly controlled and understood, an experimental and theoretical study of a simplified experimental set-up is preferred. The set-up consists in a rotating disc (the disc brake rotor) and a small friction pad pushed against the disc by weights positioned on a rigid support (figure 1).

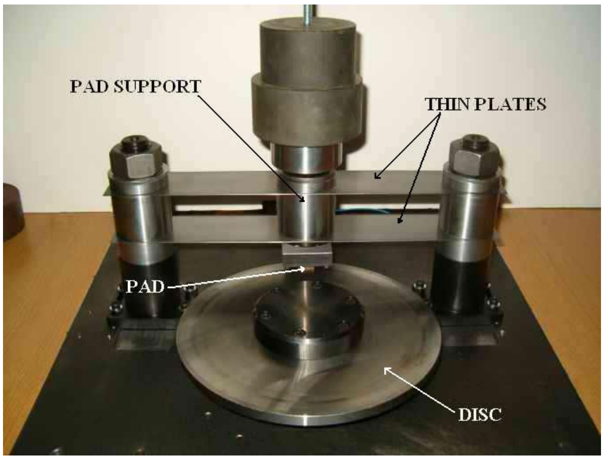

Figure 1- Experimental set-up.

The disc is made of steel (internal diameter 100 mm, external diameter 240 mm, thickness 10 mm) and is assembled with the shaft by two hubs of large thickness that insure a rigid behaviour of the connection, in the frequency range of interest. The velocity of the DC motor can be adjusted to have a disc velocity between 5 and 100 rpm. The transmission line consists of a pair of pulleys connected by a rubber toothed belt. The brake pads are made of commercial brake friction material, obtained by machining standard brake pads. Reduced pad dimensions are adopted to simplify and easily control the dynamics of the pad, by changing its dimensions. The support (the central cylindrical body in the figure) is also made of steel and its shape is chosen to simplify its dynamic behavior. The normal force between pad and disc (braking pressure) can be adjusted by adding weights on the top of the support, between 25 and 250 N. The support weighs 25 N. Adjusting the normal load with weighs placed on the pad support, that is not constrained in the vertical direction, allows the pad surface to follow the disc oscillations that are due to a not perfect planarity of the disc, and assures a constant value of the imposed normal force. Two thin-plates hold the pad support in the tangential direction. This solution permits to have a low stiffness (zero in non deformed vertical condition) in the normal direction and high stiffness in the tangential direction, necessary to oppose the friction force. A tri-axial force transducer is placed between pad and support. The transducer allows to measure the time history of the normal and friction forces. It is important to note that these forces are not measured on the real contact surface, but above the brake pad. Thus, the pad dynamics influences the measured forces.
The whole sep-up is designed to have a simple dynamics that can be analyzed and modified to obtain different squeal conditions. A simple dynamics of the set-up allows to follows its behavior when changing the driving parameters and to relate the dynamics to the rise of squeal.

Set-up dynamics

The main objective of this paper is to show how the dynamics of a brake system influences squeal occurrence and squeal frequencies. Therefore, the investigation of the set-up dynamics is the first step of the work. Particular attention is focused on the bending modes of the disc (in the normal direction) and the bending modes of the support and the pad (in the tangential direction). In fact we have seen in previous experiments [4] that these are the modes involved in squeal phenomena. Particularly, both FEM analysis and measured FRFs (Frequency Response Functions) in the in-plain direction show that in-plain modes of the disc are not involved in the squeal phenomenon.

Three different substructures are considered in the analysis: the disc, the support and the pad. The dynamics of the assembled system can be analyzed by the combination of the dynamics of the disc and the support, because of the reduced contact surface between the two substructures and the consequent low coupling between them. Therefore, we will refer to the modes involving bending vibration of the disc as “disc modes”, being the larger part of the energy concentrated on the disc. As well, we will refer to the modes involving bending vibration of the support as “support modes”. A further analysis allows to recognize the influence of the pad dynamics on the dynamics of the assembled system.

The disc modes 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. Due to the contact with the pads, the disc loses its axial symmetry. Therefore, when the structure is assembled together (contact force from 25 to 225 N), the modes of the disc are no longer double modes and they split at two different frequencies (figure 2).

We use the following notation to name coupled system modes:

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

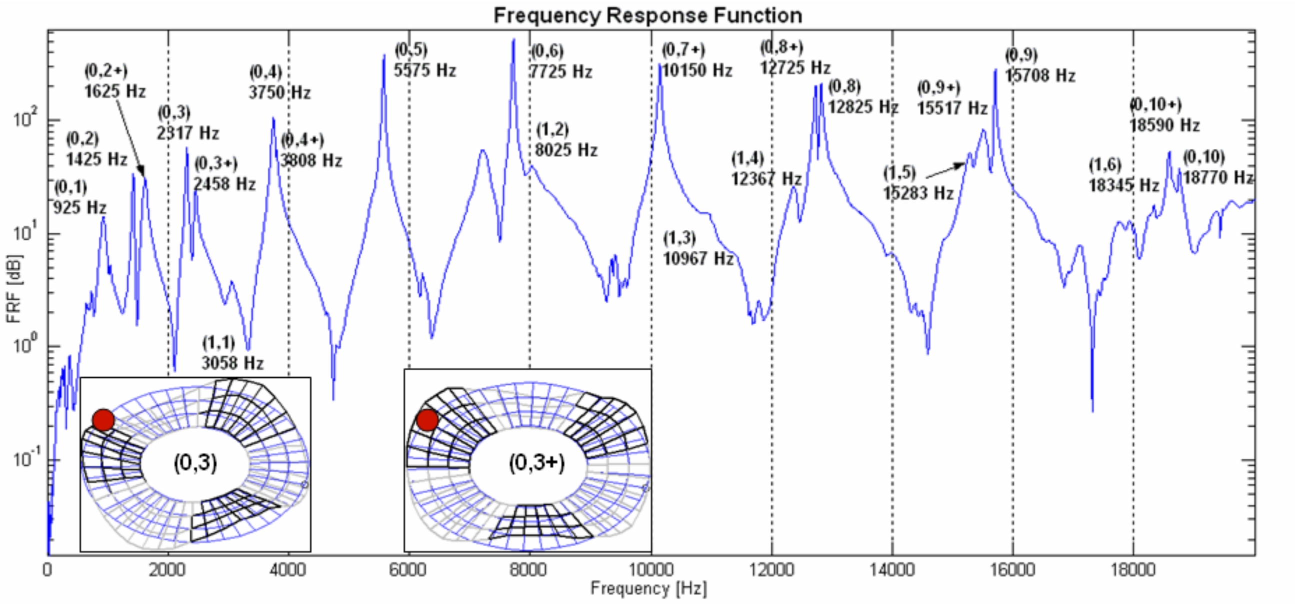

Figure 2 - FRF of the set-up measured on the disc periphery, when a normal of 225 N is applied, and samples of the disc deformed shapes (0,3) and (0,3+). The big node is the contact surface with the pad.

The support modes are analyzed by a Single-Input-Multi-Output analysis, exciting the support in the tangential direction, close to the contact surface, when no weights are placed on the top of the support. This analysis is performed without rotating the disc. However, the three peaks in frequency are found during brake simulations. The effect of the disc rotation results in a decrease of the mode frequencies. Lower frequencies can be explained considering that the thin-plates holding the support have a side compressed during brake simulation. This effect introduces a lower stiffness in the support substructure that brings to lower frequencies. Figure 3 shows the deformed shapes of the support in the tangential direction.

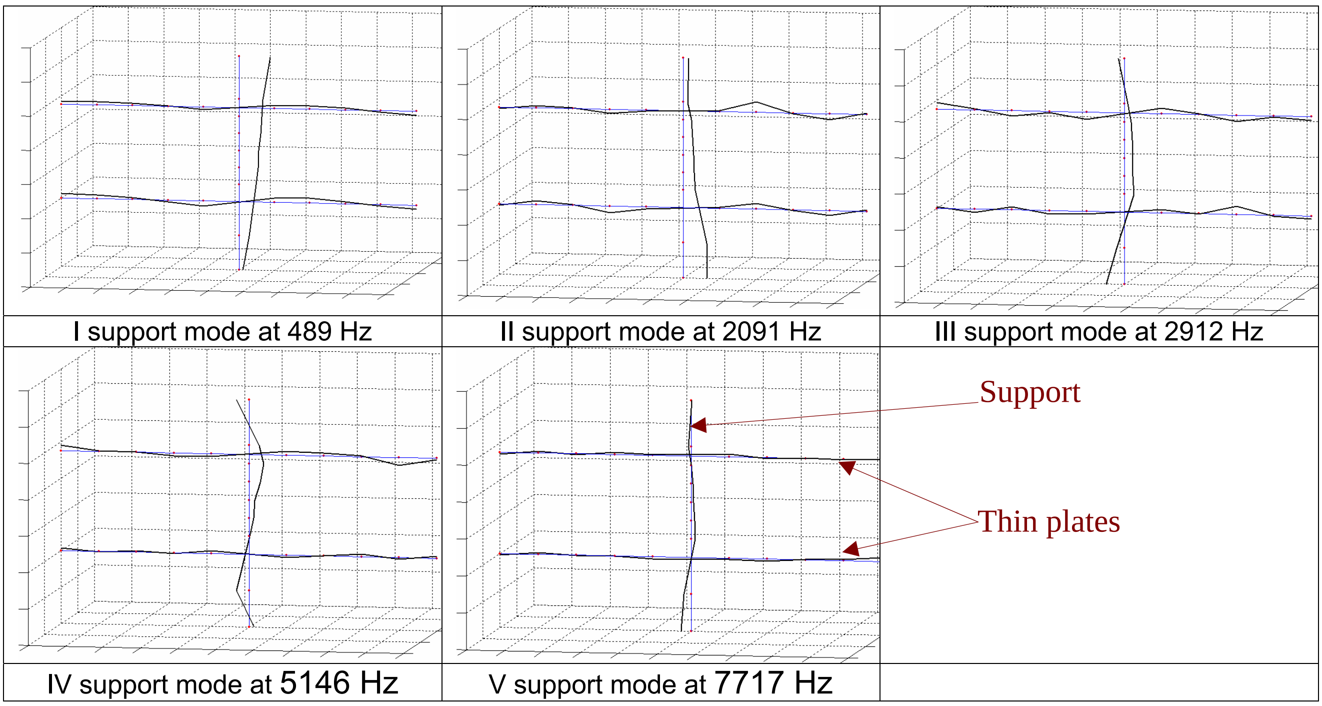

Figure 3 – Deformed shapes of the support obtained experimentally.

Table 1 lists the natural frequencies of the system obtained by EMA (Experimental Modal Analysis), when no weights are added on the top of the support, and a friction pad with 10X10 mm contact surface is mounted. The normal load is equal to 25 N.

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	(1,0)	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

Table 1 – System natural frequencies and modal damping.

The third substructure to investigate is the friction pad. Its dynamic is easily recognizable in the assembled dynamics. Figure 4 shows the PSD of the pad acceleration in the tangential direction during brake simulation when a normal load of 225 N is applied (grey line). The first three peaks in frequency are three support modes. The others two peaks at 4.5 and 11.5 kHz correspond to modes of the pad. A second test was made by dragging the pad on a rigid surface, disassembled from the disc and the support. The black line in figure 4 shows the acceleration PSD during this test. Only the two peaks related to the pad modes appear. A FE modal analysis allows to identify two pad modes characterized by the pad deformation in the tangential direction, at the same frequencies obtained experimentally.

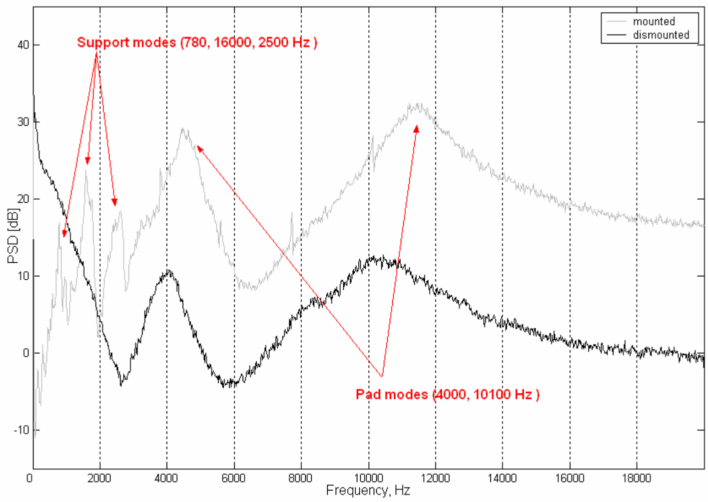

Figure 4 - PDS of the pad acceleration during brake simulation (grey), and in disassembled condition (black).

Dynamics modulation

In order to pilot the dynamics and, in particular, the modes frequencies, different driving parameters are chosen: the load applied on the top of the support, the friction pad dimensions, the stiffness of the thin-plates, insertion of damping material between the thin-plates and the support.

Figure 5-a shows the split values of the disc modes when 20 Kg are placed on the top of the support (225 N of normal load). Figure 5-b shows the split increase with the increase of the contact force for the modes with respectively two and three nodal diameters: the red line (circles) is the FRF of the disc when there is no contact with the pad, and the blue one (asterisks) is the FRF with maximum load on the top of the support. By changing the load, the mode frequencies of the disc move. It is interesting to notice that the modal split is

positive for low frequency modes, almost zero for the six nodal diameters mode, and negative for high frequency modes. We define the split positive when the (n,m+) mode has a higher frequency than the (n,m) mode. This behavior is due to the mass and stiffness effects introduced by the contact with the pad. The (n,m) mode has the contact point belonging to the nodal diameter so that it is not influenced by the contact and its frequency remains almost the same. The (n,m+) mode has the contact point in the antinode of the disc. For low frequency modes the add of stiffness due to the contact stiffness with the pad and support has more influence than the add of modal mass, and the natural frequency increases. The influence of the mass effect increases by increasing the frequency, and for higher frequency modes the natural frequency decreases with the increase of the load. In another experimental analysis [4] only a positive split was obtained, because the contact load was introduced with a mechanism involving mainly an add of stiffness. In the experimental set-up, object of this study, the contact pressure is obtained by gravity due to a huge mass placed on the top of the support. In this case, not only an add of contact stiffness but also an add of mass must be considered.

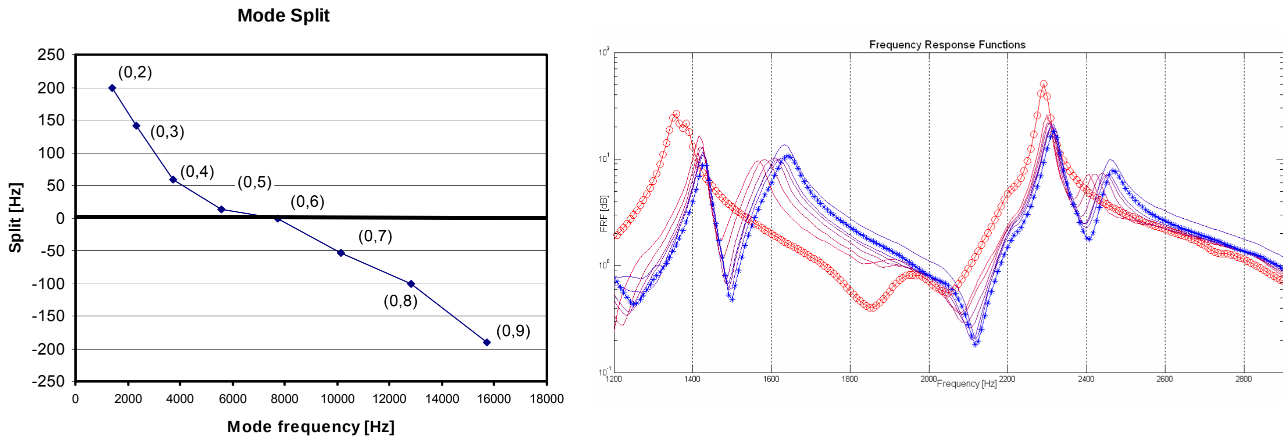

Figure 5 - a) Split of the disc modes with 225 N of normal load; **b)** Split of two and three nodal diameters modes in function of load: the red line (circles) is the FRF with no contact with the pad, the blue line (asterisks) is with maximum load.

Obviously, adding weighs on the top of the support changes the support natural frequencies. Moreover, because of the strong dependence of the properties of the friction material on the stressed state of the material itself, and because of the increase of the contact stiffness with the load, the mode frequencies of the support increase with the applied load. Table 2 shows the frequencies ranges where the modes of the support and pad fall, being the normal load varied from 25 N to 225 N. The contact surface of the pad is equal to 10X10 mm, and thin-plates thickness is equal to 0.5 mm.

MODE	FREQUENCY RANGE [Hz]
II support mode	1500 - 2100
III support mode	2400 - 3150
I pad mode	3750 - 4500
II pad mode	10100 - 12000

Table 2 – Support and pad frequency range for different load conditions

Table 2 shows only the second and third support modes, because only these two modes of the support are involved in the squeal phenomena. This can be explained looking at the deformed shapes of the support modes in figure 3. The second and third modes are the modes involving largest vibrations of the support at the bottom end, where it is in contact with the disc. The dynamic instability is due to the coupling between one normal mode of the disc and one support or pad mode. The coupling happens in the contact surface, because of the relationship between normal and friction forces. Therefore, the larger the modal deformation at the contact point, the easier is the rising of instability.

Increasing the thin-plates thickness from 0.5 to 1 mm increases the stiffness of the support mode and the natural frequency of the second and third mode increases by about 200 Hz. The thickness of the thin-plates does not affect the frequencies of the disc and pad modes. By adopting different contact surface dimensions (8X8 10X10, 10X15, 10X20), the frequencies of the pad modes can vary between 3700 and 5000 Hz for the first mode and between 10000 and 14000 for the second one.

Squeal phenomena

The previous section describes the detailed dynamic analysis performed in function of the main driving parameters. An extensive experimental campaign was then performed to find as many possible squeal frequencies by modulating the parameters values. During this campaign the dynamics of the system was followed to relate its variation to the rising of instabilities.

Five different squeal frequencies are found: 1566 Hz, 2467 Hz, 3767 Hz, 7850 Hz and 10150 Hz. These squeal conditions are obtained for well defined values of the driving parameters, and all of them are easily reproducible. Figure 6 shows the system behaviour during the squeal event at 3767 Hz, obtained with 45 N of normal load, contact surface equal to 8X8 mm and thickness of the thin-plates equal to 0.5 mm. The disc velocity was maintained at 10 rpm and the support was leaned on the disc to start the brake simulation. Consequently the normal and friction forces show a starting ramp due to the initial contact between pad and disc. In figure 6 a slow non-physical decrease of their mean values is observed due to the discharging of the capacitive transducer. The global friction coefficient is equal to 2.8. After the first contact, the system vibrations start to increase as shown by the time history of the in-plain acceleration of the pad and by the increase of sound pressure vibration. The PSD of the pad acceleration and the sound pressure level reveal the harmonic nature of the system vibrations at 3767 Hz. The parameter values are adjusted to have squeal instability. As soon as the contact happens the instability rises and the vibrations grow up to a maximum value. The small variations of the vibration amplitude, after its stabilization, are due to the non uniformity of the disc roughness.

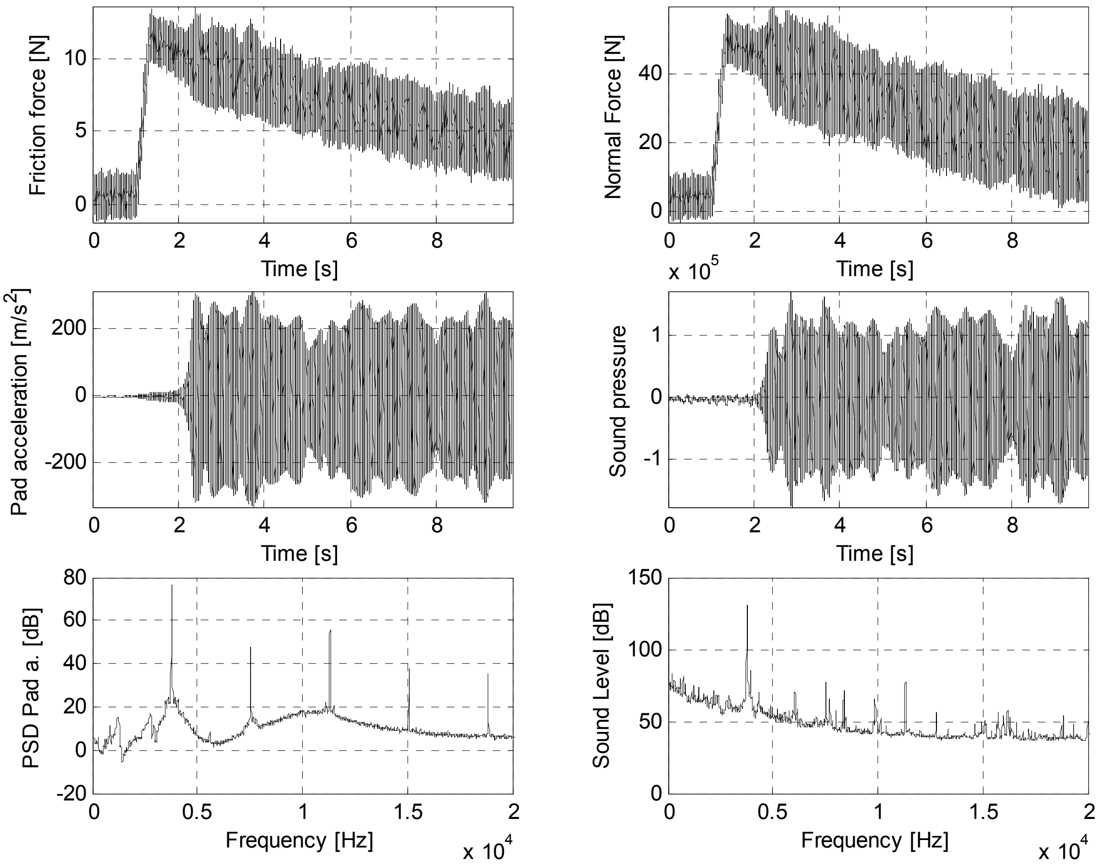

Figure 6 – System vibration during squeal phenomenon at 3767 Hz.

Figure 7 shows the PSDs of the pad acceleration in the tangential force direction for different values of the normal load from 250 N to 30 N. The grey line is the FRF measured on the disc periphery with normal load equal to 30 N. The amplitude of the peak related to the pad mode decreases with the decrease of normal load, because also the pad excitation in the tangential direction, due to the friction force, decreases proportionally to the normal load. By lowering the normal load the first mode of the pad moves to lower frequencies and it gets close to the (0,4+) mode of the disc. This can be explained by the lower Young modulus value of the pad material and by the lower contact stiffness with the disc. It is important to notice that only when the pad mode is enough close to the disc mode squeal happens. In figure 7 only the red line (PSD for 30 N of normal load) presents a peak in frequency, characteristic of the squeal phenomenon. The same behaviour is noticed for the other four squeal frequencies obtained during this experimental campaign. Similar plots can be obtained varying other parameters. This means that we can have squeal instability only when a coincidence of two system modes is obtained, as predicted by the complex modal analysis and by the lock-in theory. Moreover we obtain squeal only when a disc mode, characterized by bending vibrations, couples with a mode of the pad or the support, characterized by vibration in the tangential direction. Precisely, we have squeal at 1566 Hz when the II support mode frequency coincides with (0,2+) disc mode frequency, squeal at 2467 Hz when the III support mode frequency coincides with (0,3+) disc mode frequency, squeal at 7850 Hz when the II pad mode frequency coincides with (0,6+) disc mode frequency, squeal at 7850 Hz when the I pad mode frequency coincides with (0,4+) disc mode frequency, squeal at 10150 Hz when the II pad mode frequency coincides with (0,7+) disc mode frequency.

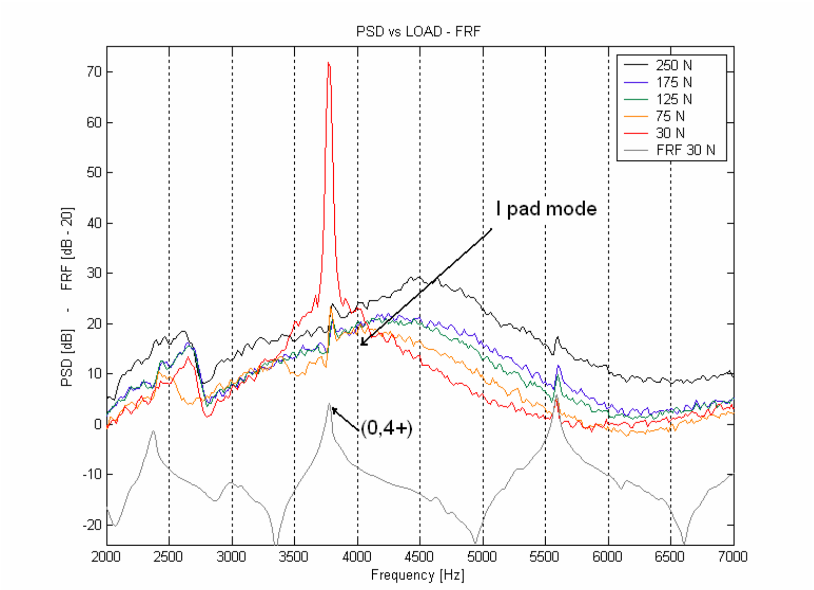

Figure 7 – PSD of pad acceleration in function of normal load, and FRF in squeal condition.

It is worth to underline the clear distinction obtained between the two substructures that present modes characterized by tangential vibrations along the contact surface: pad and support (calliper). This study shows that the disc dynamics can couple indifferently with the dynamics of the pad or with the dynamics of the support, bringing in both cases to squeal instabilities. This means that an effective "squeal free" design should take into account both the pad and calliper dynamic interaction with the disc.

Figure 8 shows all the disc modes involved in squeal events (on the FRF of the disc), and the frequency ranges covered by the support and pad modes at variation of the driving parameters. As expected, all the normal loads of the disc in the range of frequencies covered by the support and pad modes are involved in instability and they become unstable when their frequencies coincide with the respective support or pad mode frequencies. The (0,5+) mode frequency of the disc doesn't fall in any range covered by the pad and support tangential modes and, consequently, squeal does not occur.

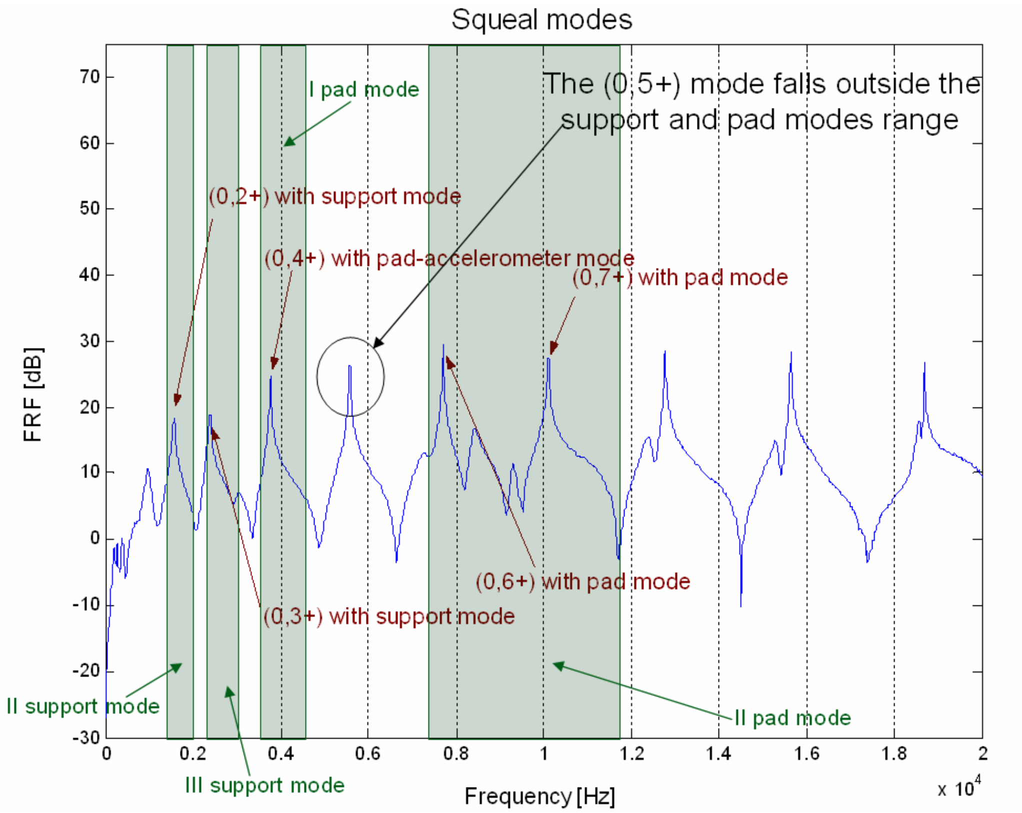

Figure 8 – Disc, pad and support modes involved in squeal instability.

Squeal triggering

During brake simulation several squeal frequencies are found, for defined parameters values. All the squeal events are obtained for modal coincidence of a normal mode of the disc and a tangential mode of the pad or support. This agrees with the lock-in theory that attributes the phenomenon to a dynamic instability, due to the coalescence of two system modes, one of which becomes unstable.

This section brings a further proof that squeal phenomena are dynamic instabilities related to the dynamics of the mechanism. During brake simulation in "silent" conditions (without squeal noise), but with the parameters values set to have instability, an impulse was given at the disc surface in the normal direction, with a hammer. The impulse was given in the opposite side of the contact with the pad, in order to excite all the modes with an antinode at the contact point. Figure 9 shows that, as soon as the impact happens, squeal starts and goes on until the disc is stopped. Following the FFT of the pad acceleration during the test, three different key-steps are identified:

- Phase 1: before the impulse, an almost white noise with only two low peaks due to the pad modes is acquired;
- Phase 2: during the impulse, the FFT shows all the peaks due to the bending modes of the disc with an antinode at the contact surface;
- Phase 3: after the impact, only the frequency peak (squeal) coincident both with a bending mode of the disc and with a mode of the pad is observed, and doesn't decrease.

It is needless to specify that, when the driving parameters are set to values that do not allow squeal, an impulse on the disc has not any effect.

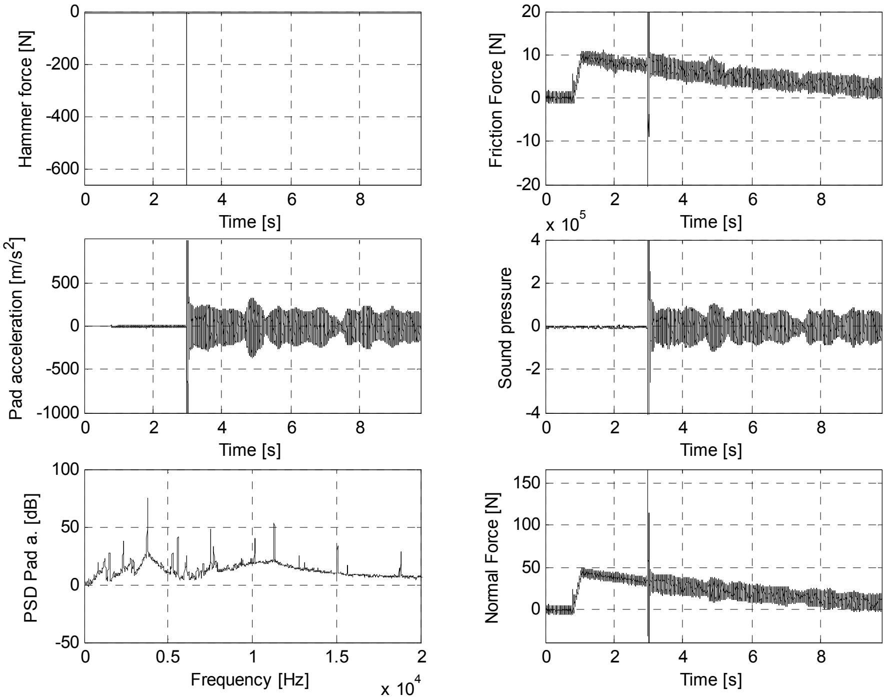

Figure 9 – Squeal vibration starts after an impulse normal to the disc surface at t ≈ 3s. The system parameters are set to have squeal instability.

Excitation of squeal instability can be also obtained by exciting the pad in the tangential direction by touching the disc surface in a point of the contact circumference with a humid wad. By this expedient it is possible to obtain an impulse in the friction force (figure 10) when the humid point passes under the pad. Using this kind of excitation, phases 1 and 3, described above remain the same, while in phase 2 only the frequency peak related to the disc mode, that is close to the pad mode, increases. Afterwards, the squeal peak goes on like in the previous case (phase 3).

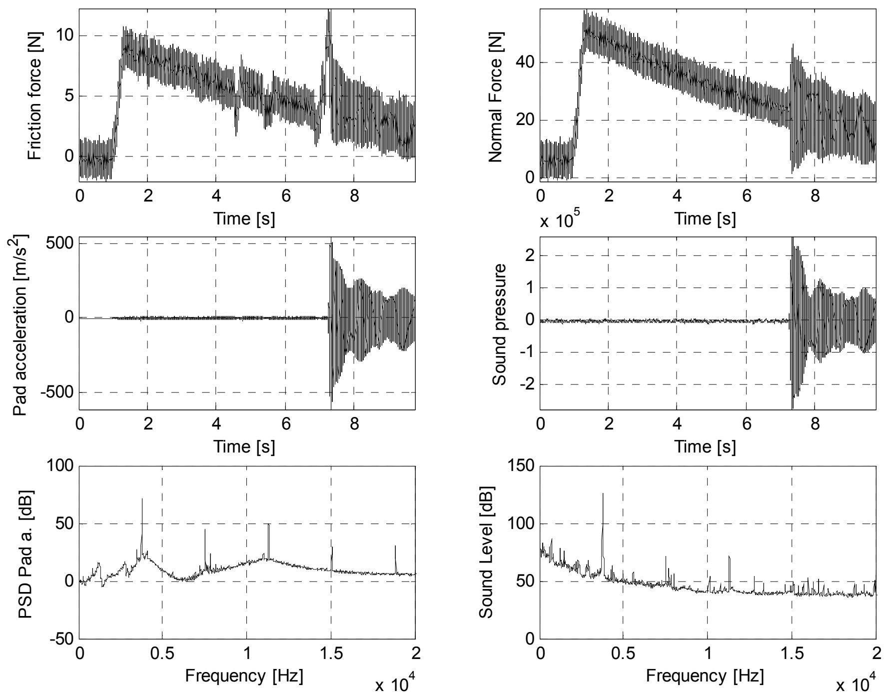

Figure 10 – Squeal vibration starts after an impulse (t ≈ 7s) in the friction force.

Conclusions

The behaviour of the proposed experimental set-up and the measurements presented in this paper agree with the mode lock-in theory and the complex eigenvalues analysis for squeal prediction. This means that we can predict the rise of a non-linear phenomenon, characterized by material and contact non-linearities, by a linear numerical tool. In fact squeal starts in linear behaviour, and only during its rise it becomes strongly non-linear. The dynamic analysis, parallel to the squeal campaign, clearly relates the squeal phenomenon with a system mode instability. In particular, squeal happens when a mode, characterized by large tangential vibrations of the pad surface couples with a mode characterized by large bending vibration of the disc in the contact point.

An important distinction between pad and support (calliper) dynamics must be underlined. Experiments show that squeal can be obtained both from modes coupling between disc and pad and modes coupling between disc and calliper. This suggests that an effective “squeal-free” brake design should take into account both these phenomena.

A further analysis shows that the squeal instability can be easily triggered when the system dynamics is favourable. It is important to note that a brake apparatus in real operating conditions is subjected to several circumstances that can trigger the instability. Therefore, during brake events, in order to have the squeal instability it is sufficient to reach the presented dynamic condition, i.e. coupling between support or pad mode with a bending disc mode.

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