Receptance based structural modification in a simple brake-clutch model for squeal noise supression

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Abstract

Unlike brake squeal, brake-clutch squeal has rarely been studied, even though the cause of squeal noise is identical — dry friction acting at the contact interface. Similarly, brake-clutch squeal is difficult to get rid of. In this paper, a theoretical and experimental combined study is reported on squeal noise of a brake-clutch. On the theoretical side, a finite element model of a simplified brake-clutch is created and complex eigenvalue analysis is conducted to predict unstable frequencies associated with squeal noise. Then a receptance-based inverse dynamic method is adopted to identify the mass or stiffness required to split the coupled modes of the brake-clutch to achieve noise suppression. On the experimental side, a simplified brake-clutch is first tested to reveal its noise characteristics and one squeal frequency is found. Then the stiffness that has been theoretically identified for squeal suppression is implemented on the brake-clutch test rig in the form of a grounded spring and it is thus shown that the actual structural modification has removed the squeal noise. It is believed that this is the first time that a theoretically derived structural modification is made on a brake-clutch and shown to be capable of completely suppressing actual squeal noise. This study establishes a way of suppressing friction-induced high-frequency noise through structural modification.

Keywords: brake - clutch, friction, squeal suppression, receptance, structural modification, experimental validation

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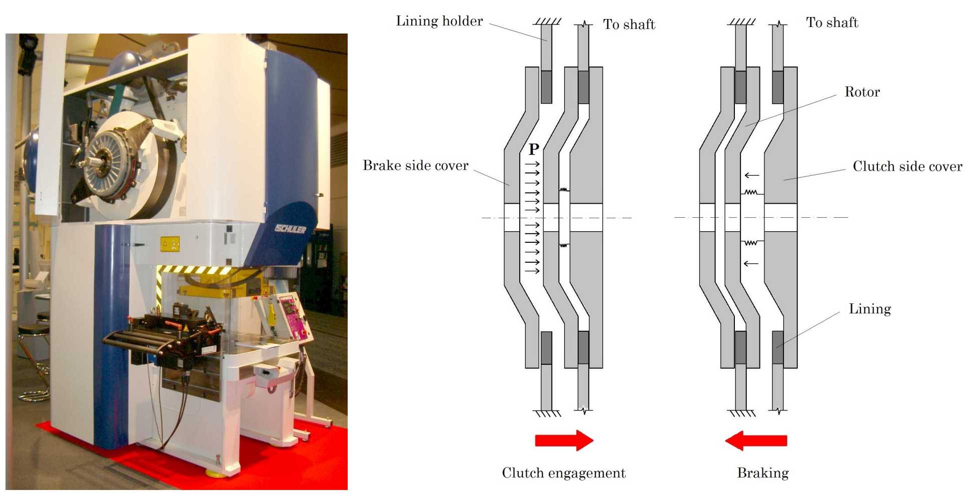
1. Introduction

Squeal is a major concern in industrial brake-clutch applications because of occupational health issues. Although the techniques used for squeal prediction and suppression in automotive brakes can be extended to analyse brake-clutch squeal, there is a need for a specific model that captures the distinctive features of this application, such as the ring shaped contact area and the asymmetric boundary conditions.

Typically, the role of a brake-clutch is to control the power transmission in a punch-press (Figure 1 (a)). When the clutch is engaged, it forces the flywheel and the shaft to spin together; in turn, when the brake is engaged, it couples the lining holder on brake side with the rotor to stop the system (Figure 1 (b)).



(a) (b)

Figure 1: (a)A brake-clutch in a punch-press, its typical application in industry; (b) schematic of the brake-clutch showing its main parts and functioning

For safety reasons, at rest position the brake is engaged thanks to the force exerted by some springs and it is necessary to apply either pneumatic or hydraulic pressure to start the clutch engagement manoeuvre.

Unlike in automotive brakes, the contact in brake-clutches occurs in a ring shaped area. Besides, as it is possible for the brake-clutch rotor to move axially, contact only takes place on one of its sides depending on the manoeuvre. Accordingly, contact stiffness is different on each side due to the two different pressure application methods.

Another difference is that the application time of the normal force is much shorter in a brake-clutch, lasting only few milliseconds. It is for this reason that brake-clutch squeal is short lived even if it reaches high sound pressure values.

Several brake squeal models can be found in the available literature, specially regarding automotive brake systems, either lumped [1] or distributed parameter models [2], or finite element models [3] [4], with most of them validated by experiments [5]. A comprehensive review of them can be found in [6].

As for clutch models, most papers about clutch noise focused on either low frequency noise [7] [8] [9] or thermoelastic effects [10] [11].

Regarding high frequency noise (up to a few kHz according to [12]), two models can be highlighted: (i) the 6-DOF squeal model presented in [13] in which mode coupling instability was studied and validated with experiments and (ii) the 2- DOF nonlinear model of a squealing clutch in [12] and [14] in which the effect of friction forces and gyroscopic action in mode coupling was analysed. Anyway, as stated in [15], nowadays there is no industrial model in the literature for studying squeal in clutch systems. With the brake-clutch being a combined unit, it is necessary for a useful model to comprise the features of both applications.

On the subject of squeal suppression, design modifications are the preferred means to achieve this objective. These modifications are mainly based on mode shapes [6] [16] and are generally performed following an iterative scheme [17] [18].

Another way to determine a modification for squeal suppression is using the receptance method. This method has been successfully applied in other fields to solve the inverse problem of structural modifications [19] [20]. Regarding squeal, only the direct problem has been solved this way. In [21], for instance, experimental receptances were used to compute the effect of the addition of a single-DOF system in a simplified test bench.

Therefore, the aim of this work is to take a step towards the development of a brake-clutch squeal model that can be used for squeal prediction and, then, apply in it a methodology for squeal suppression by either mass or stiffness point modifications based on the receptance method.

The paper is organised as follows: first, the brake-clutch squeal model developed is described; then, squeal tests are performed and the numerical model is validated; then, the receptance method is used to propose a point structural modification for squeal suppression and the results obtained are applied in a case study. Finally, some major conclusions are drawn.

2. Description of the simple model of the brake-clutch

The model used in this work and described in a previous work by the authors [22], has three distinguished features:

- *Simple geometry* of the theoretical model to avoid unnecessary details.
- *Experiments conducted on a commercial tribometer* because of its integrated sensors and control for rotating speed and pressure.
- *Representation of the particularities of the brake-clutch* such as the ring shaped contact area and the asymmetric boundary conditions.

In order to fulfill the objectives, two parts are designed taking as a starting point the components of the commercial tribometer Falex High Performance (Figure 2): the fixed part, which assumes the role of the brake-clutch rotor, and the mobile part, which represents either the clutch or the brake side and to which the friction material is attached.

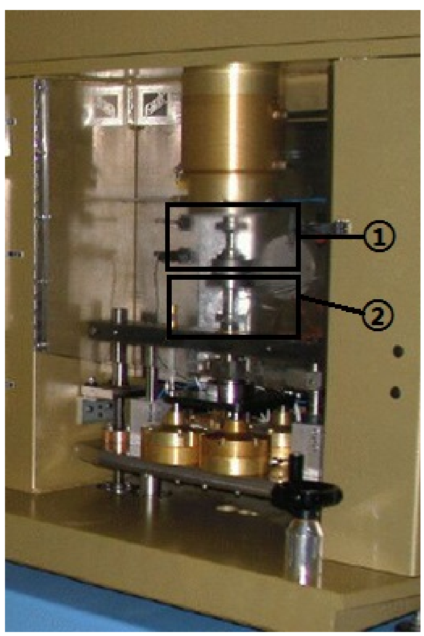


Figure 2: Falex High Performance tribometer. (1) Mobile part (2) Fixed part

The fixed part is designed in order to confine the unstable modes in the frequency range of interest (1 kHz – 20 kHz) in its upper part (Figure 3). In doing so, independence of the boundary conditions at the base of the fixed part can be achieved. The mobile part is intended to be as simple and robust as possible. Having one part fixed and the other mobile permits a relative motion between the two of them while keeping boundary conditions simple.

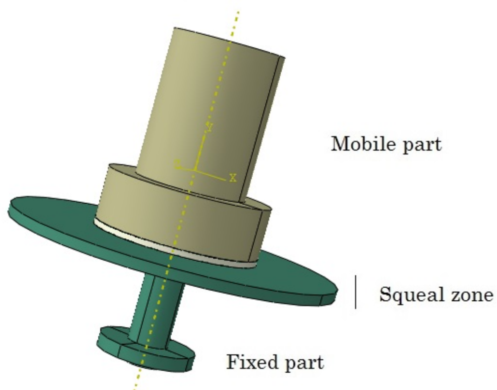


Figure 3: Design of the simple brake-clutch model

A finite element model of the system is developed in ABAQUS. A Complex Eigenvalue Analysis is performed to identify the unstable vibration modes of the system following the process described in [23] and summarised in [24] as:

1. Nonlinear static analysis for applying brake-line pressure and establish the contact area
2. Nonlinear static analysis to impose rotational speed on the disk so that friction is developed on the contact interface
3. Normal mode analysis to extract natural frequency of the undamped system
4. Complex eigenvalue analysis that incorporates the effect of friction coupling and produce the system eigenvalues

The finite element model has 83217 degrees of freedom and incompatible mode linear hexaedra elements are used because of their good response to bending and their moderate computation time [25]. The mesh was previously validated by an EMA of the components in static conditions correlating 13 modes with a maximum error lower than 10% and a MAC value higher than 0.7.

Pressure and speed dependent friction coefficient, friction damping and orthotropic properties for the friction material characterised using the SAEJ3013 standard [26] (Table 1) are included in the FE model in order to reduce over-prediction.

Table 1: Elastic properties obtained following the updating process described in the SAEJ3013 standard

E_1	6.92 GPa
E_2	6.92 GPa
E_3	2.42 GPa
ν_{12}	0.27
ν_{13}	0.309
ν_{23}	0.296
G_{12}	4.87 GPa
G_{23}	2.15 GPa
G_{13}	2.39 GPa

The complex eigenvalues obtained after following this process are presented in Figure 4.

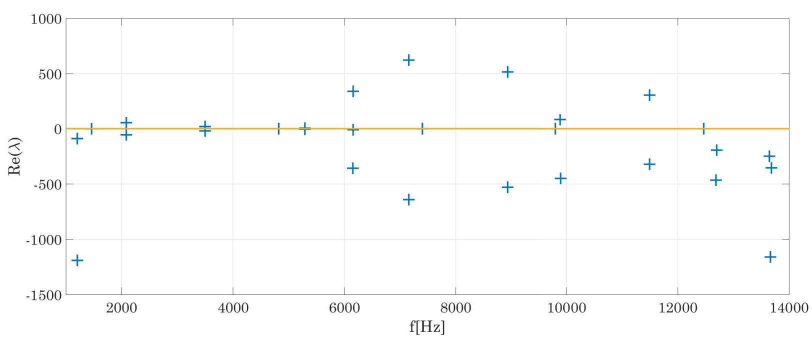


Figure 4: Results of the Complex Eigenvalue Analysis

3. Identification of the squeal frequency

3.1. Experimental setup

After designing the experimental setup, squeal tests are performed in order to check the validity of the FE model. First, pressure is applied and once the target value has been reached, rotation speed is imposed. Each test was repeated 3 times to ensure repetitiveness.

Normal force, torque, rotation speed and temperature measurements are provided by the tribometrer itself. Acceleration and sound pressure, in turn, are measured using external sensors as shown in Figure 5.

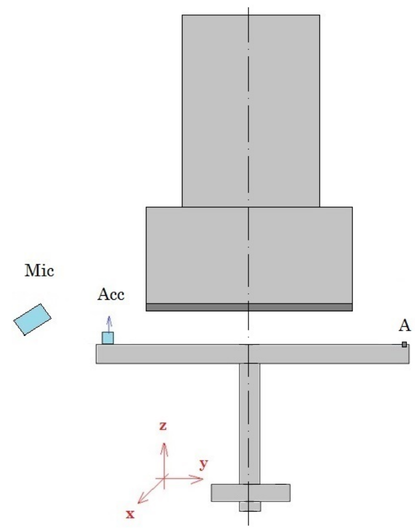


Figure 5: Positions of accelerometer and microphone

Several normal force and rotation speed combinations have been tested [22], but the particular case of 470N and 50rpm will be taken as a case study since a single dominant squeal frequency of 7.5kHz occurs in this case.

3.2. Correlation with FE simulation

The experimental squeal frequency of 7.5kHz corresponds to the simulation mode that has six nodal diameters and a frequency of 7.4kHz (Figure 6).

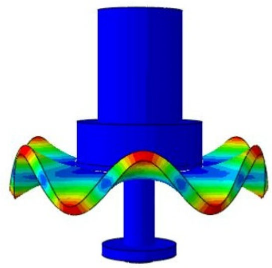


Figure 6: Shape of the unstable squealing mode

In Figure 7 a comparison between the measured noise spectrum and the unstable frequencies is presented, even though some overprediction exists, the simulation results show good correlation with the experimental data.

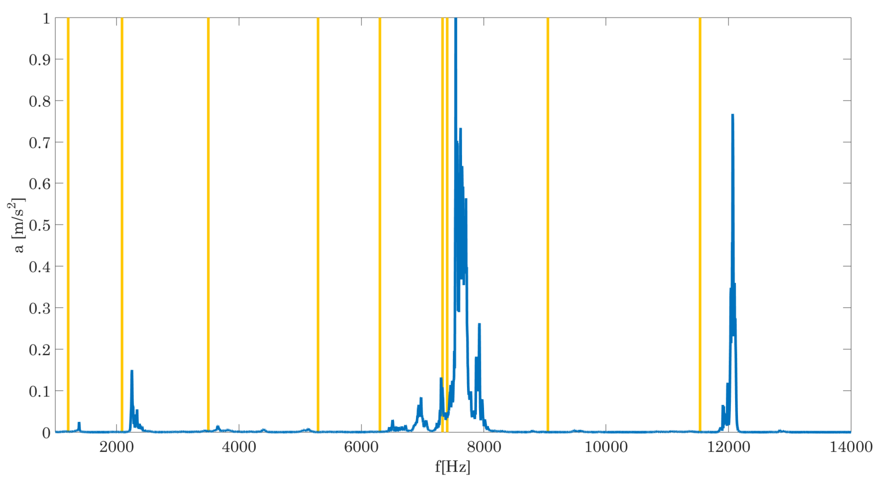


Figure 7: Comparison between CEA (vertical lines) and experimental squeal spectrum

Apparently there is an over-predication from CEA. This is expected as material damping is not included in the CEA runs, as a common practice in brake squeal analysis. On the other hand, importantly measured squeal frequencies are all well covered by CEA.

4. Development of receptance method for squeal suppression

After identifying the squeal frequency, a way to suppress squeal is investigated. Two possibilities are analysed: a point mass modification and a ground connected spring modification.

In this case, the additon of either a point mass or spring to the fixed part implies fixing the position of the nodal lines and separating the frequencies of the doublet mode [27]. With this in mind, a point in the periphery of the fixed part (point A in Figure 5) is selected and the needed mass or stiffness for shifting the frequency of the mode with an antinode in the position A is computed. The idea underneath is to obtain enough separation between the frequencies of the two modes so as to avoid mode coupling due to friction.

The amount of mass or stiffness is determined using the receptance similarly to the method proposed by Ouyang in [28] for structural modifications. The procedure is formulated below. The key is to express the receptance of the modified system by means of the receptance of the original system.

The equation of motion of the modified system in the Laplace domain can be written as follows:

$$\left[(\mathbf{M}+\mathbf{\Delta M})s^2+\mathbf{C}s+(\mathbf{K}+\mathbf{\Delta K})\right]\mathbf{X(s)}=\mathbf{0} \tag{1}$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are the mass, damping and stiffness matrices of the original system; $\mathbf{X}$ is the Laplace transform of the the nodal displacement vector of the system; $\mathbf{\Delta M}$ and $\mathbf{\Delta K}$ are respectively the modifications to the mass and stiffness matrices.

Taking into account that the receptance matrix is defined as $\mathbf{H}(s) = (\mathbf{M}s^2+\mathbf{C}s+\mathbf{K})^{-1}$, then Equation 5 can be rewritten as:

$$\left[\mathbf{I}+\mathbf{H}(s)(\mathbf{\Delta M}s^2+\mathbf{\Delta K})\right]\mathbf{X(s)}=\mathbf{0} \tag{2}$$

For N point masses, $\mathbf{\Delta M}$ can be expressed as:

$$\mathbf{\Delta M}=\sum_{i=1}^{N}\Delta m_i\mathbf{e}_i\mathbf{e}_i^{\intercal} \tag{3}$$

where $\mathbf{e}_i$ is a vector whose elements are zero except the ith element that is 1 and corresponds to the degree of freedom affected by the point modification and Δm_i is the ith added point mass.

Similarly, for M ground connected springs, $\mathbf{\Delta K}$ can be expressed as:

$$\mathbf{\Delta K} = \sum_{i=1}^{M} \Delta k_i \mathbf{e}_i \mathbf{e}_i^{\intercal} \tag{4}$$

where Δk_i is the ith added grounded spring.

Introducing these two terms in Equation 2 yields:

$$\left[\mathbf{I} + \mathbf{H}(s)\left(s^2 \sum_{i=1}^{N} \Delta m_i \mathbf{e}_i \mathbf{e}_i^{\intercal} + \sum_{i=1}^{M} \Delta k_i \mathbf{e}_i \mathbf{e}_i^{\intercal}\right)\right] \mathbf{X}(\mathbf{s}) = \mathbf{0} \tag{5}$$

This system will only have a solution other than the trivial if:

$$\det\left(\mathbf{I} + \mathbf{H}(s)(s^2 \sum_{i=1}^{N} \Delta m_i \mathbf{e}_i \mathbf{e}_i^{\intercal} + \sum_{i=1}^{M} \Delta k_i \mathbf{e}_i \mathbf{e}_i^{\intercal})\right) = 0 \tag{6}$$

If the modifications are conducted at a degree-of-freedom A, the needed point mass for shifting an existing frequency to a desired frequency ω_h can be computed as:

$$m = -\frac{1}{s^2\, h_A(\omega_h)} \tag{7}$$

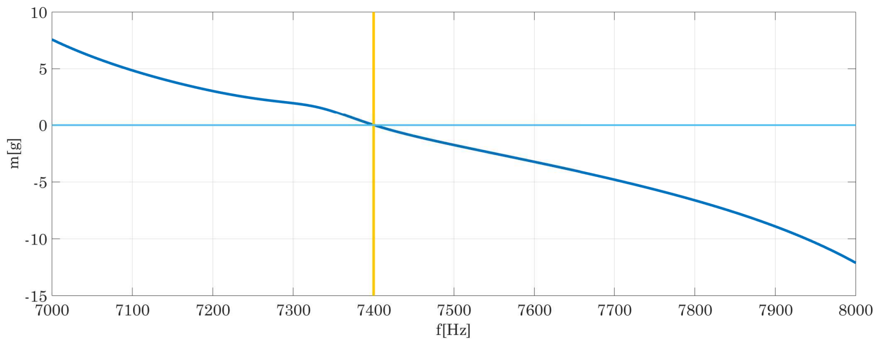
And, similarly the needed ground connected spring will be:

$$k = -\frac{1}{h_A(\omega_h)} \tag{8}$$

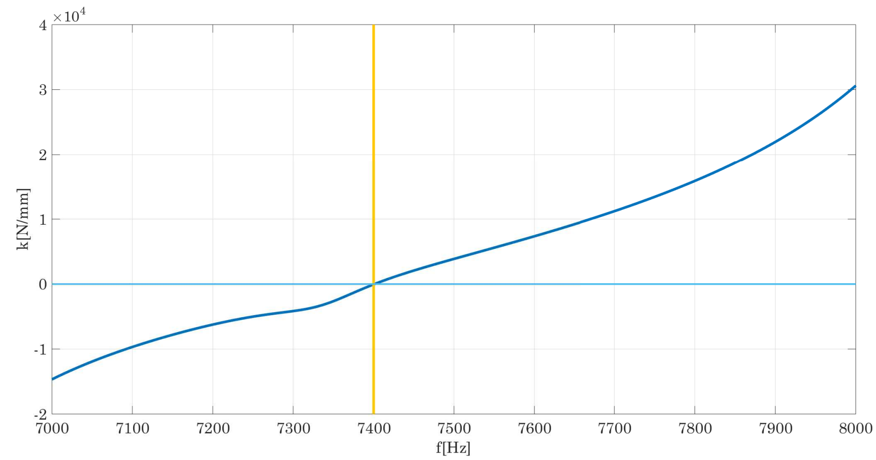
where $h_A(\omega_h)$ stands for the point receptance at the degree-of-freedom A for the desired frequency.

In the case under study, simulation receptances are used as they already include the effect of friction in the stiffness matrix [29]. The process for computing them is equivalent to the one used for complex modes in Section 2 but substituting the two last steps of the simulation with the computation of the response to an impulse force in the frequency domain.

Receptance is computed in the vicinity of squealing mode, from 7kHz to 8kHz, and using Equation 7 and Equation 8 the amount of mass or stiffness needed to assign a frequency in the range is obtained (Figure 8).



(a) Needed mass

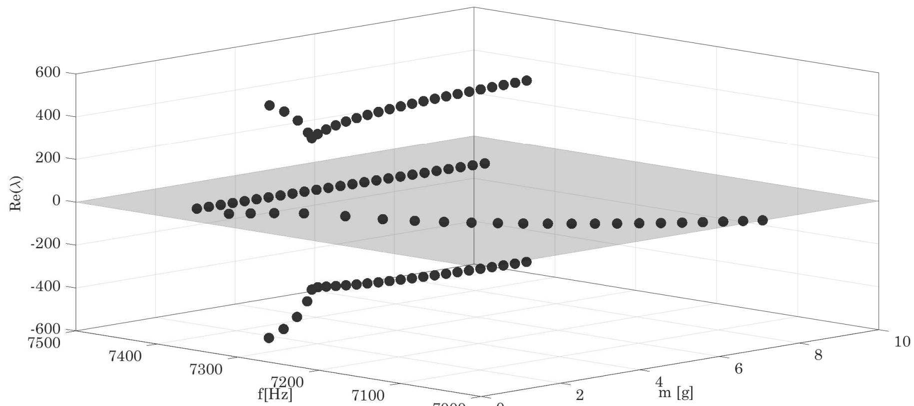


(b) Needed stiffness

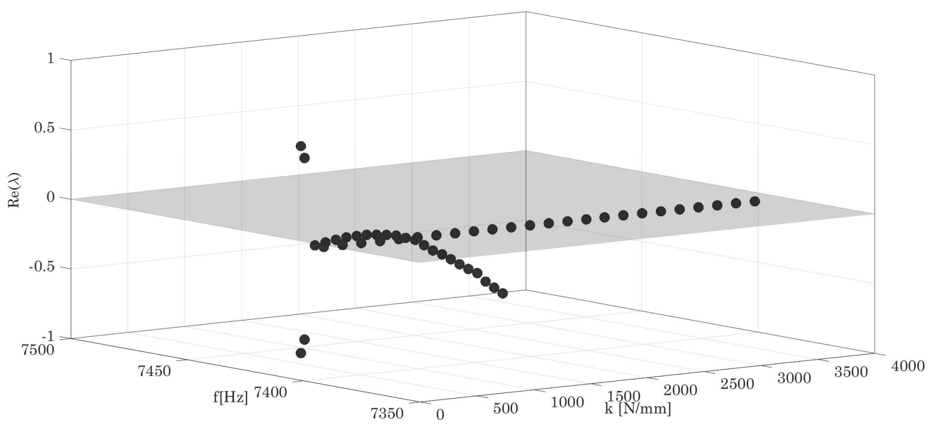
Figure 8: Needed point modification to assign a certain frequency

Imaginary part of the receptance h is neglected for both options because in this case it represents the amount of negative damping needed to shift the eigenvalue along the real axis of the complex plane, as only frequency (not real part) of the eigenvalue has been assigned.

Once the range of the parameters is obtained, two parametric studies are performed to check the effect of the modifications in complex modes: one for the case of adding a single point mass shown in Figure 9 (a) and the other for the case of adding a ground connected spring shown in Figure 9 (b). In this case, a point mass cannot be used to avoid coupling since it only aggravates the problem as it causes another unstable mode to appear. Therefore, a tool to include a ground connected spring in the test bench is designed.



(a) Stability diagram for different values of mass



(b) Stability diagram for different values of stiffness

Figure 9: Stability diagrams from the parametric studies

4.1. Experimental validation

According to the simulation, a k value greater than 1000 N/mm is enough to suppress squeal, but adding such a small value of stiffness is challenging from the point of view of design. With this in mind, a stiffness value of approximately 7000 N/mm is selected for the ground connected spring since such a value does not cause other modes to coalesce to result in a new unstable mode.

The desired value of stiffness is accomplished by means of a 0.8 mm diameter steel wire of length of 14.6 mm, that adds 6890 N/mm to the system. The setup is shown in Figure 10 and simply consists of a support for the steel wire. This tool is firmly fixed to the structure of the tribometer, which in this case serves as the ground. The steel wire can be preloaded by tightening the screw on top so it is also effective under compression.

Figure 10: Setup including a ground connected spring

Squeal tests are repeated on the modified system. The addition of the spring to the system causes the squeal to disappear and its removal takes squeal back as it can be seen in the spectrogram in Figure 11.

5. Conclusions

This work presents a theoretical and experimental study of a simple model for brake-clutch squeal prediction and suppression. On the one hand, a finite element model of a simplified brake-clutch is developed and complex eigenvalue analysis is performed in order to identify the possible squeal frequencies. On the other, squeal tests are performed in the test rig and a single squeal frequency is found.

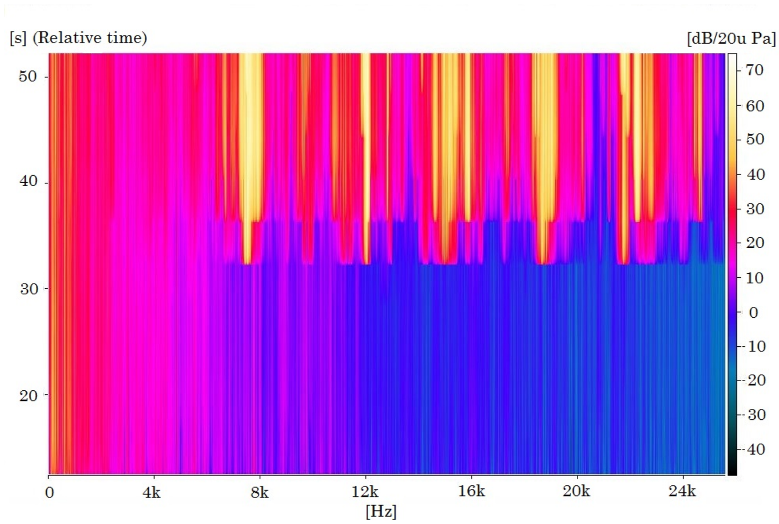


Figure 11: Spectrogram of squeal test with added ground connected spring. At second 32 the tool is removed and squeal comes back

Since the correlation between theoretical and simulation results is good, the needed point modification is computed by a receptance-based inverse dynamic method using simulation receptances including contact.

Then, the computed stiffness is added to the test rig in the form of a ground connected spring so as to validate the simulation results, with the outcome of suppressing actual squeal noise. Thus, it is proved that a methodology comprising the computation of simulation receptances followed by a parametric study of the effect of the values of the point modifications is useful to determine a structural modification to eliminate squeal noise in the designed test bench.

This study is believed to be the first to be capable to suppress squeal in a brake-clutch model using a theoretically derived structural modification.

6. Acknowledgements

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