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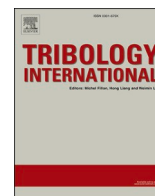
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Experimental investigation on brake squeal unpredictability: Role of the friction noise

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ABSTRACT

Dynamic contact instabilities represent a common issue in brake industry, causing noise and passenger dissatisfaction. Past works identified mode coupling as the cause of brake squeal, but the conditions for its occurrence are still not fully understood, making this phenomenon hardly predictable. This work investigates the relationship between the friction noise, intended as the dynamic excitation generated by the local interface interactions (ruptures, impacts ...), and the unstable response of the system. An experimental measurement campaign, conducted on a dedicated tribometer, highlighted that, even in presence of the same macroscopic friction coefficient, mode coupling instability occurred only when in presence of high values of friction noise. Therefore, the friction noise is proposed as a trigger factor for brake squeal occurrence.

1. Introduction

Friction-induced vibrations (FIVs) are an ever present phenomenon when dealing with dry contacts [1]. When interactions occur between asperities of the contact surfaces, a part of the kinetic energy of the system is released in the form of acoustic energy [2] rather than being dissipated as heat. A large number of local acoustic sources (impacts and ruptures between asperities), distributed along the whole contact interface, provides a broadband excitation of the system dynamics. This excitation, referred here as "Friction Noise", is caused by mechano-physio-chemical interactions between asperities in contact, such as adhesion, ruptures, impacts etc. Recent works tried to characterize the friction noise and its dependence on contact and material parameters [3–7], pointing out a strong dependence on relative velocity, contact pressure and surface roughness.

The contact dynamic excitation can result in different types of dynamic response of the system, either stable [8] or unstable [9]. The occurrence of unstable friction-induced vibrations has been extensively analyzed in literature, being associated with the sound emission from braking systems, cause of passenger dissatisfaction and large warranty costs afforded by brake industry [1]. Different phenomena have been identified to be the cause of sound emission at different frequencies, such as stick-slip [10–12], negative friction-velocity slope [13,14] and

mode coupling [15,16]. This latter, commonly accepted to be the cause of brake squeal [17,18], happens when two system modes coalesce [19] at the same frequency (modes lock-in), resulting in an apparent negative damping for one of the two coalescing modes [20]. The associated unstable vibrations are mainly characterized by a harmonic spectrum, as well as the typical emitted whistle. Despite the amount of works oriented to provide a squeal-free design of brakes [21–24], its occurrence is still extremely unpredictable. Many different reasons are at the origin of the unpredictability of the brake squeal, such as the transient and non-linear nature of the phenomenon [25–28], the strong dependence on system geometry and material parameters [29–31], evolution of the wear state of the friction pairs [32], the high modal density of brake systems in the squeal frequency range, 1–20 kHz. [33]. As a consequence, further investigation of mode coupling instability is necessary to clarify all the parameters that can affect the brake squeal occurrence, in order to better understand this phenomenon and to move towards a deterministic approach to predict brake squeal [34].

Some work in literature investigated the so called "dither actuator" [35,36], an active control system developed to avoid the occurrence of unstable friction induced vibrations. This controller provides to the system an external excitation on a different frequency range with respect to the unstable modes one, reducing the occurrence and the amplitude of the unstable FIVs observed for the uncontrolled system. This approach

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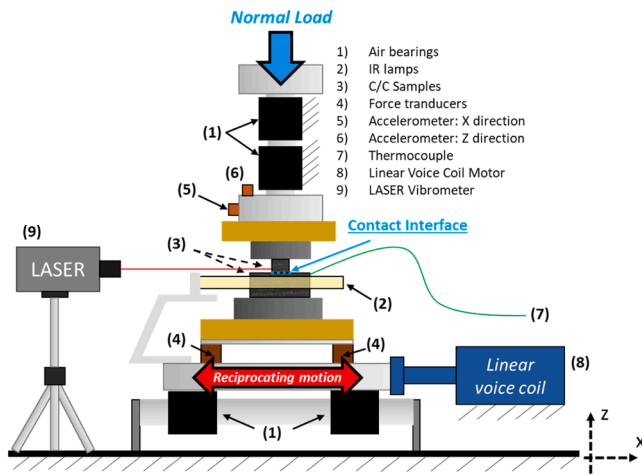


Fig. 1. TriboAir test bench functional scheme. The main components of the Test bench have been evidenced: 1) Air Bearings, 2) IR Lamps, 3) C/C Samples, 4) Force Transducers, 5) Accelerometer X direction, 6) Accelerometer Z direction, 7) Thermocouple, 8) Linear Voice Coil Motor; 9) LASER Vibrometer.

points out a relationship between the external excitation and the dynamic response of the system at the macroscopic scale.

This work aims to investigate the correlation between the broadband dynamic excitation generated at the contact interface (friction noise) and the occurrence of brake squeal events, proposing the friction noise as a key factor for brake squeal triggering. While previous analyses can be found in literature on different contact pairs, correlating friction noise arising from each sample material to the squeal occurrence [37], in this work the same contact pair has been used, modifying the friction noise thanks to the sensitivity of the material to the applied temperature. In such a way, this work provide an original contribution allowing to corroborate the influence of friction noise in triggering a mode coupling instability.

The experimental investigation, here proposed, has been based on measurement campaigns performed on a dedicated test bench, allowing for simulating the frictional interaction without providing parasitic vibrations [14,38]. The processing of the measured signals in terms of contact forces, relative velocity, temperature and vibrations (acceleration, vibrational velocity) allowed to characterize the evolution of the friction noise spectrum as a function of the different contact parameters, pointing out its correlation with the occurrence of squeal events.

First, the experimental set-up and the used contact samples are introduced in Section 2, together with the measurement protocol. Then, in Section 3, the signal acquisition and post-processing are detailed, highlighting how the friction noise amplitude and distribution in

frequency has been defined. In Section 4, the outcomes of friction noise characterization have been used to provide an explanation to the occurrence of the squeal unstable vibrations. Finally, the conclusions highlight how friction noise amplitude and frequency distribution have been observed to be positively correlated both to the squeal occurrence and to the squeal unstable frequency.

2. Materials and method

2.1. Experimental setup

It is worth to remark that friction induced vibrations and, in particular, mode coupling instability, are extremely sensitive to the system dynamics. As a consequence, many difficulties are encountered when performing the measurements on a real braking system, because of its complex and not controllable dynamics. For this reason, most of the experimental works dealing with brake squeal have been conducted on reduced scale test benches [17], characterized by a simple dynamics with a better control of the imposed boundary conditions, if compared to a full scale system. Moreover, being the friction noise characterized by a broadband and low amplitude vibration spectrum, the reduction of parasitic noise, coming from setup components (joints, bearings, driving motors), represents a crucial requirement for the adopted experimental setup. In this context, the experimental measurement campaigns have been conducted on a dedicated tribometer, specifically designed to measure low amplitude friction-induced vibrations, named TriboAir. A functional scheme of the test bench is reported in Fig. 1.

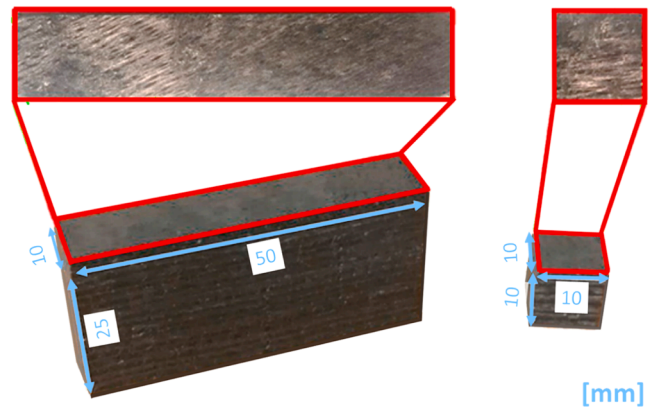


Fig. 3. Example of specimens used in the experimental campaign. The lower specimen is a 50×10×25 mm prism; the upper specimen is a cube with 10 mm edge. A detail of frictional surfaces is also provided.

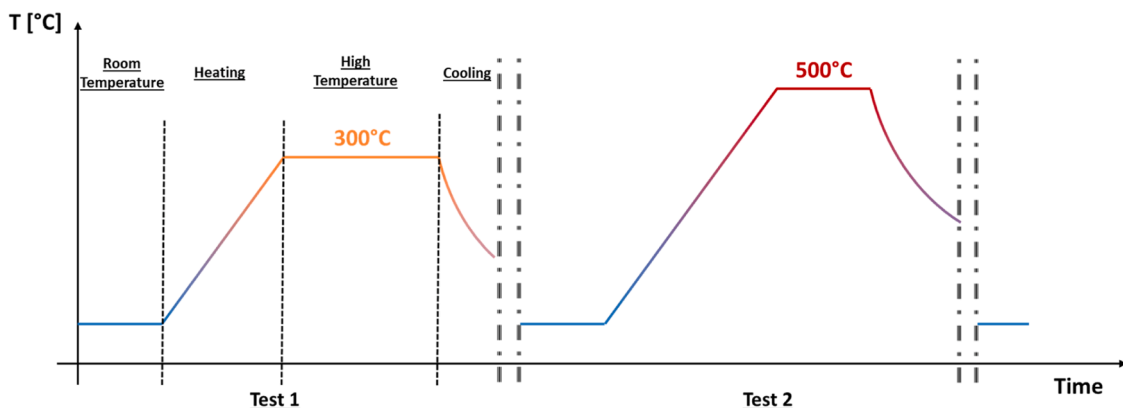


Fig. 2. Imposed temperature law. Two successive tests, with maximum temperatures of 300 °C and 500 °C, respectively, have been performed. Four conditions can be analyzed for each test: Room temperature (RT), heating phase, high temperature, cooling phase.

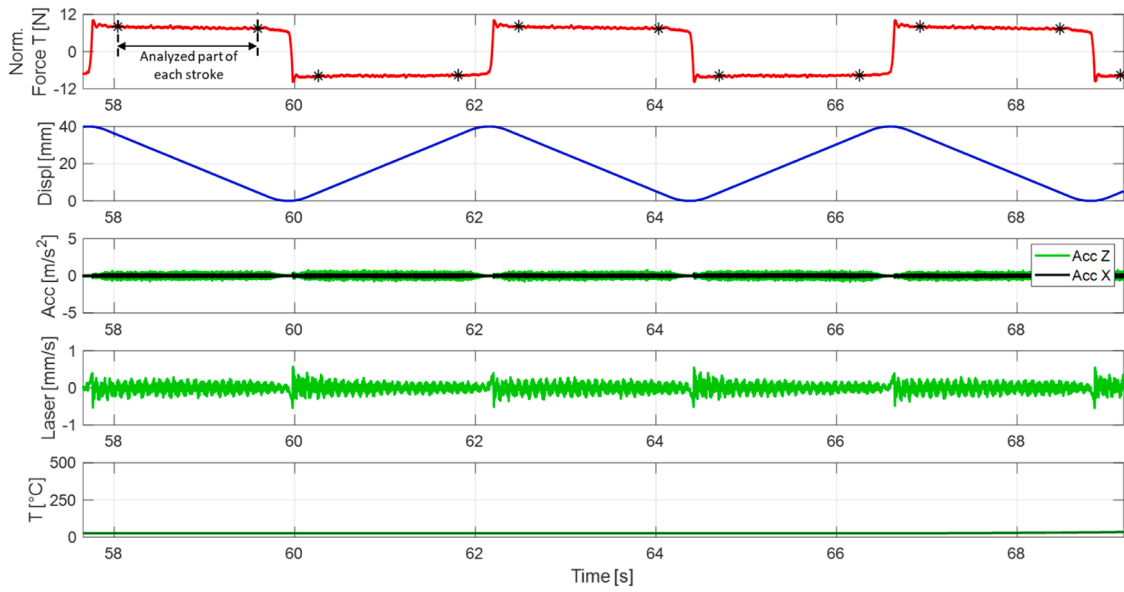


Fig. 4. Example of the recorded signals. From top to bottom: Normalized Tangential Force, Imposed Displacement, Normal (z-axis) and Tangential (x-axis) Accelerometers, Laser Velocity, Sample Temperature. The analyzed constant velocity part of each frictional stroke is evidenced.

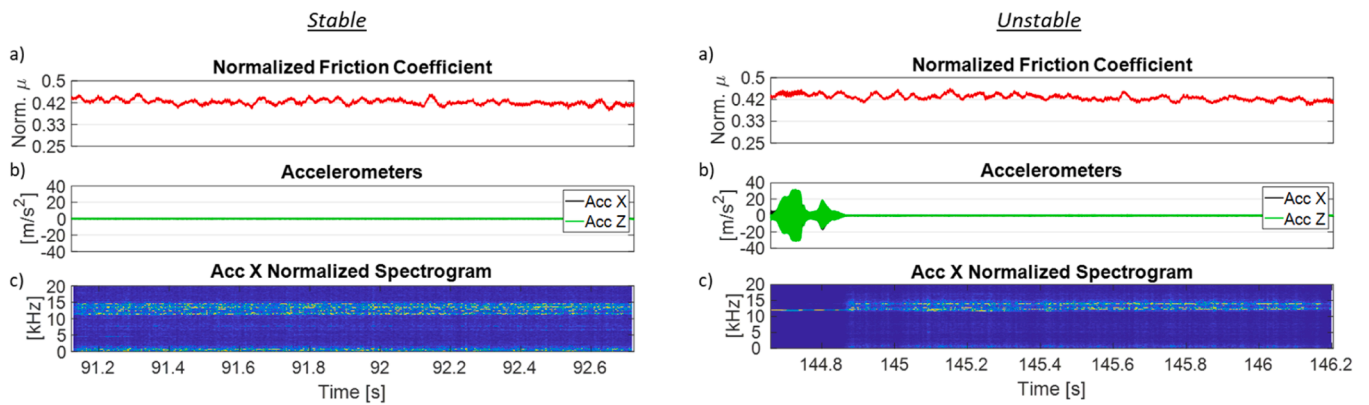


Fig. 5. Example of the analyzed frictional and dynamic response for a stable (left) and an unstable (right) strokes: a) Normalized friction coefficient observed during a single sliding stroke; b) Tangential (black) and normal (green) accelerometers time signals; c) Tangential accelerometer normalized spectrogram (for each time window the spectrum is normalized with respect to maximum computed value).

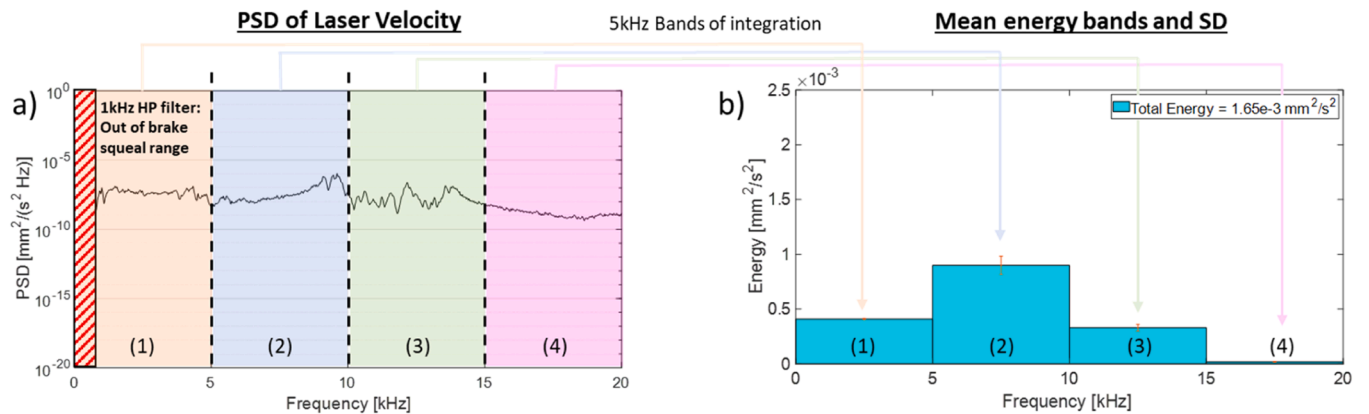


Fig. 6. a) Example of power spectral density (PSD) of the velocity signal recorded by the laser vibrometer, previously filter in the frequency range below 1 kHz b) Example of energy bands retrieved integrating the PSD in 5 kHz bands.

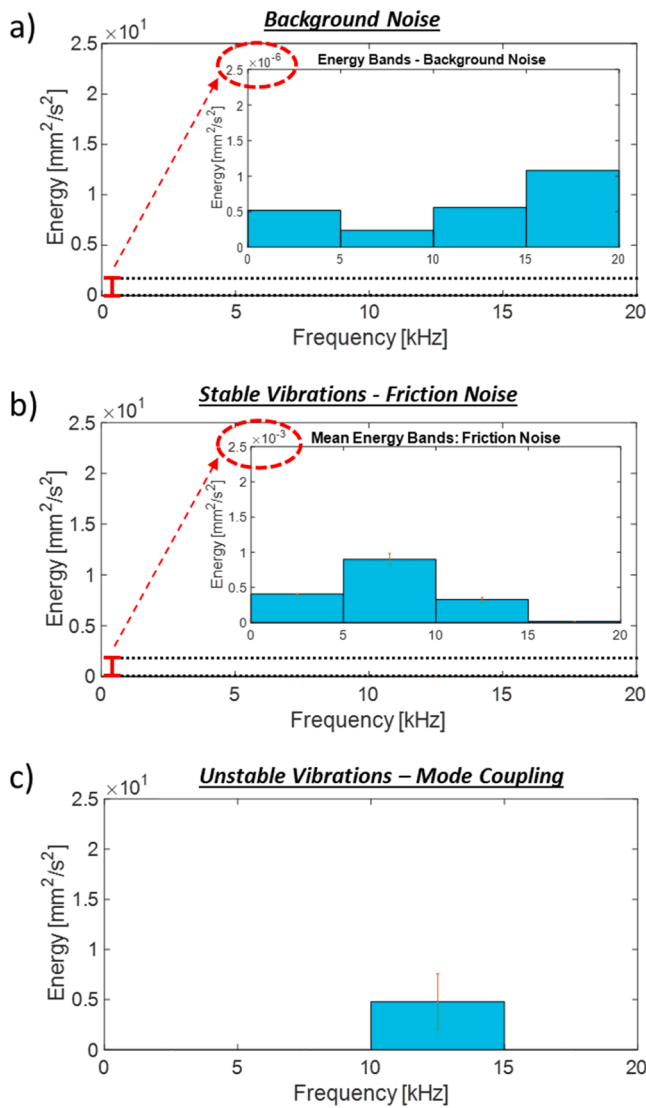


Fig. 7. Energy bands obtained in three different conditions: background noise (without sliding contact between samples), friction noise (stable vibration), mode coupling instability (unstable vibration). Due to the different scales, respective zooms are presented in a and b.

Being the TriboAir test bench a linear tribometer, a reciprocating linear motion is imposed to the lower sample by a linear voice coil motor ((8), BEI KIMCO LA30-75-001A) while the upper sample is fixed. The motor is coupled with an optical encoder (MicroE OPS-200-1-1) and a controller (Elmo G-DCWHI10/100EE) in order to ensure a closed-loop feedback control. This kind of motor, in combination with the use of air bearings (1), allows to have the moving part completely suspended in air, achieving the required reduction of external parasitic noise, as well as a better control of the boundary conditions. The contact interface between the samples (3), being the only interface in relative frictional motion, is, therefore, the only source of friction-induced vibrations. As reported in literature [31], the friction noise strongly depends on sliding velocity and contact pressure, therefore a constant sliding velocity motion law (20 mm/s) and a constant normal load (17 N, imposed by dead weights) have been adopted. Moreover, the normal and tangential forces are recovered by using 3D piezoelectric force transducers (Kistler 9017 C), placed on the moving plate, as reported in the Fig. 1. Being the friction noise a distributed excitation originated at the contact interface, its direct measure is impossible. As a consequence, regardless the choose of the measurement point, the obtained signal will be always “filtered”

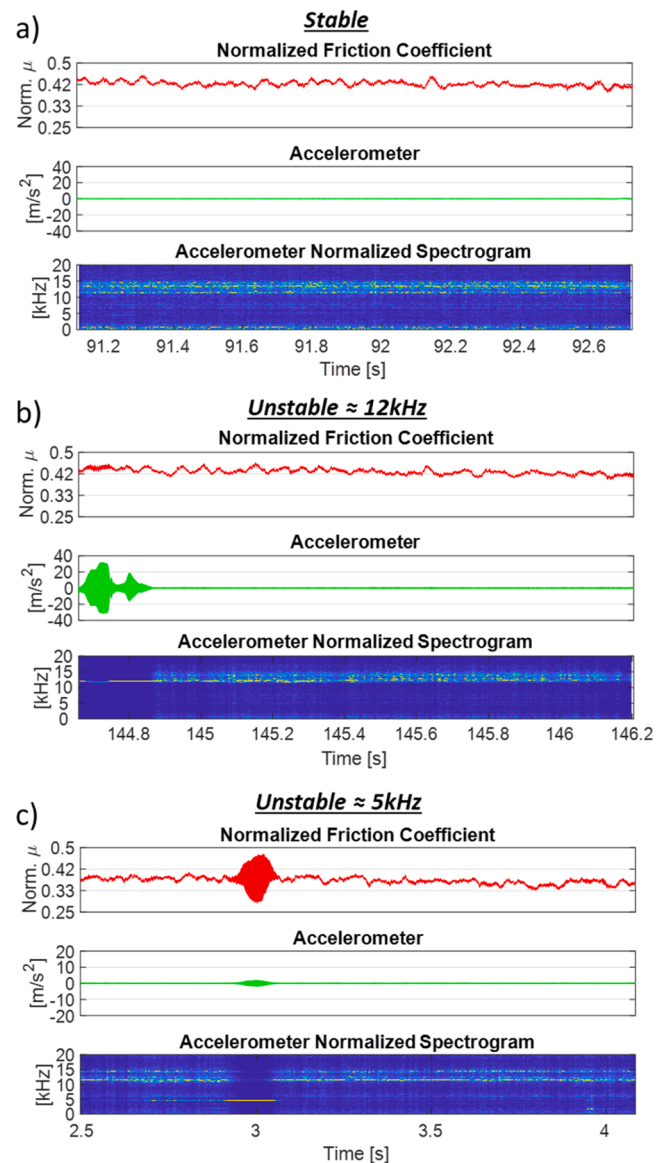


Fig. 8. Three different dynamic responses of the system obtained in correspondence of the similar values of the macroscopic friction coefficient: (a) stable behavior; (b) unstable response at 12 kHz; (c) unstable response at 5 kHz.

by the system dynamics, which acts as a transfer function between the excitation area (contact interface) and the measurement point. Nevertheless, in order to retrieve the most meaningful information on the friction noise, it has been chosen to perform the measurement as close as possible to the contact interface. A laser vibrometer (9) (Polytech, VFX-BW-500 kHz), focused 2 mm above the contact interface, has been chosen to perform this measurement without any insertion error. On the other hand, the dynamic response of the system, either stable or unstable, has been measured using two accelerometers (5–6) placed on the test bench structure, allowing to measure accelerations along both normal (z-axis) and tangential (x-axis) directions to the contact. The measurements of the accelerometer signals allows to identify the occurrence of unstable events by measuring the associated high amplitude harmonic vibrations, as well as to identify the observed instability evaluating the phase shift between the normal and the tangential accelerations, as suggested in [39].

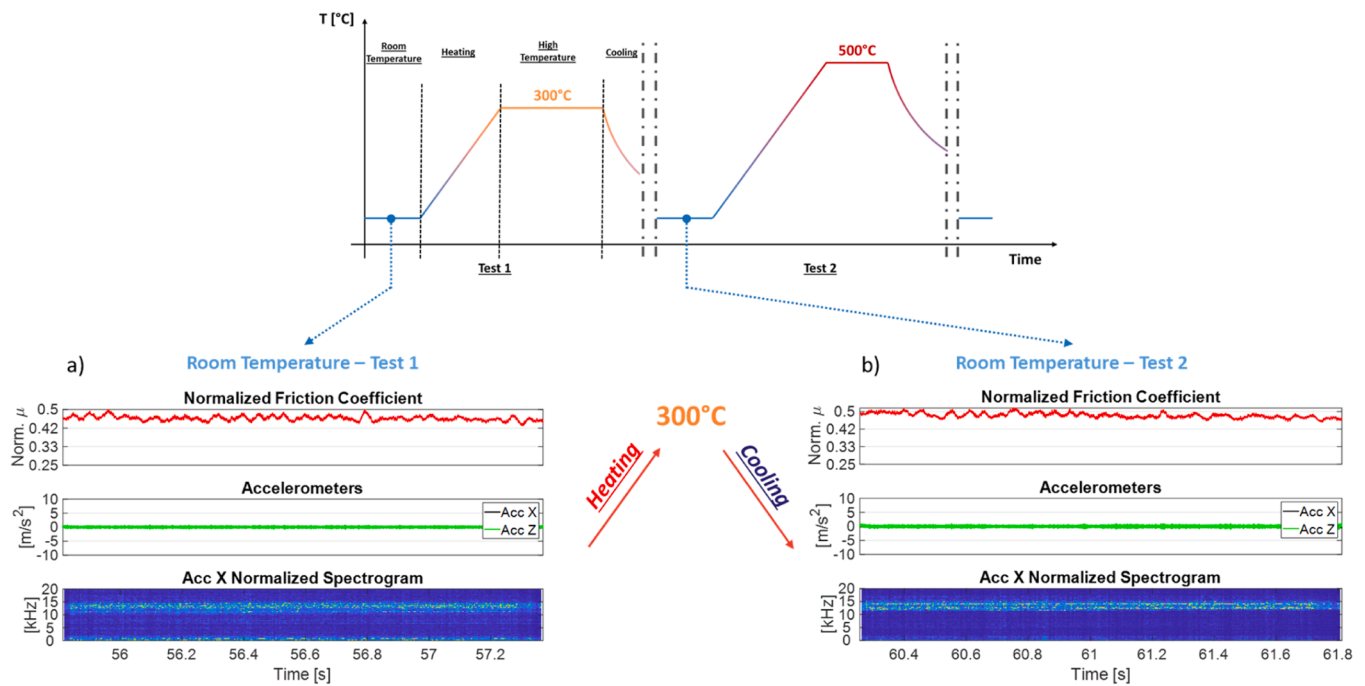


Fig. 9. Macroscopic frictional and dynamic response obtained at room temperature conditions before (a) and after (b) heating up to 300 °C. Similar values of the normalized friction coefficient, as well as stable dynamic response, are observed in both cases.

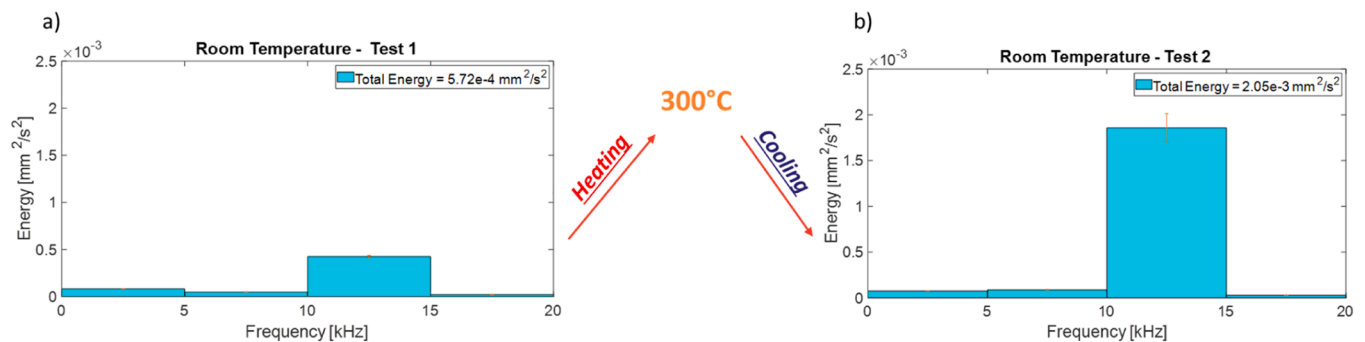


Fig. 10. Friction noise energy bands obtained at room temperature conditions, before (a) and after (b) heating up the system at 300 °C.

2.2. Measurement protocol and tested contact samples

In order to modify the frictional response of the materials in contact, different temperature laws have been imposed using InfraRed (IR) lamps (2), driven by a PID controller (GFRAN F650). The measurement of the sample temperature, necessary also to feedback the controller, is performed by a thermocouple (7) placed in a hole drilled 3 millimeters below the contact interface. The experimental campaign has been carried out considering two successive tests performed with a maximum temperature of 300 and 500 °C respectively, as shown in Fig. 2. Four different conditions have been analyzed for test 1 and test 2: room temperature, heating phase, high temperature and cooling phase. The different imposed temperatures allow to activate and modify the physicochemical reactivity of the material at the interface, affecting directly the overall frictional response and the released acoustic energy and, then, the friction noise.

Data are recorded by an acquisition system SIRIUSI - DEWESOFT, based on DualCoreADC® technology with dual 24-bit delta-sigma analog to digital converter (ADC), using a sampling frequency of 100 kHz, and then post-processed by MATLAB ©.

As concern the tested materials, two carbon/carbon composite samples have been used in this analysis, machined with a simple

geometry in order to simplify their own system dynamics. In fact, Carbon/Carbon (C/C) composites materials are widely used in aeronautic and high-performance automotive braking, and are consequently faced to a variety of usage and temperature conditions. The complex physiochemistry of C/C surfaces, strongly dependent on temperature and thermal history [40], makes them excellent potential candidates for the purpose of the study.

All the measurement have been performed using the same samples, without demounting them between the different tests. The geometries and the dimensions of the samples are shown in Fig. 3.

3. Friction noise analysis

The experimental campaign is here devoted to characterize both the frictional and dynamic responses of the system under well-defined and controlled boundary conditions. Moreover, the use of the laser vibrometer allows to retrieve information about the dynamic excitation generated at the interface level, namely the friction noise. To retrieve these information the main signals recorded during the test, presented in Fig. 4, have to be adequately processed. The constant velocity part of each frictional stroke, i.e. the part considered for all the successive analyses, is highlighted by two successive star markers in Fig. 4.

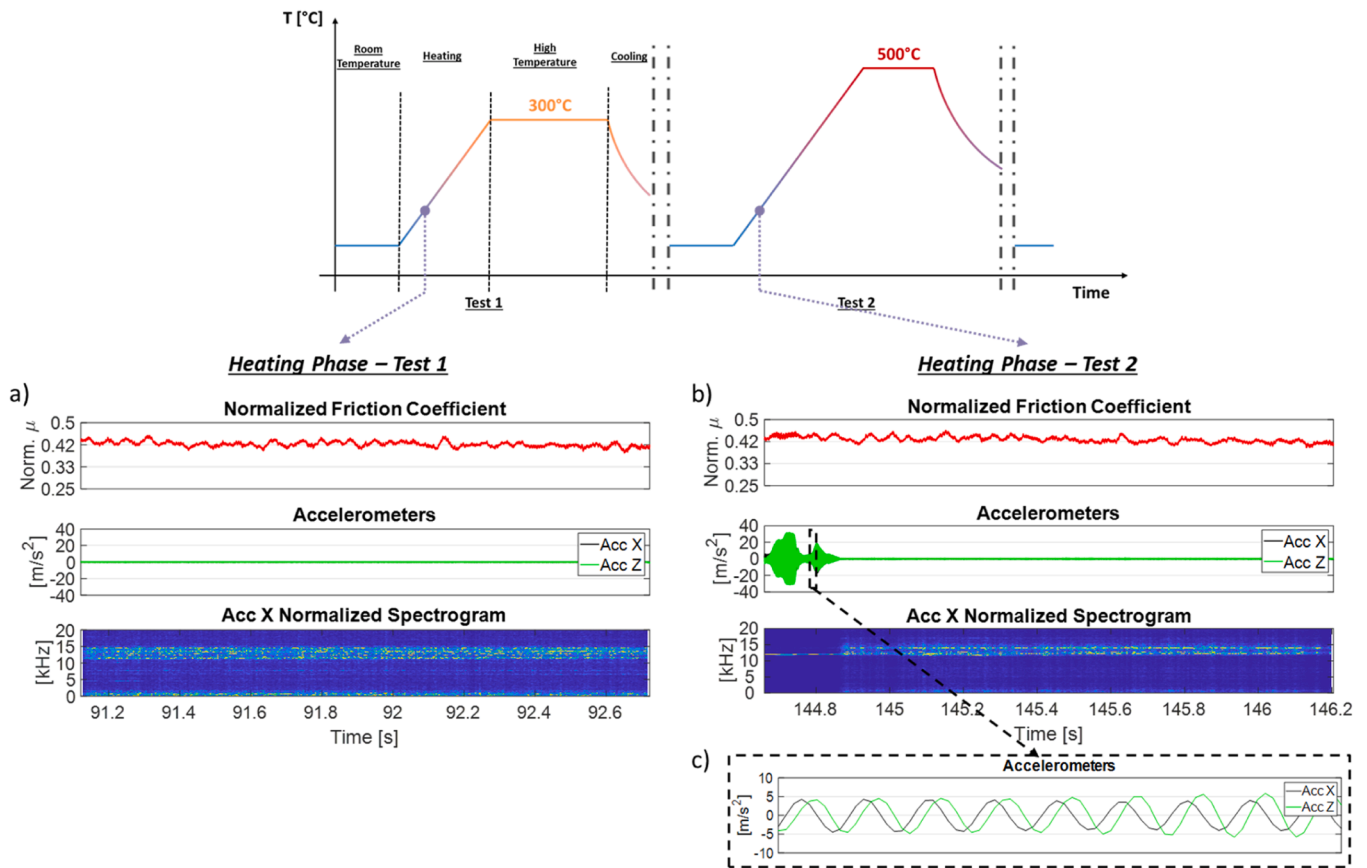


Fig. 11. Macroscopic frictional and dynamic response of the system observed during the heating phase ($\approx 75^\circ\text{C}$) of first (a, stable response, before thermal activation of surfaces) and second (b, unstable 12 kHz response, after thermal activation of surfaces) tests. A detail of the phase shift between normal and tangential acceleration is reported (c).

The recorded signals have been firstly processed to analyze the overall frictional and dynamic response of the system, observed for each frictional stroke. Since the friction coefficient is one of the most important parameters for mode coupling occurrence [41], its normalized value is evaluated over the time. The dynamic response of the system is analyzed by the accelerometers time signals and the associated normalized spectrogram, which allow to identify the occurrence of mode coupling instability, characterized by high vibration amplitude (time acceleration signal) and harmonic frequency content (normalized spectrogram) at the unstable mode frequency. Fig. 5 presents two examples of the frictional and dynamic response obtained in the case of a stable (left) and unstable (right) dynamic behavior of the investigated frictional system.

To characterize the friction noise, the power spectral density (PSD) of the velocity signal, recorded by the laser vibrometer, has been evaluated. The signal is previously filtered with a high pass filter at 1 kHz, because the lower frequencies are out of the brake squeal frequency range of interest. Since the PSD provides an estimation of the power of the signal spread over the frequency domain, an integration of the PSD in 5 kHz frequency bands allow to obtain an estimation of the energy content [mm^2/s^2] within each band, providing an immediate view of the friction noise magnitude and overall frequency distribution. The evaluation of the energy content is then repeated for several sliding strokes (at least five), recorded under the same boundary conditions (velocity, load and temperature). The average value and the standard deviation of each energy band have been obtained, as shown in the example in Fig. 6.

Before the analysis, the suitability of the developed setup to characterize the low amplitude vibrations of the friction noise has been evaluated analyzing the signal to noise ratio with respect to the background noise of the measurement chain. Fig. 7 shows the energy bands

obtained integrating the PSD of the laser signal in three different conditions: a) when the linear magnetic motor is activated, but no sliding contact is imposed between the samples (background noise), b) when friction noise is measured during sliding, in presence of a stable dynamic response of the system and c) in correspondence of the occurrence of an unstable response (mode coupling instability).

The three analyzed contact scenarios result in energy bands with completely different amplitudes, with a difference of several orders of magnitude between each one. The background noise of the measurement chain (10^{-6}), therefore, can be neglected with respect to the friction noise (10^{-3}) recorded during stable friction induced vibrations. On the other hand, during the unstable frictional response (mode coupling) the velocity signal result in a much higher energy content, characterized by harmonic spectrum with a dominant energy band (10 - 15 kHz), where the unstable mode occurs.

4. Friction noise vs squeal occurrence

The friction coefficient is one of the key parameters for the occurrence of mode coupling instability [42] and it is often numerically used as a critical index for predicting the onset on squeal. Consequently, for a given system, under the same boundary conditions, it would be expectable to have the same type of dynamic response (either stable or unstable) when the same value of the friction coefficient is registered.

Nevertheless, for equivalent friction coefficients, a certain amount of unpredictability is observed experimentally. As an example, in Fig. 8, three different dynamic responses of the investigated system have been obtained, under the same boundary conditions and in correspondence of same values of the friction coefficient. The signals presented in Fig. 8 have been obtained testing the same contact pair, under the same

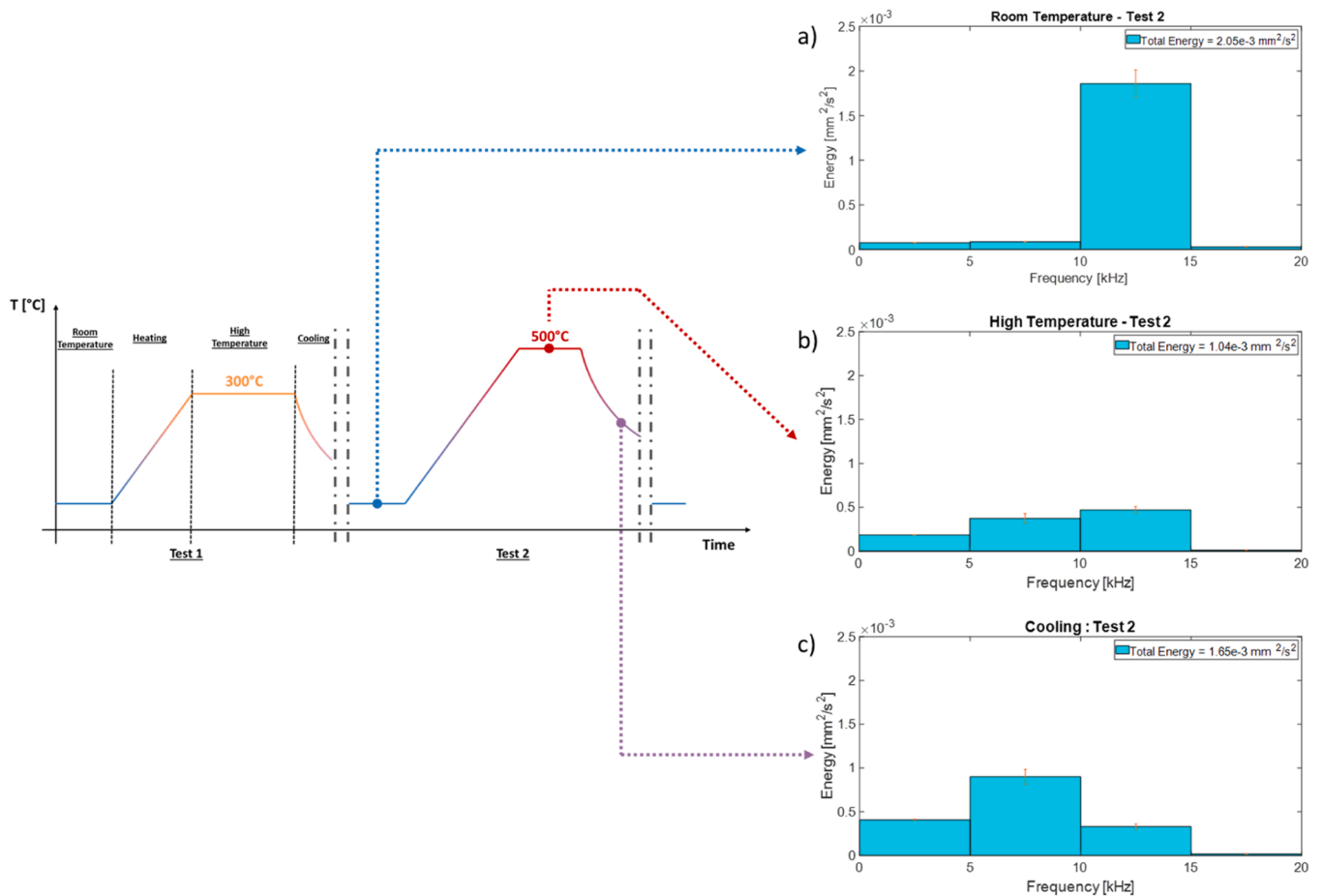


Fig. 12. Friction noise values measured at room temperature (a), high temperature (b, 500 °C) and during cooling (c) of the second test. A strong redistribution of the friction noise frequency content towards lower frequencies is observable.

velocity and load conditions, when only the temperature varies: a) heating phase of first test, at 75 °C, before the 300 °C tests; b) heating phase of second test, at 75 °C, after the 300 °C test; c) room temperature after cooling down from the 500 °C high temperature test. For the same value of the overall friction coefficient, the system results to be stable (a), unstable with a mode coupling at 12 kHz (b) and unstable with an unstable mode coupling at 5 kHz (c).

4.1. Triggering of the instability

To provide an explanation to this apparently random occurrence of brake squeal, the results presented in the following investigate the friction noise as a key factor to take into account when dealing with mode coupling instability.

To analyze this possibility, firstly, in Fig. 9, it is provided a comparison between the macroscopic frictional and dynamic responses observed at room temperature conditions, before and after heating up to 300 °C. It can be noted that the macroscopic friction coefficient was not modified by the passage at high temperature conditions of the C/C samples, as well as the dynamic response of the system, observed in both cases to be stable.

Nevertheless, while the macroscopic frictional and dynamic response is stable, with same value of the macroscopic friction coefficient, the analysis of friction noise revealed an important increase of the friction noise in the frequency band 10–15 kHz after heating up the system at 300 °C and cooling down to room temperature, as shown in Fig. 10.

This increase of friction noise has been correlated with the surface physio-chemical activation of the carbon-carbon material [43,44], occurred during the high temperature test (300 °C), that led to stronger

physio-chemical interactions between asperities, enhancing the amplitude of the contact dynamic excitation.

During the second test, when the heating phase has started, it has been recorded the occurrence of an unstable response, differently from what happened during the heating phase of the first test. In fact, comparing the behaviors observed during the heating phases of the first and of the second tests, as shown in Fig. 11, it can be observed that two different types of dynamic responses of the system occurred, even if in presence of the same value of the macroscopic friction coefficient.

The 12 kHz unstable vibrations registered during the second test (Fig. 11b) have been identified to be caused by mode coupling instability, since a 90° phase shift between normal and tangential acceleration [39] has been recorded (see zoom of acceleration signals in Fig. 11).

Recalling the friction noise energy bands presented in Fig. 10, it can be observed that 12 kHz mode coupling instability occurred only when in presence of a high value of friction noise within the frequency band of the unstable mode (10–15 kHz). During the first test, when the friction noise content was lower, the dynamic response of the system remained stable, even with the same value of macroscopic friction coefficient. It can be hypothesized, therefore, that in correspondence of same frictional response, a different dynamic excitation coming from the contact interface, identified by a different spectrum of friction noise, can trigger or not the mode coupling instability (squeal), which is a nonlinear dynamic instability and need a certain level of excitation to be triggered [45–48].

4.2. Squeal frequency shift with friction noise

In order to support the hypothesis presented above, successive

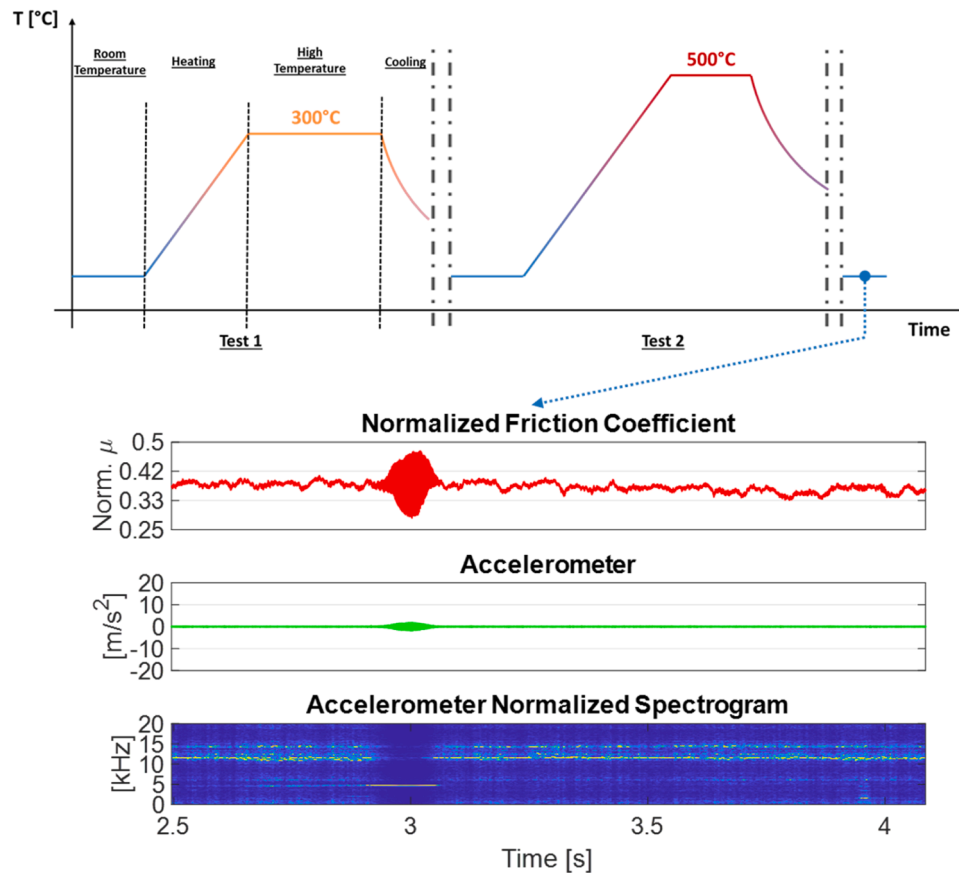


Fig. 13. Macroscopic frictional and dynamic response of the system observed successively to the friction noise frequency shift. A lower frequency mode coupling instability (≈ 5 kHz, instead of ≈ 12 kHz previously recorded one) is observable around $t = 3$ s.

observations and analyses have been performed heating up the contact pair to a maximum temperature of 500°C . In fact, a strong redistribution of friction noise frequency content has been observed when heating up to 500°C and cooling down to room temperature conditions, as shown in Fig. 12.

Despite the total energy associated to the friction noise excitation in the frequency range 1–20 kHz slightly changed during the test, its frequency content has been observed to strongly redistribute towards lower frequencies. The second band (5–10 kHz) becomes the most excited one. This behavior can be the result of longer (in time) and stronger (in amplitude) physio-chemical interactions between the asperities in contact, associated in the frequency domain with a larger low frequency content.

Consequently, the shift of the friction noise towards lower frequencies has been associated to the occurrence of unstable vibrations at lower frequency (about 5 kHz), as shown in Fig. 13. A new unstable mode of the system is triggered and becomes unstable, in the frequency range in which the friction noise (dynamic excitation) is the highest.

5. Conclusions

Few works in literature have investigated the relationship between friction noise (dynamic excitation coming from the contact) and the occurrence of brake squeal. In [37] the friction noise spectra, measured on a reduced scale test bench for different brake lining materials, has been correlated to the occurrence of squeal events on a full scale brake system, pointing out a consistent correlation between the friction noise level and the squeal instability occurrence. In this work, the same approach has been adopted, but both the friction noise and the instability occurrence have been measured on the same experimental system. The used approach allows to perform a direct comparison between the

friction noise evaluated in the range 1–20 kHz and the observed macroscopic dynamic response of the system, with a well-controlled system dynamics.

As described in Fig. 11 and Fig. 13, under the same applied boundary conditions with the same overall frictional response, three different dynamic responses of the system have been observed. These observations, firstly, confirm the unpredictability of brake squeal occurrence, suggesting that, even in presence of the same macroscopic friction coefficient, an appropriate dynamic excitation is required to trigger the mode coupling instability, namely the friction noise.

A correlation has been here pointed out between the magnitude and frequency distribution of the friction noise and the occurrence of unstable events, caused by mode coupling instability:

1. Under the same applied conditions, with same values of macroscopic friction coefficient, mode coupling instability has been observed to happen only when high values of the friction noise are recovered in the frequency band of the resulting unstable mode.
2. The shift of the energy content of the friction noise is associated with the frequency shift of the squeal instability to a mode within the respective frequency range.

As a main outcome, it can be pointed out that the brake squeal occurrence depends not only from the well-known macroscopic parameters reported in literature (friction coefficient, normal load, system dynamics etc. [18,20,29]), but it depends also on the dynamic excitation generated at the contact scale, that can act as a trigger for the switch from a stable to an unstable dynamic response.

CRedit authorship contribution statement

S. Ciprari: Writing – original draft, Software, Project administration, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **D. Tonazzi:** Writing – review & editing, Visualization, Validation, Supervision, Software, Resources, Project administration, Methodology, Investigation, Funding acquisition, Formal analysis, Conceptualization. **V. Ripard:** Writing – review & editing, Visualization, Validation, Supervision, Resources, Methodology, Funding acquisition. **A. Saulot:** Writing – review & editing, Visualization, Validation, Supervision, Resources, Funding acquisition. **F. Massi:** Writing – review & editing, Validation, Supervision, Resources, Project administration, Methodology, Investigation, Funding acquisition, Conceptualization.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data Availability

The data that has been used is confidential.

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