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Contact surface topography and system dynamics of brake squeal

Francesco Massi [a,*], Yves Berthier [a], Laurent Baillet [b]

[a] *LaMCoS, INSA-Lyon, CNRS UMR5259, F69621, France*
[b] *Laboratoire de G´eophysique Interne et Tectonophysique, CNRS, Universite´ J. Fourier, BP 53, 38041 Grenoble Cedex 9, France*

Brake noise is an example of noise caused by vibrations induced by friction forces. During brake operation, the friction between the pad and the disc can induce a dynamic instability in the system. Brake squeal can occur in the frequency range between 1 and 20 kHz, and more. Approaches to this problem are usually confined to the analysis of the system dynamics or to the tribological studies (analysis of the contact surfaces). In a previous dynamic analysis, the squeal events treated in this paper were identified to be dynamic instabilities of the brake system. This paper presents an experimental tribological analysis performed in parallel to the dynamic one and aimed to highlight the role of the contact problems in the squeal phenomena. The topography of the contact surface after braking phase with and without squeal is presented. The features of the third body and the surface topography were found to be completely different in the two cases. After squeal events the topography of the surface and the internal structure of the pad material close to the contact show several cracks and material exfoliations; on the contrary, after braking without squeal the contact surfaces were found to be smooth and compact. The analysis of the system vibrations during squeal and the numerical analysis of the contact stresses allow to relate these features to the fatigue excitation due to the local oscillation of the contact stresses at the contact surface at the squeal frequencies. The surface topography after squeal, by revealing several fatigue evidences, points out a further problem, i.e., the possible increasing of wear rates that are consequent to the squeal vibrations.

1. Introduction

Friction-induced vibrations and noise emanating from brake disc 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 non-linear contact problems at the friction interface. Kinkaid et al. [1], in their extensive review paper on brake squeal phenomenon, concluded that "despite a century of developing disc brake systems, disc brake squeal remains a largely unresolved problem". Akay [2] asserted that warranty costs, due to noise and vibration in brake systems, overcome one billion dollars each year just in North America.

Squeal involves the dynamics of the whole system as much as the local dynamics at the contact surface, including the STTs (Surface Tribological Transformations) and the role of the third body (particles, coming from the pad and the disc, constitute the third body and have a key role on the behaviour of the contact triplet [3,4]).

Mills [5] and Fosberry and Holubecki [6] tried to correlate the occurrence of squeal with a negative slope of the coefficient of friction versus relative velocity curve, while Spurr [7] proposed his sprag-slip theory, followed later by Earles [8] and Jarvis and Mills [9]. Akay et al. [10], Tucinda et al. [11] and Allgaier et al. [12], conducted extensive analyses of the beam-on-disc set-up. Among others approaches, the modal coupling [10,13,14] (mode lock-in) between two system modes is one of the most accepted and the complex eigenvalues analysis of the brake system is a popular numerical tool for squeal instability prediction [15].

Different approaches have been adopted by the tribological analysis, in which the attention is focused on investigations (after brake events with squeal) of the contact surfaces of both the pad and the disc [16,17]. Thus the analysis focuses mainly on the STT study and on the role of the third body.

Rhee et al. [18] hypothesized that squeal can be due to an effect of "local hammering" at the contact surface that excites a mode of the brake system. Chen et al. [19] explained the "local hammering" as material detachments or asperity formations and deformations. This tribological behaviour would be reached in the advanced phase of braking, when the friction coefficient reaches a larger value than the starting value. The critical friction coefficient found by Bergman et al. [17] is also related to local contact phenomena appearing after a consistent number of braking events. In their experimental analysis Chen et al. [20] showed also a coincidence between the squeal frequency and a natural frequency of the system.

* Corresponding author. Tel.: +33 4 72437962; fax: +33 4 78890980.
E-mail address: francesco.massi@insa-lyon.fr (F. Massi).

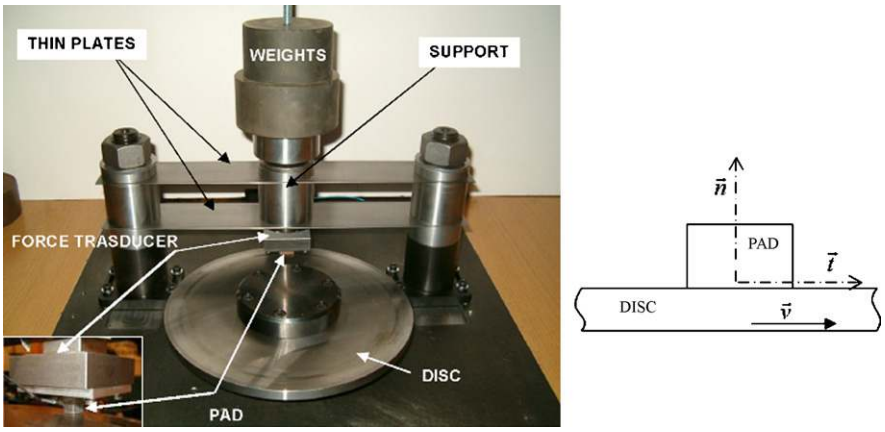


Fig. 1. Left: experimental set-up, TriboBrake COLRIS (Collaboration Lyon-Rome for Investigation on Squeal). Right: schema of the contact.

The squeal phenomenon involves the dynamics of the whole system as much as the local dynamics at the contact surface. This paper compares the wear characteristics of a brake system when breaking with and without squeal vibrations. The aim is to show how the dynamic behaviour of the brake affects the contact topography.

First, the simplified brake set-up, object of this analysis, is presented. A previous experimental [21] and numerical [22] analysis identified squeal to be a dynamic instability of the brake system, due to the modal coupling of two opportune modes of the system. The local behaviour of the contact stresses is simulated by the numerical transient analysis [22].

The analysis of the tribological triplet [3] is discussed. A preliminary analysis of the contact between the pad and the disc surfaces is performed to characterize the third body. The behaviour of the system with and without squeal is then resumed.

A comparative investigation (mainly post-mortem) on the characteristics of the contact surfaces is carried out under both silent and squealing conditions, i.e., after the braking phase with and without squeal, revealing a possible increase of wear rates due to squeal. An analysis of the internal status of the pad material is presented too.

Finally, STTs and the third body are correlated with the behaviour of the local contact stresses obtained by the numerical simulations and with the dynamic behaviour of the brake.

This paper present part of a research developed on simplified brake systems that allows for reproducing squeal events due to the modal coupling between two modes of the system [21,22,23,24]; the results reported below are thus referred to this kind of squeal instability.

2. Experimental rig

2.1. Description of the set-up

A simplified experimental set-up has been designed for this analysis because a real brake system is characterized by a geometry and dynamics that can be hardly controlled and understood. In fact, squeal is a phenomenon extremely sensitive to the system dynamics so that a minimum change in the boundary conditions can modify the squeal occurrence. The proposed set-up allows to have consistent and reproducible squeal events. Moreover, the simple geometry allows to analyze the behaviour of the system modes in function of the driving parameters [21,22,23,24]. The set-up consisting of a rotating disc (the disc brake rotor) and a small friction pad pushed against the disc by weights positioned on a floating support (Fig. 1).

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 with large thickness that insures a rigid behaviour of the connection. Brake pads are made of commercial brake friction material, obtained by machining standard brake pads. Reduced pad dimensions (8 mm $\times$ 8 mm $\times$ 10 mm) have been adopted, in order to simplify and easily control the dynamic behaviour of the pad. This solution allows studying the relationships between squeal and system dynamics; moreover, it allows for a robust reproducibility of squeal events, which is difficult to obtain with commercial brakes [21].

The support (central cylindrical body in Fig. 1) is also made of steel and its shape is chosen to simplify its dynamic behaviour. The normal force between pad and disc (braking pressure) can be adjusted, by adding weights on top of the support, between 25 and 225 N. Introducing normal load with weighs placed upon the pad support, that is not constrained in the vertical direction, allows the pad surface to follow the disc oscillations, due to its not perfect planarity, and assures a constant average value of the contact force. Two thin plates hold the pad support in the tangential (sliding) direction. The breakings have been performed at a constant velocity of 10 rpm.

2.2. Set-up dynamics

An experimental modal analysis of the brake system was performed [21]. The analysis led to distinguish three different substructures: disc, support and pad. The dynamics of the assembly can be analyzed like the sum of the disc, pad, and support dynamics; in fact the reduced contact surface (10 mm $\times$ 10 mm) between pad and disc allows for a low coupling between the tangential dynamics of the pad (or support) and the normal (bending) dynamics of the disc. Therefore, when referring to the whole system, the following notation is used: the modes of the coupled system that involve bending vibration of the disc are named here "disc modes", since the greater part of the energy is concentrated on the disc; as well the modes involving bending vibration of the pad (or support) are named "pad (or support) modes". When the modes get close in frequency the vibration involve both the substructures [21].

Particular attention was focused on modes involving bending vibration of the disc (in the normal direction) and modes involving tangential vibration of the support and of the pad (in the friction

Table 1
System natural frequencies and modal damping of the set-up with normal load equal to 25 N

Mode	Frequency [Hz]	Hysteretical damping (%)
I support	489	7.27
(0,1+)	925	5.36
(0,2−)	1425	1.67
(0,2+)	1625	3.69
II support	2091	0.72
(0,3−)	2317	1.18
(0,3+)	2458	2.02
III support	2912	3.99
(1,0)	3058	2.11
(0,4−)	3750	1.31
(0,4+)	3808	0.67
IV support	5146	2.09
(0,5−)	5575	0.49
(0,5+)	5589	0.32
(1,1)	7217	2.35
V support	7717	0.75
(0,6)	7725	0.26
(1,2)	8025	3.06
(0,7+)	10088	0.37
(0,7−)	10141	0.38
(1,4)	12367	1.07
(0,8+)	12725	0.27
(0,8−)	12825	0.17
(1,5)	15283	0.58
(0,9+)	15517	0.54
(0,9−)	15708	0.13

direction), because in [21] these modes are identified to be involved in squeal.

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 system looses the axial symmetry; therefore, the modes involving vibration of the disc (the "disc modes" of the coupled system) are no longer double modes and they split at two different frequencies.

The following notation is adopted to name coupled system modes:

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

Table 1 shows the natural frequencies and the modal damping of the set-up calculated when a contact load equal to 25 N is applied and a friction pad with 10 mm × 10 mm of contact surface is mounted. The first two modes of the support are rigid modes while the third, fourth and fifth modes of the support are bending modes. In the frequency range of interest (1–20 kHz) two tangential modes of the pad are recognized and the natural frequencies can range from 3.7 to 5 kHz and from 10 to 14 kHz, respectively, by changing the set-up parameters, i.e., pad dimensions, normal load, etc. The dynamics of the set-up can be adjusted in order to have different squeal conditions.

2.3. Squeal phenomena

Experimental tests are performed in order to find as many as possible squeal frequencies by modulating the parameters values. Five different squeal conditions are found: 1566, 2467, 3767, 7850 and 10,150 Hz. These squeal phenomena are obtained for well-defined values of the driving parameters, and all of them are easily reproducible. During this campaign the dynamic of the system is followed to relate its variation to the rising of the instabilities.

As shown in [21] squeal happens only when a coincidence in frequency of two opportune system modes is obtained, i.e., 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. The condition characterized by the closeness of the two natural frequencies, obtained by modulating the system dynamics, is named modal "tuning-in" [21].

Precisely, squeal at 1566 Hz occurs when the second mode frequency of the support tunes-in with the (0,2+) mode frequency of the disc, squeal at 2467 Hz when the third mode frequency of the support tunes-in with the (0,3+) mode frequency of the disc, squeal at 3767 Hz when the first mode frequency of the pad tunes-in with the (0,4+) mode frequency of the disc, squeal at 7850 Hz when the second mode frequency of the pad tunes-in with the (0,6+) mode frequency of the disc, squeal at 10,150 Hz when the second mode frequency of the pad tunes-in with the (0,7+) mode frequency of the disc.

During squeal the vibration shape of the system is the one of the unstable mode and, at the contact area, it is characterized by normal vibration of the disc contact surface and by tangential and normal vibration of the surface of the pad.

3. The tribological triplet

3.1. First bodies

The first bodies in contact are the brake disc and the brake pad that is assembled to the floating support.

The disc is made of steel with Young modulus equal to 210 GPa and material density equal to 7800 kg/m^3. Since the disc surface was machined unidirectionally a discontinuity in the roughness of the contact area is due to the disc rotation. Such discontinuity affects the squeal amplitude that varies periodically with the period of the disc rotation. Nevertheless, the discontinuity is eliminated after some braking phases, when the disc surface is worn up by the sliding contact with the pad.

The friction pad is obtained by machining commercial automotive brake pads. The materials of automotive brake pads are usually composites formed by hot compaction of powders that

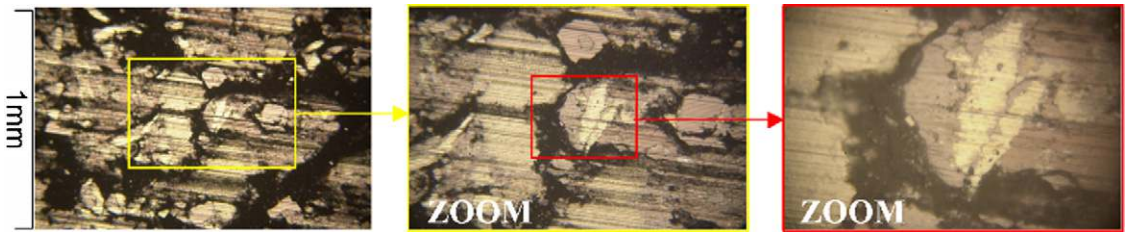


Fig. 2. Three consecutive zooms of the pad contact surface characterized by contact plateaus. V_s is the sliding velocity.

include many different components. Each component has different function and can be divided into four main categories: binders, structural materials, fillers and friction modifiers.

The exact formulation of the friction material is unknown. Moreover, as the pad gradually is worn off, the different wear rates of the different components cause a different material composition at the contact surface. In particular, particles and fibers with high wear resistance (e.g., structural materials) resist better than the neighbouring material and form areas of high contact pressure.

3.2. The role of the third body

Particles coming from the pad and the disc constitute the third body [16] and have a key role in the tribological triplet [17]. The third body separates the first bodies, transmits the load between the first bodies and accommodates the relative velocity.

Eriksson et al. [25] showed that the real contact between brake pads and disc consists of a number of small contact plateaus. The plateaus usually show signs of wear and sliding contact, including a pattern of parallel grooves along the direction of sliding Fig. 2. The nucleation of the contact plateaus takes place typically where the metal fibers come out from the surface, with hardness values considerably higher than the mean hardness of the pad composite. The third body agglomerates and compacts around the harder point, enlarging the effective area of the contact plateaus Fig. 2. A different kind of contact plateaus is observed where a large portion of the contact area is not interested by fibers and an uniform sliding area is in contact with the disc see Fig. 6a. An energy dispersive spectrometry (EDS) allows for characterizing these areas by their composition: an extremely high percentage in carbon is identified. The graphite is used into the automotive pads to stabilize the friction coefficient. The area around the contact plateaus does not show signs of sliding contact.

With such configuration the contact load is concentrated in reduced areas where the effective contact pressure can reach extremely large values with respect to the mean contact pressure applied on the whole pad surface.

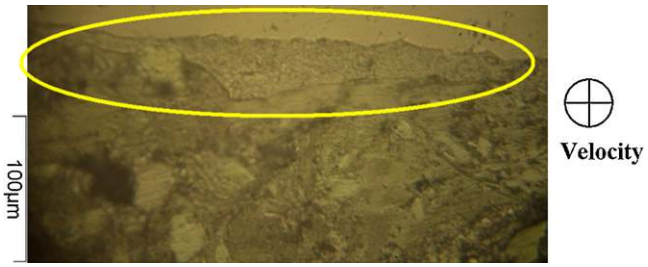


Fig. 3. Layer of third body accommodating the pad and disc contact surfaces.

Fig. 3 shows a section of the contact surface perpendicular to the sliding direction. After treatment with resin, preventing loose wear particles from being removed from the surface, the pad is cut in a plane normal to the sliding velocity to highlight the third body layer after brake events. A layer (about 10 μm) of third body accommodates the pad and disc profiles and forms the contact surface. The same situation is observed at all the contact areas; the areas in contact are constituted or by contact plateau of third body, or by structural fibers in contact with the disc surface and surrounded by third body. A layer of third body is always found between the disc and pad surface, as showed by the SEM analysis in the following. The parallel grooves due to the disc surface are recognizable on the top of the layer of third body, where the contact takes place. The analysis underlines the importance of the third body in the contact topography between pad and disc.

3.3. System behaviour with and without squeal

The behaviour of the brake system during squeal is mainly described by the dynamic analysis reported in [21]. Here, the effects on the contact configuration and the differences between the dynamic behaviour with and without squeal are considered.

The numerical analysis of the contact constraints reported in [22] shows the calculated temporal distribution of contact pressure. The contact stresses are not uniformly distributed on the contact

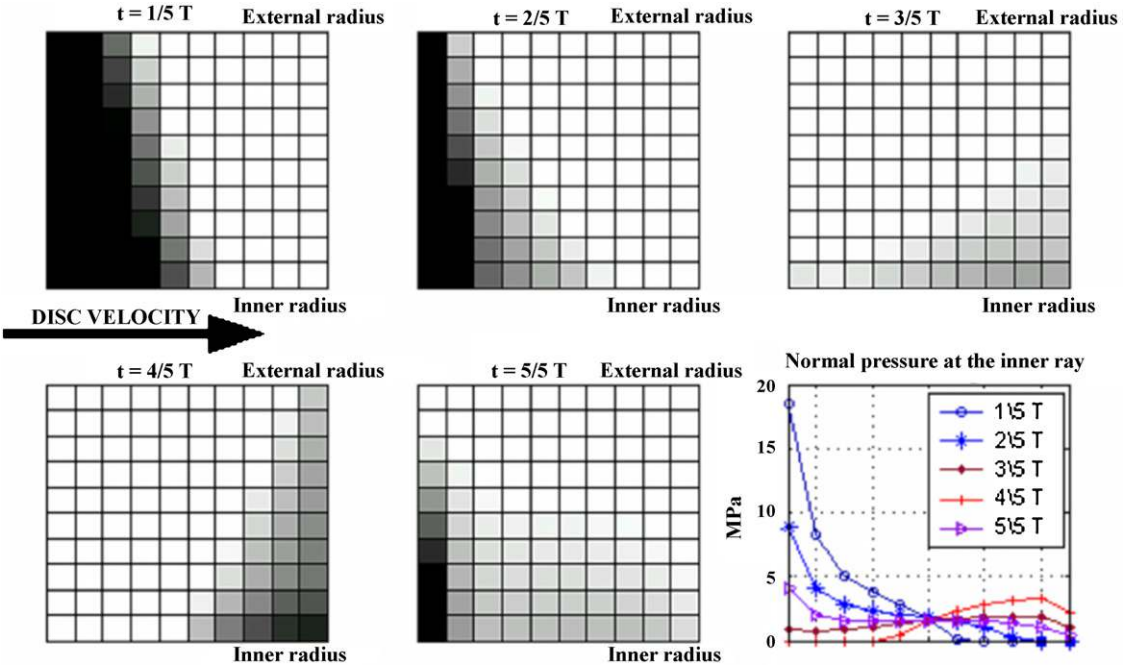


Fig. 4. Contact normal pressure distribution calculated at the pad surface (10 mm × 10 mm) by FEM, for five time steps t along the squeal period T = 2.9e−4. The average pressure is $\bar{\sigma}_n = 1$ MPa.

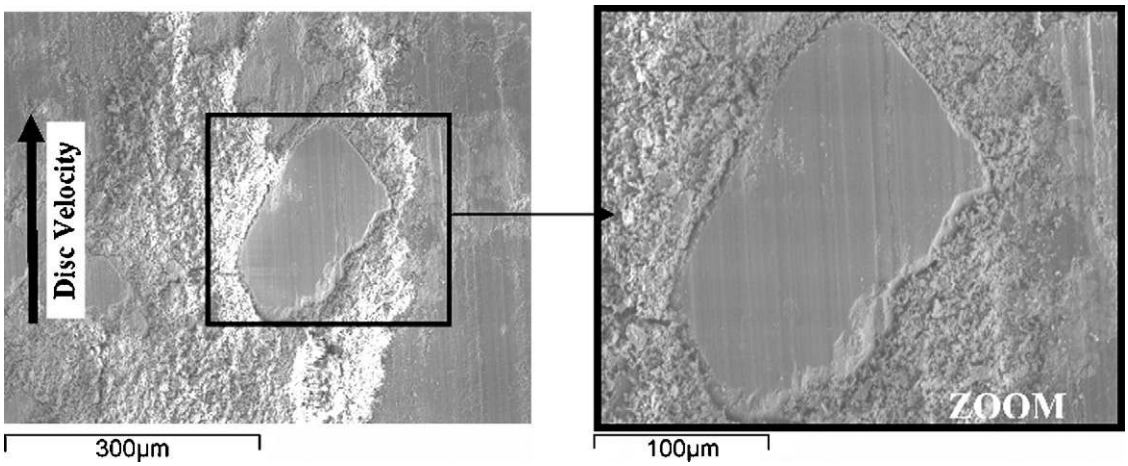


Fig. 5. Contact plateau due to the hard material fiber at the contact surface.

surface and the largest values are calculated at the leading edge, along the inner radius. In fact, because of the tangential deformation of the support under the friction torque (due to the friction force and the constraint at the connection with the thin plates), the contact stresses are the highest at the leading edge of the pad surface. The largest value of the contact stresses at the inner radius is due to the bending (static) deformation of the disc under the normal load.

When squeal does not occur, despite the non-uniform spatial distribution of the contact stresses, the local values are almost constant in time [22].

During squeal the system vibrates at the squeal frequency and the vibration shape is the one of the unstable mode, characterized by normal (bending) vibration of the disc and tangential vibration of either the support or the pad [21]. The effects at the contact area can be considered as the sum of the vibration of the disc contact surface and the vibration of the pad contact surface. While the disc surface is vibrating in the direction normal to the contact area, the pad surface is at the extremity of the support–pad assembly. Therefore, when the system vibrates with a tangential mode of either the pad or the support, the pad surface is characterized by both tangential and normal components of the vibration. Being in contact with the disc surface, the normal component produces a large local oscillation (at the squeal frequency) of the contact stresses that are both space and time dependant. At the leading edge such oscillation ranges from zero to fifteen times the mean value of the contact pressure: Fig. 4 shows the local contact pressure (σ_n) calculated by FEM at five different time steps during the temporal braking simulation in squeal conditions ($\omega_{\text{disc}} = 100$ rpm, $\vec{v}_r = 0.02$ m/s, $\mu = 0.5$, global normal force $N = 100$ N, mean contact pressure $\bar{\sigma}_n = 1$ MPa). The time steps are chosen to cover the period of vibration. The white area indicates the local detachment ($\sigma_n = 0$) between pad and disc surface. The black area corresponds to the maximum of the contact pressure. As in the previous case, the highest peak of oscillation is located at the leading edge and the larger contact stresses are located at the inner radius. The last graphic in Fig. 4 shows the pressure distribution at the inner radius corresponding to the five time steps. Again, because of the tangential vibrations of the pad surface, the relative sliding velocity between the pad and the disc oscillates at the squeal frequency around the average sliding velocity $\vec{v}_r$ due to the disc rotation. The local impacts and local relative slidings at high velocity and high frequency cause much higher local normal and tangential stresses [26]; these oscillations at high frequency (squeal frequency) of the local stresses involve the first layer of the material at the contact surface.

During the numerical analysis both surface are considered flat as the initial roughness is erased in the first moment of contact between the pad and the disc. The third body in the contact and the micro-distribution of the contact pressure (contact plateaus) are not modelled in these finite element simulations. The aim of the simulation is to get an idea of the local contact conditions due to the vibrations of the system.

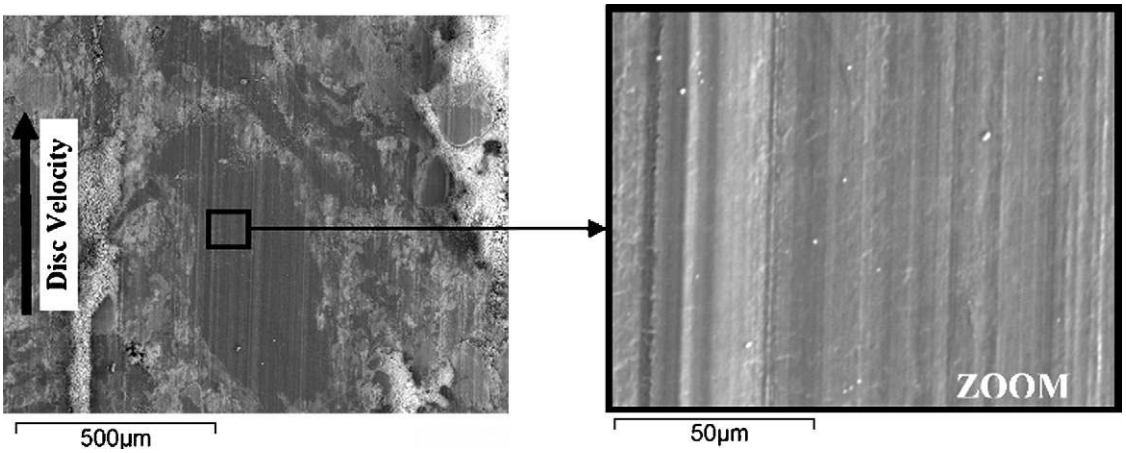


Fig. 6. Contact plateau due to the third body accumulated over the soft pad components (no fibers in the contact area). The third body is compact.

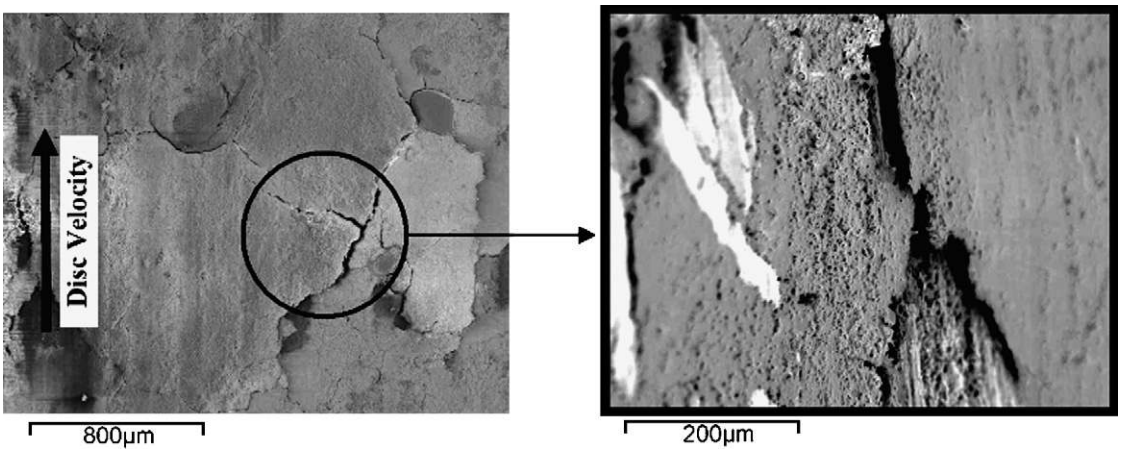


Fig. 7. Material exfoliations at the pad contact surface after squeal.

In conclusion, during the squeal, the dynamic behaviour affects drastically the contact conditions by introducing large oscillations of the local contact stresses at the squeal frequency. On the contrary, without squeal, the system behaviour affects only the spatial distribution of the local contact stresses that remain constant in time.

4. Contact surface topography

4.1. Pad surface topography without squeal

A preliminary analysis of the contact surface of the pad after a braking phase without squeal is carried out by the SEM (Scanning Electron Microscope). After a consistent braking period (about 100 min at 10 rpm) the whole pad surface is worn and the pad is in full contact with the disc. Nevertheless, the real contact area is localized at the contact plateaus. Fig. 5 shows a contact plateau formed around a structural fiber that is composed by hard material. It is characterized by a compact smooth area with small parallel grooves along the sliding direction that are determined by the sliding with the disc. The plateau is surrounded by a no-contact area where the material has been worn off.

Fig. 6 shows a contact plateau where structural fibers do not compare. The contact plateau is constituted by a smooth layer of third body.

Despite the pad topography obtained after braking events without squeal reveals a non-homogenous surface characterized by contact and non-contact areas of different dimensions, the material results to be compact. No cracks or detachments are observed in the analysis.

4.2. Pad surface topography during squeal

The pad surface topography after squeal reveals a completely different aspect with respect to the topography without squeal.

The analysis is performed on a sample identical to the one used for the previous analysis, machined from the same brake pad. After a previous braking phase to uniform the contact area of the pad, the values of the driving parameters are imposed to have squeal at 3767 Hz, i.e., when the first tangential mode of the pad tunes in with the (0,4+) mode of the disc [21]. Several brake events with squeal are reproduced for a total time of about 20 min.

Figs. 7 and 8 show a material exfoliation at different scales. Fig. 7 shows a material exfoliation covering an area larger than 1 mm^2; while Fig. 8 shows a contact plateau where the dimensions of the exfoliations are of some μm^2. Most of the contact plateaus on the pad surface are characterized by a no compact third body.

Fig. 9a shows a contact plateau due to a superficial fiber with the surrounding no-contact area. The pad surface around the fiber is characterized by several cracks. Same results are shown in Fig. 9b where a contact plateau of soft material is interested by the cracks, together with the surrounding no-contact zone (note that the scales in Fig. 9 are different). The whole pad surface is characterized by superficial cracks or material exfoliations. Cracks can nucleate either in proximity of structural fibers or not.

The topography of the pad surface highlights the effects of the system vibrations at the contact zone. The high frequency vibrations cause oscillations of the local contact stresses as evidenced by the numerical analysis [22]. Therefore, the pad material presents signs of fatigue, i.e., cracks and material exfoliations.

Once imposed the system parameters that bring to squeal (modal coupling), new tests were performed by cleaning the disc and pad surface to remove the detached third body: squeal is still obtained as soon as the brake starts. On the contrary, by changing the values of the macroscopic parameters to obtain the lock-out between the two system modes, squeal does not occur. This means that the analyzed status of the contact surface is an effect and not a cause of squeal.

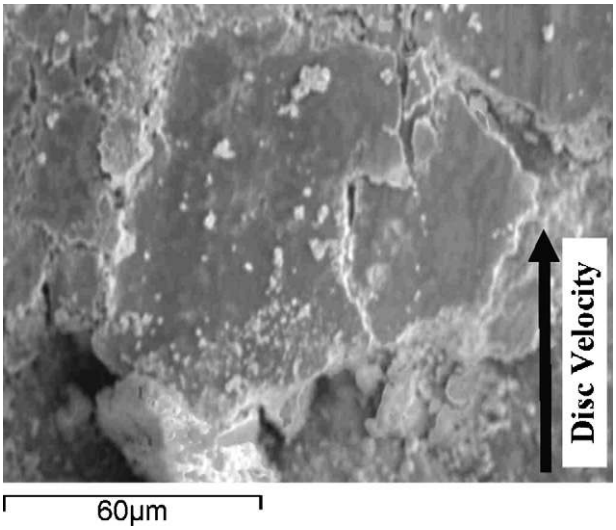


Fig. 8. Micro-exfoliation of the third body (contact plateaus) after squeal.

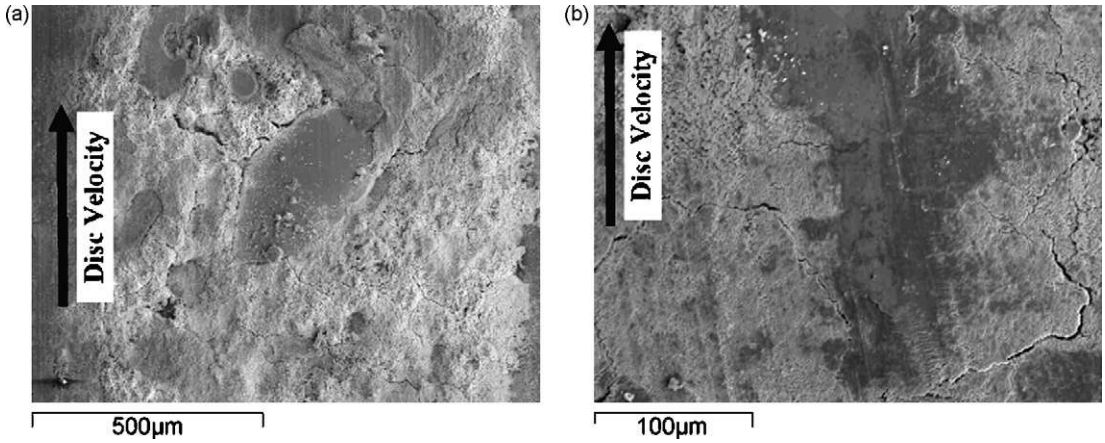


Fig. 9. Superficial cracks both at the no-contact area (a) and at the contact plateaus (b), after squeal.

4.3. Disc surface topography

Since the disc was machined unidirectionally along a diameter direction, the initial roughness of the disc surface is not constant during the disc rotation. The roughness of the surface affects the squeal amplitude that oscillates with the periodicity of the disc rotation. In particular the squeal amplitude is lower when the roughness of the disc contact surface is higher, as shown also by Bergman et al. [17]. This behaviour can be explained by the lower macroscopic friction coefficient due to the reduction of the real contact area which in turn is caused by the frequent impacts between the pits of the disc surface and the contact plateaus, preventing them from growing to their normal "steady state" size [15].

Nevertheless, after a large number of runs, the starting roughness is worn up, and the disc surface is characterized by concentric circular grooves along the sliding direction (Fig. 10). The grooves are formed because of the plastic deformation of the disc material that can be ascribed either to the contact with the hardest points (structural fibers) of the pad surface or to the local high pressure due to the third body. During this stage the squeal amplitude is constant with the disc rotation.

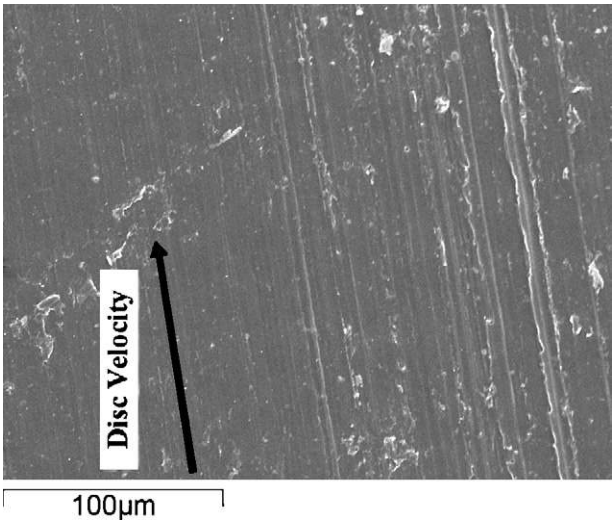


Fig. 10. Disc contact surface; circumferential grooves due to the sliding with the pad.

5. Analysis of the pad superficial material during squeal

In order to understand if the cracks and the material exfoliations involve only the superficial layer of the third body or also the pad, the friction pad was cut along the direction normal to the sliding velocity. Before cutting, the pad surface was fixed by immersing it into a specific resin. In the following figures the upper uniform grey zone is the resin and the third body layer is recognizable between the resin and the pad composite. Different kinds of superficial transformations are observed.

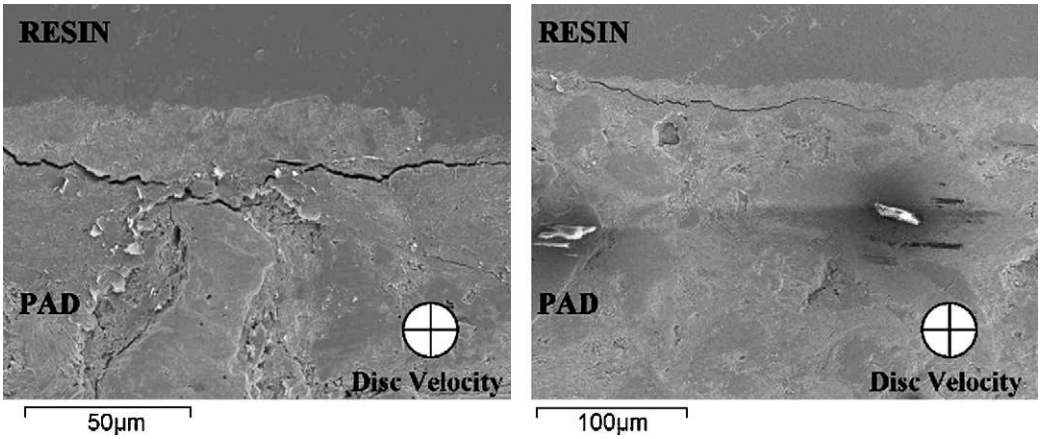


Fig. 11. Section of the pad contact surface normal to the sliding direction, highlighting detachments of the third body layers after squeal.

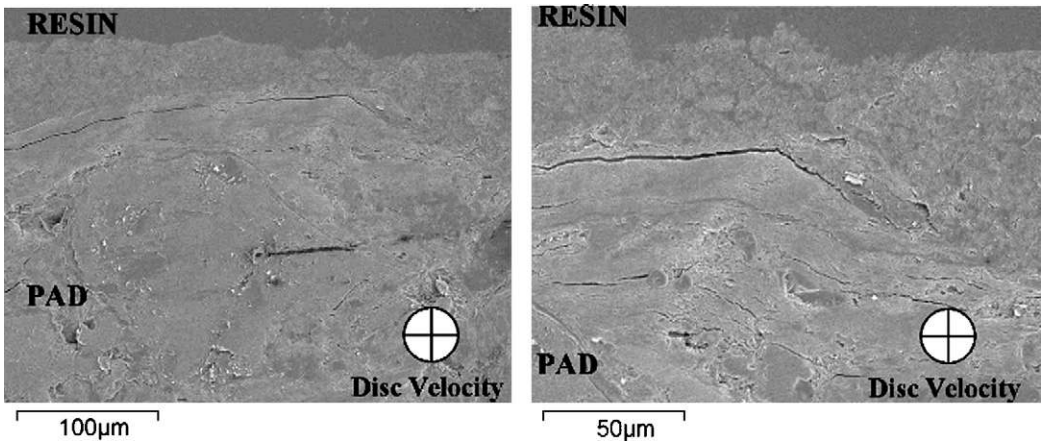


Fig. 12. Detachments of layers of pad material under the third body, after squeal.

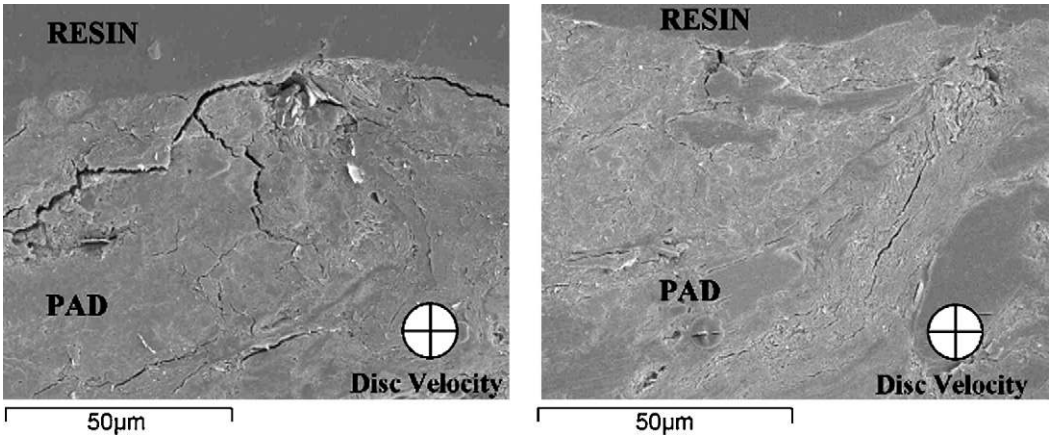


Fig. 13. Superficial cracks develop inside the pad material, after squeal.

Fig. 11 shows two cases where the third body layer is detached from the pad material. Such detachments can be ascribed to the squeal vibrations, because the interface between pad and third body is weak. These superficial detachments correspond to the exfoliations visible in the pad topography before the cut of the sample, and are not observed in samples without squeal.

Fig. 12 shows further superficial detachments. In this case the detachment does not happen between the pad and the third body,

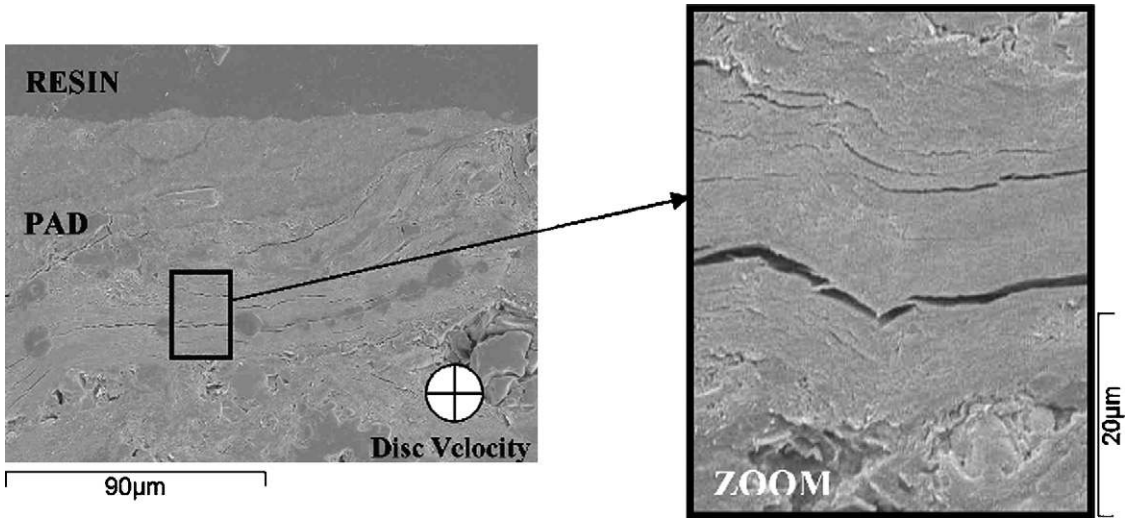


Fig. 14. Internal cracks close to the pad surface developing along layers of pad composites.

but a thin layer of pad material detaches together with the third body layer. Thus, a crack parallel to the pad surface develops under the layer of third body, inside the pad material.

Moreover, the cracks do not involve only the surface of the pad and the detachments due to the third body layer. Starting from the contact surface several cracks develop inside the pad material up to a depth of about 10 μm. Fig. 13 shows two examples.

The material cracks can also nucleate inside the volume of the pad, close to the contact surface (Fig. 14). In the first layer of pad material (about 400 μm from the contact surface) several cracks are observed without any preferential direction. The cracks follow the material composition by developing along areas filled by the same material component. This component expands throughout all the volume of the pad and the cracks develop along the direction of the continuous layers made of such component.

The whole surface of the pad and the superficial layer (up to about 400 μm of depth) are covered by several cracks, while the internal volume of the pad is compact and does not present cracks.

The material exfoliations and the superficial and internal cracks close to the contact surface highlight the fatigue excitation of the pad material due to squeal vibrations. In fact, the high frequency (squeal at 3437 Hz) oscillations of the contact stresses, shown by the numerical analysis of the contact stresses during squeal [22], cause fatigue in the superficial layer of the pad material and fragmentation of the third body layer.

6. Conclusions

The tribological analysis reported in this paper is complementary to the dynamic analysis presented in [21] and to the numerical analysis presented in [22].

The analysis is addressed mainly to study the differences between contact surfaces with and without squeal. To understand the findings, the behaviour of the system, analyzed through the dynamic analysis, is taken into account.

After squeal events, the superficial layer of the pad material presents several signs of fatigue. Both exfoliations of the third body and cracks nucleated at the pad contact surface and inside the first layer of the pad material are observed.

On the contrary the disc surface topography is not affected by the squeal. The contact with the hardest points of the pad surface produces concentric grooves at the disc periphery, where the contact happens. The disc roughness, tangential to the sliding direction, affects the squeal amplitude; it is worn up after a number of brake cycles.

The comparison between the post-mortem surface topography and the behaviour of the local contact pressure calculated by numerical simulations [22] shows that the numerical simulations and the surface topography are consistent with each others.

The large oscillations of the contact stresses due to the high frequency vibrations of the system, vibrating at its unstable mode, causes the rise of fatigue cracks and the fragmentation of the third body. Tests were made to highlight that the status of the contact surface is an effect and not a cause of squeal.

The results of the tribological analysis highlight a further problem that can arise from squeal vibrations: the reduction of the pad life-time. In fact, the fragmentation of the third body (consider that some components of the third bodies are properly added to reduce the wear rate) and the fatigue damages of the superficial pad material can seriously affect the wear rate of the material and reduce the life-time of the friction pad. An estimation of the wear rates with and without squeal on the same test bench is left for future work.

References

[1] N.M. Kinkaid, O.M. O'Reilly, P. Papadopoulos, Automotive disc brake squeal, Journal of Sound and Vibration 267 (2003) 105–166.
[2] A. Akay, Acoustic of friction, Journal of Acoustical Society of America 111 (4) (2002) 1525–1548.
[3] M. Godet, The third-body approach: a mechanical view of wear, Wear 100 (1984) 437–452.
[4] Y. Berthier, Background on friction and wear, in: Lemaître Handbook of Materials Behavior Models, Academic Press, 2001, pp. 676–699, Section 8.2.
[5] H.R. Mills, Brake squeak, in: Technical Report 9000 B, Institution of Automobile Engineers, 1938.
[6] R.A.C. Fosberry, Z. Holubecki, Interim report on disc brake squeal, in: Technical Report 1959/4, Motor Industry Research Association, Warwickshire, England, 1959.
[7] R.T. Spurr, A theory of brake squeal, in: Proceedings of the Automobile Division, Institution of Mechanical Engineers 1961–1962, 1, 1961, pp. 33–52.
[8] S.W.E. Earles, A mechanism of disc-brake squeal, in: Technical Report 770181, SAE, Warrendale, PA, 1977.
[9] R.P. Jarvis, B. Mills, Vibrations induced by friction, in: Proceedings of the Institution of Mechanical Engineers 178, 32, 1963, pp. 847–857.
[10] A. Akay, J. Wickert, Z. Xu, Investigation of mode lock-in and friction interface, in: Final Report, Department of Mechanical Engeneering, Carnegie Mellon University, Pittsburgh, 2000.
[11] A. Tuchinda, N.P. Hoffmann, D.J. Ewins, W. Keiper, Mode lock-in characteristics and instability study of the pin-on-disc system, in: Proceedings of the International Modal Analysis Conference–IMAC, 1, 2001, pp. 71–77.
[12] R. Allgaier, L. Gaul, W. Keiper, K. Willnery, N. Hoffmann, A study on brake squeal using a beam on disc, in: Proceedings of the International Modal Analysis Conference – IMAC, 1, 2002, pp. 528–534.
[13] M.R. North, Disc brake squeal-a theoretical model, in: Technical Report 1972/5, Motor Industry Research Association, Warwickshire, England, 1972.
[14] O. Giannini, A. Akay, F. Massi, Experimental analysis of brake squeal noise on a laboratory brake set-up, Journal of Sound and Vibration 292 (1–2) (2006) 1–20.
[15] H. Ouyang, W. Nack, Y. Yuan, F. Chen, Numerical analysis of automotive disc brake squeal: a review, International Journal of Vehicle Noise and Vibrations 1 (3–4) (2005) 207–230.
[16] M. Eriksson, F. Bergman, S. Jacobson, Surface characteristic of brake pads after running under silent and squealing conditions, Wear 232 (1999) 621–628.
[17] F. Bergman, M. Eriksson, S. Jacobson, Influence of disc topography on generation of brake squeal, Wear 225–229 (1999) 621–628.
[18] S.K. Rhee, P.H.S. Tsang, Y.S. Wang, Friction-induced noise and vibration of disc brakes, Wear 133 (1989) 39–45.
[19] G.X. Chen, Z. Zhou, P. Kapsa, L. Vincent, Effect of surface topography on formation of squeal under reciprocating sliding, Wear 253 (2002) 411–423.
[20] G.X. Chen, Z. Zhou, P. Kapsa, L. Vincent, Experimental investigation into squeal under reciprocating sliding, Tribology International 36 (2003) 961–971.
[21] F. Massi, O. Giannini, L. Baillet, Brake squeal as dynamic instability: an experimental investigation, Journal of the Acoustical Society of America 120 (3) (2006) 1388–1399.
[22] F. Massi, L. Baillet, O. Giannini, A. Sestieri, Brake squeal phenomenon: linear and non-linear numerical approach, Mechanical Systems and Signal Processing 21 (6) (2007) 2374–2393.
[23] O. Giannini, F. Massi, Characterization of the high-frequency squeal on a laboratory brake setup, Journal of Sound and Vibration 310 (1–2) (2008) 394–408.
[24] F. Massi, O. Giannini, Effect of damping on the propensity of squeal instability: an experimental investigation, Journal of the Acoustical Society of America 123 (4) (2008) 2017–2024.
[25] M. Eriksson, F. Bergman, S. Jacobson, On the nature of tribological contact in automotive brakes, Wear 252 (2002) 26–36.
[26] V. Linck, L. Baillet, Y. Berthier, Modeling the consequences of local kinematics of the first body on friction and on third body sources in wear, Wear 255 (2003) 299–308.