

# Finite element parametric study of the influence of friction pad material and morphological characteristics on disc brake vibration phenomena

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**Abstract.** Since nowadays the NVH performance of vehicles has become an important priority, the noise radiating from brakes is considered a source of considerable passenger discomfort and dissatisfaction. Creep groan and squeal that show up with annoying vibrations and noise in specific frequency ranges are typical examples of self-excited brake vibrations caused by the stick-slip effect, the former, by the mode coupling of brake disc and friction pads or calliper, the latter. In both cases, the friction coefficient, which depends, among other factors, on the morphology of the mating surfaces and on the operating conditions, is a fundamental parameter but not the only one for the occurrence of the vibratory phenomena.

Finite element complex eigenvalue parametric analyses were performed on a disc brake assembly to evaluate propensity to dynamic instability of brakes with multiple pads, as in railway brakes, as a function of the number of pads, pad shape and size, and material parameters.

## 1. Introduction

Since the beginning of its history the automotive industry has experienced a progressive increase of the standards on comfort of vehicles. As a result of this improvement and of the level of customers' demand, noise and unwanted vibrations are now responsible for considerable costs allocated during the vehicle warranty period [1]. Among the various sources of noise and vibration, the brake system is responsible for a substantial portion of the registered complaints. Despite all the research efforts and the updated technology, the noise phenomenon generated during braking still presents many knowledge gaps to be filled.

Brake noises are classified according to their frequency and elements responsible for its radiation as reviewed by Flint [2]. Other authors, such as Balvedi et al.[3], present a somewhat distinct classification, describing subcategories for these noises or providing other frequency ranges. These variations are understandable if one considers the different brake designs and vehicle systems studied by each author. In an effort to reduce the divergence of classification, the *Society of Automotive Engineering* (SAE) edited the standard SAE J2786 [4] that offers a wide and comprehensive classification. Regardless of the classification adopted, it is established that all the vibratory phenomena originate from the interaction between the pair of friction elements during the braking process.

Among the types of noise associated to brakes, squeal has certainly gained an increasing interest from researchers; it is a characteristic noise caused by friction induced, self-excited and self-sustaining (for a short period of time) vibrations [5]. Typically, high frequency squeal occurs in the frequency range from 8 to 16 kHz, while low frequency squeal from 1 to 7 kHz. Since the human ear is most sensitive in the frequency range from 1 to 4 kHz, low frequency squeal is considered the most annoying type of brake noise [6]. Squeal is a challenging issue for researchers and engineers since the 30's, because of its extreme complexity involving multiple disciplines such as nonlinear dynamics, contact mechanics and tribology/nanotribology. There are several theories to explain the squeal phenomenon well described in the review of Kinkaid *et al.* [7].

Noise research on brakes has led to several experimental and numerical analyses. Experimentally, the best way to study such a noise is to analyze the components of the braking system and its features. The two most important and studied components of the brake system are the friction pad and the disc. Besides the need to provide safety, the friction material is also required to satisfy comfort features related to noise. Despite the apparent simplicity, friction materials are complex combinations of different materials, aimed at safely generating repetitive and stable forces by friction in a wide range of brake conditions.



The performance of the pad depends on a large number of factors, such as irregularities on the friction surfaces between disc and pad, particle size variations of the friction material, geometry variations. Variations of the brake operating conditions such as applied pressure, braking start speed, disc temperature, ground conditions, climatic conditions, etc. also affect the pad behavior.

Brake disc materials are as important as the pad friction material. There are a number of limitations to the development of special materials for disc brake application, the main one being the cost. Among the material properties related to noise the following have to be considered: shape and homogeneity of the microstructure, chemical composition, hardness, tensile strength, and, in particular, Young's modulus, density, damping ratio that influence the disc natural frequencies and mode shapes.

Numerical as well as experimental analysis has great importance in the study on automotive brake noise. In recent years, due to the large computational advances, many researchers have been using the finite element method (FEM) to deal with the noise phenomenon in automotive brakes. The simulation and analysis of the squeal phenomenon can basically be divided into two large categories [8]: analysis of complex eigenvalues in the frequency domain and transient analysis in the time domain, the complex eigenvalues analysis requiring generally lower computational effort. The latter allows for the identification of mode coupling responsible for squeal instability.

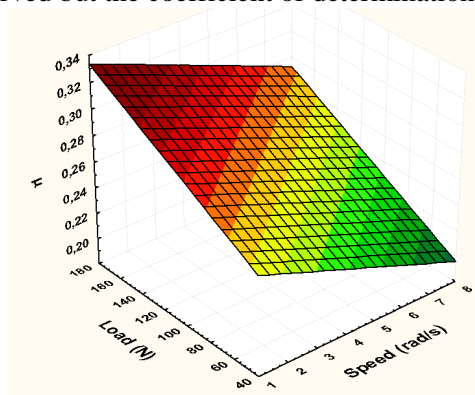
The present paper concerns a numerical analysis of a disc brake. This paper presents a FE complex eigenvalue parametric analysis performed on the brake assembly to evaluate the propensity to squeal instability of brakes with multiple pads as a function of the number of pads and of material parameters. The dependence of the friction coefficient on load and sliding speed based on correlations obtained in previous experimental tests [9] was taken into account.

## 2. Friction properties of the pad material

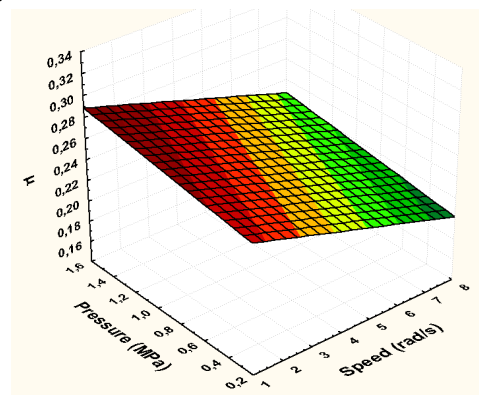
In order to properly set up the numerical simulations, the friction properties of the pad materials obtained by a series of experimental tests and taken from the literature are briefly summarized.

In a previous work the authors [9] presented the experimental characterization of the pad material carried out on a pin on disc tribometer with the brake disc in place of the tribometer disc and cylindrical samples of different size, made of the pad material, as pins. The tests were aimed at measuring the friction coefficient of the pad samples and at investigating possible different responses to load, area (nominal contact pressure) and rotational speed variations.

A reduction in the friction coefficient with speed was observed in all tests and for all samples, i.e. lower friction coefficients were obtained at greater speeds. A similar behavior between friction and speed was observed by Chowdhury et al.[10] and Dezi [11]. Regarding the normal load, the results indicate that higher applied loads yield an increase of the friction coefficient. This is shown in figure 1 which reports the best fit ( $R^2=0.81$ ) linear regression between load, speed and friction coefficient, based on the results obtained with all the samples. In particular, the regression equation is  $\mu=0.223 - 0.0082 \times \text{Speed}(\text{rad/s}) + 0.0005 \times \text{Load}(\text{N})$ . Figure 2 depicts instead the friction coefficient as function of the average nominal contact pressure and speed. In particular, the regression equation is  $\mu=0.27 - 0.0082 \times \text{Speed}(\text{rad/s}) - 0.0006 \times \text{Pressure}(\text{MPa})$ . A slight tendency of the friction coefficient to increase with increasing nominal contact pressure can be observed but the coefficient of determination is low ( $R^2=0.43$ ).



**Figure 1.** Linear regression of friction on load and speed for all samples ( $R^2=0.81$ ).



**Figure 2.** Linear regression of friction coefficient on pressure and speed for all samples ( $R^2=0.43$ ).

Friction materials can be considered as composite materials due to the high number of elements in their composition. Predominant components are organic and carbon based elements which are very sensitive to temperature variations, in terms of their elastic properties. In order to correctly evaluate the effect of this parameter on the lining material, Trichês [12] conducted experiments based on the Standard Test Method for Measuring Vibration-Damping Properties of Materials (ASTM E756-98). He noted that an increase in temperature induces a decrease in the Young's modulus and the experimental data could be well fitted by a linear function.

In the following simulations we'll refer to a friction pad coupled with a cast iron disc. The pad friction properties vary with load and speed while its elastic properties vary due to temperature variations.

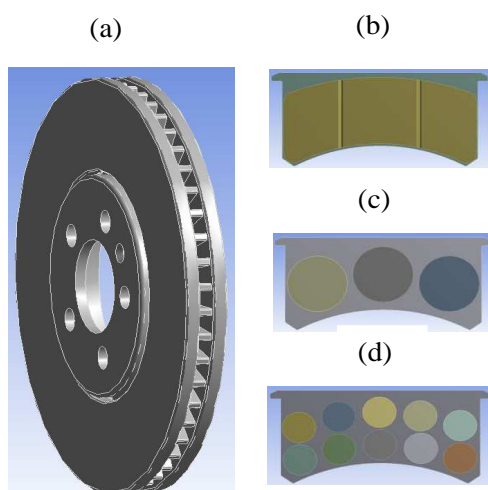
### 3. Finite Element Analysis

On the basis of the experimental results, numerical simulations in ANSYS environment were carried out in order to study the stability of the disc-pad system, by using the complex eigenvalue analysis. A decreasing friction coefficient with increasing sliding speed, such as that found in this application could also introduce instability due to negative damping and stick-slip effect, responsible of low frequency range brake noise. The present study does not take into account such a phenomenon, focusing on squeal high frequency vibrations ascribed to mode coupling, effectively identified by the adopted complex eigenvalue approach.

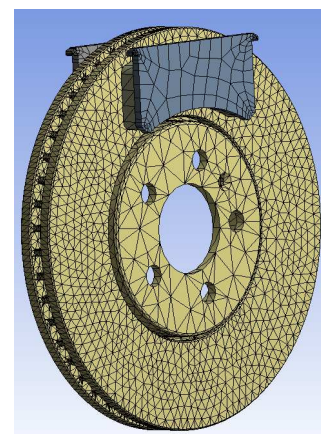
In particular, a complex eigenvalue modal analysis combined with a non-linear pre-stress analysis was carried out. Contact elements were used taking into account the sliding friction between the parts in contact producing a non-symmetric stiffness matrix. The QR damped eigenvalue extraction method was employed [13]. The imaginary component of the eigenvalues represents the damped frequencies. The real part of the eigenvalues indicates if the mode is stable, unstable modes having a positive real part of the eigenvalue. This method is usually considered effective in determining possible unstable modes and their corresponding frequencies as a source of acoustic discomfort.

#### 3.1. Modeling

For this type of analysis a FE model of the system composed of the disc and the friction pad was set up. The model is a compromise of accuracy and computational effort. The actual cast iron disc used in the experimental activity, with an average contact radius of 110 mm, was reproduced in the FE model (figure 3 a); pads of different shapes and sizes (figure 3 b, c and d) were modelled assuming linear elastic isotropic material properties. Three different models were developed based on commercial brake configurations: the first model was composed of a single pad with a contact area of 5107 mm<sup>2</sup>; model 2 was composed of three cylindrical pads with a total contact area of 3240 mm<sup>2</sup>; model 3 has ten cylindrical pads with a total contact area of 3243 mm<sup>2</sup>. These characteristics for pad configuration were selected in order to verify by simulation any correlation between the number of pads and the stability of the system.



**Figure 3.** Solid models. (a) Brake disc; (b) Pad model 1; (c) Pad model 2; (d) Pad model 3.



**Figure 4.** FE models of brake disc and pads.

The model in ANSYS (figure 4) was meshed with a total of 74414,72800 and 71751 nodes and 41630, 40837 and 40173 hexahedral elements respectively for models 1,2, and 3. The Sweep method was used to

generate a hexahedral dominant mesh of the brake system assembly. Brake discs, pads and all other associated components were meshed with 20-node structural solid SOLID186 elements with uniform reduced-integration element technology. The edge sizing tool was used to obtain a refined mesh at the pad-disc interface to improve the solution accuracy. CONTA174 (3-D 8 node surface to surface contact) elements were used to define the contact surface and TARGE170 (3-D target segment) elements were used to define the target surface.

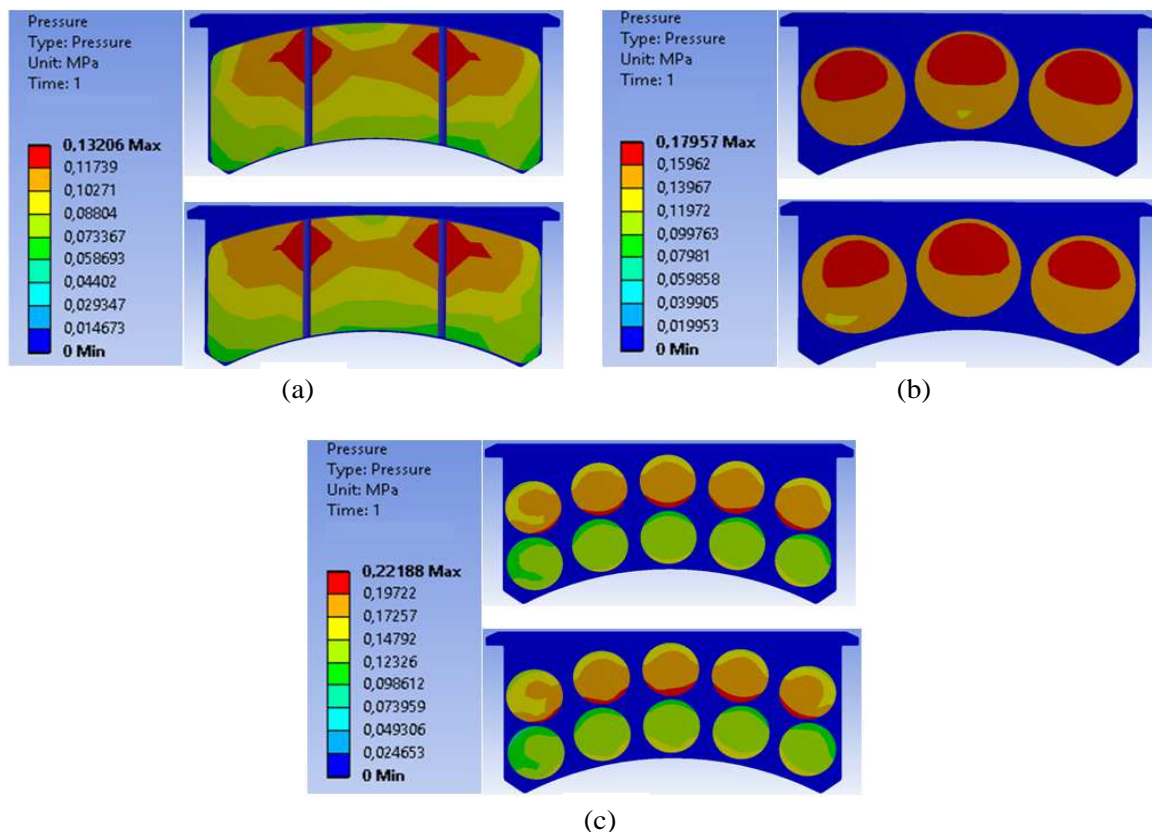
The augmented Lagrange algorithm was used for the frictional contact pairs because it requires a lower computational effort than the standard Lagrange multiplier algorithm, which normally requires additional iterations to stabilize the contact conditions, and also because it is well suited for modeling general frictional contact, such as the contact between the brake pad and the disc defined in this work. An internal multipoint constraint (MPC) contact algorithm, between the brake pad and the clamping plate, was used for bonded contact because it ties contact and target surface together efficiently for solid-solid assembly.

### 3.2. Simulation results

Initially, a simulation with all the three models was conducted to verify the behavior of the system. The material properties set in the simulation are listed in Table 1. Additional simulation parameters were the rotational speed of 10 rad/s, the load of 500 N and the coefficient of friction of 0.39 based on the regression on load and speed obtained experimentally (figure 1).

**Table 1.** Simulation material properties

	Pad lining	Pad back plate	Disc
<b>Young's modulus (GPa)</b>	0.37	200	110
<b>Poisson's ratio</b>	0.25	0.3	0.28
<b>Density (kg/m<sup>3</sup>)</b>	2770	7850	7200

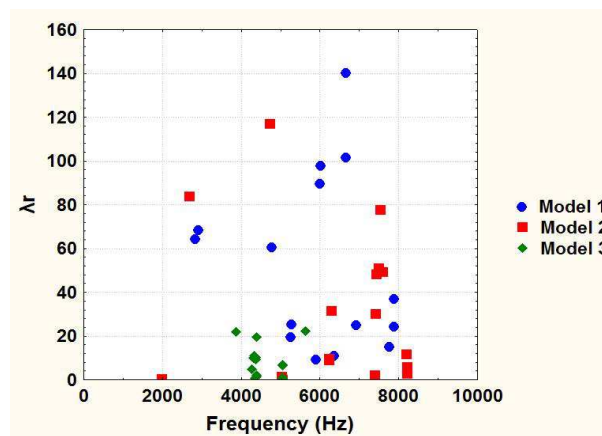


**Figure 5.** Pressure distribution for each pad model. (a) Pad model 1; (b) Pad model 2; (c) Pad model 3.

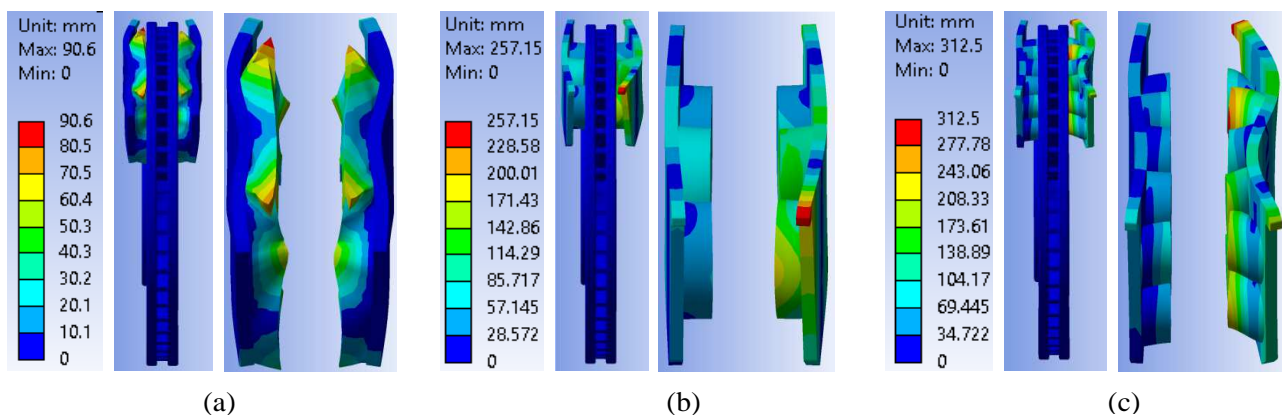


Figure 5 shows the pressure distribution for each model. It appears rather regular with maximum values of 0.13, 0.18 and 0.22 MPa, respectively, for models 1, 2 and 3, at larger contact radii. Figure 6 shows the different results among the models in terms of positive real part of eigenvalues as a function of frequency. Model 1 showed the highest number of instabilities, 22 in total, and the highest positive real part of the eigenvalue, while models 2 and 3 obtained, respectively, 20 and 19 points of instability. Model 3 showed the smaller positive real part of eigenvalues.

In addition, considering that the unstable points are between 2 and 8 kHz, all points of instability are potentially related to the frequency field of the squeal phenomenon. Model 3 showed a maximum real part value of 22.3 while the other models have most of their real part values above 40 with a maximum value of 140.1 and 116.9, respectively. Figure 7 shows the natural vibration modes corresponding to the maximum eigenvalue positive real part for each model where it is evident that the disc has slight out-of-plane vibrations while the pads have considerable out-of-plane vibrations for all models. Such a behavior was observed also when the parameters of the simulation were varied. A similar behavior was observed by Liu et al.[14] who deduced that the brake pads may be the source of the disc brake squeal. Therefore, in order to eliminate brake squeal, methods aimed at reducing the pad out-of-plane vibration are suggested such as the use of viscoelastic material (damping material) on the back of the back plates of the pads [15] or a modified shape of the brake pads to avoid the coupling between the pads and the disc [16].



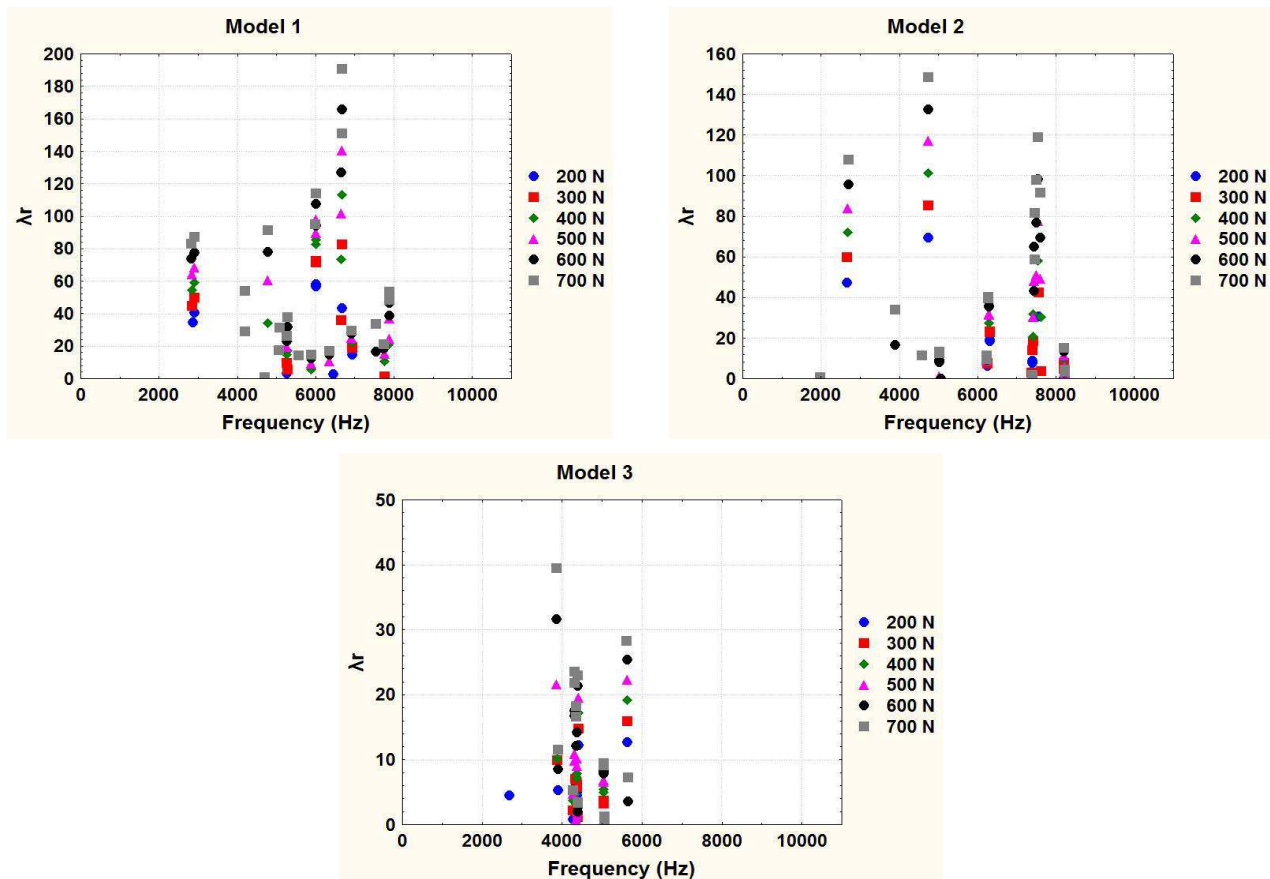
**Figure 6.** Eigenvalue positive real part vs frequency.



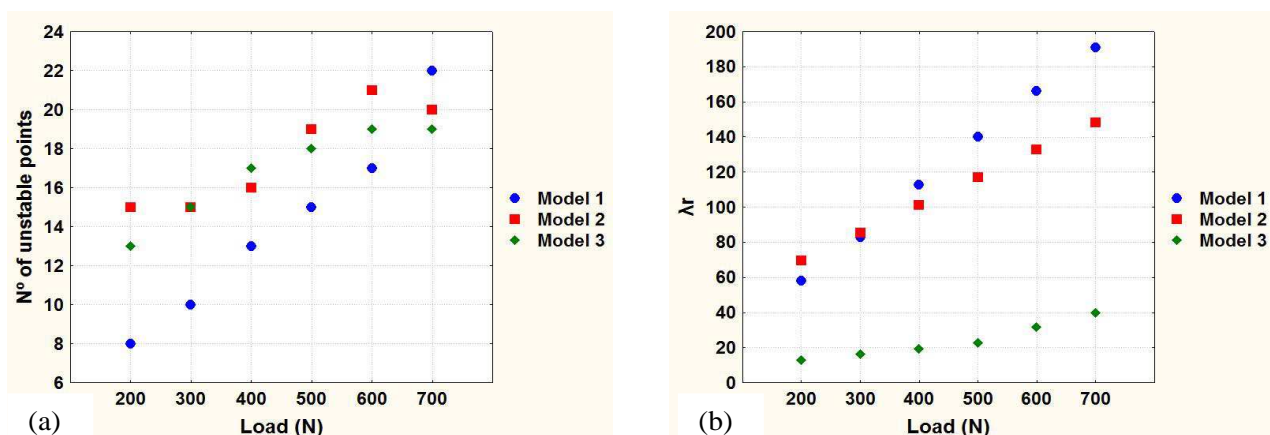
**Figure 7.** Natural vibration modes corresponding to maximum eigenvalue positive real part. (a) Pad model 1; (b) Pad model 2; (c) Pad model 3.

Then, simulations were performed varying some operative parameters such as disc speed, applied load and material properties such as pad Young's modulus in order to verify if there was a correlation of these parameters with the number of unstable points and maximum positive real parts of eigenvalues. Actually, the disc speed is an indirect simulation parameter in the sense that it affects the friction coefficient which plays a major role in the analysis. For each simulation, the friction coefficient was set on the basis of the regression depicted in figure 1 for the considered speed and applied load.

Figure 8 shows the effect of load on the brake system stability for each model. One can see that for all pad models the system unstable points are found in the frequency range of low squeal (2 - 8 kHz) and that model 3 has a reduced frequency range of instability of 4000-6000 Hz. In addition, it can be observed that, for all frequencies, the real part of the eigenvalue increases with the applied load. This is best assessed with reference to the following figure. Figure 9 a shows an increase of the number of unstable points while figure 9 b that of the maximum eigenvalue positive real part with load for all models.



**Figure 8.** Effect of load on the brake system stability for each model.

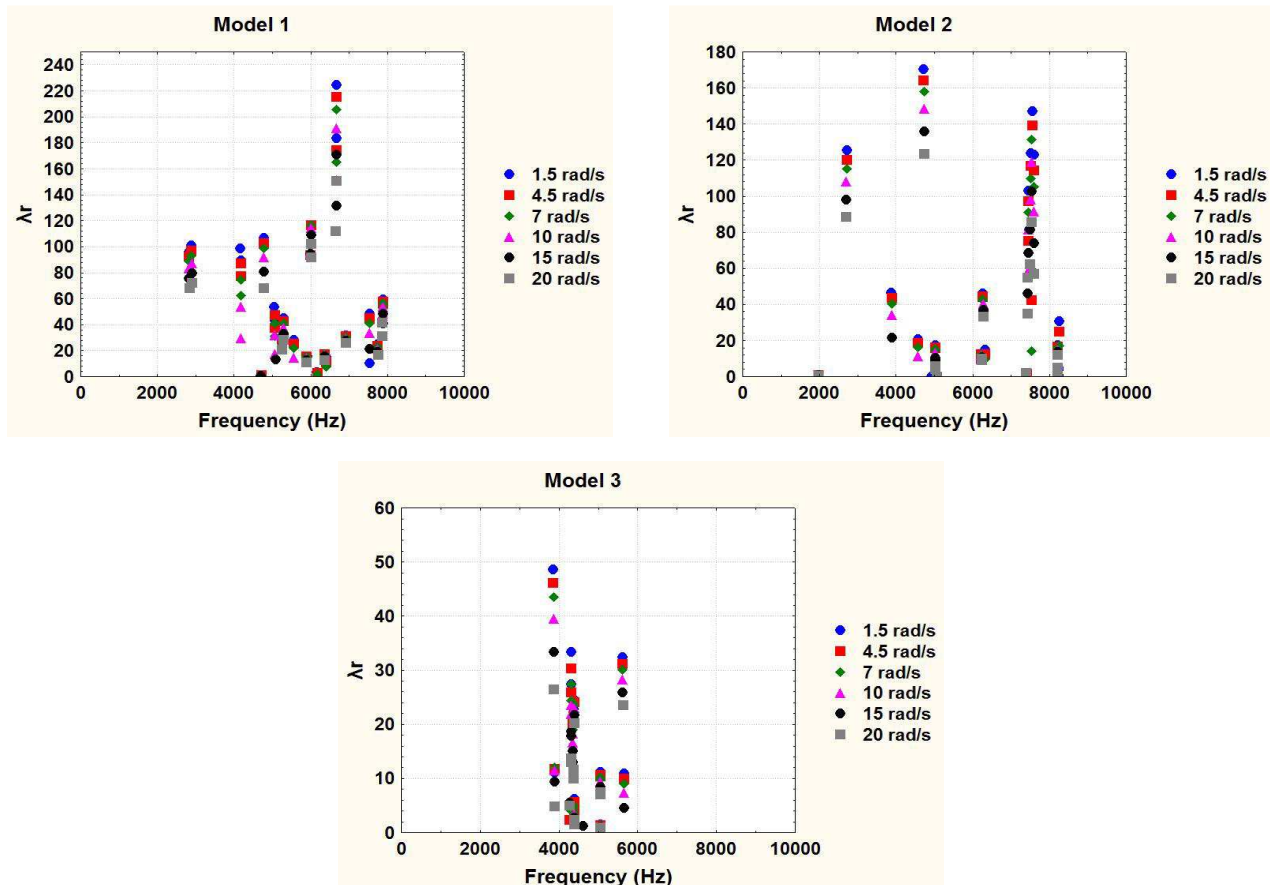


**Figure 9.** Effect of load: (a) no. of instability points; (b) maximum positive real part.

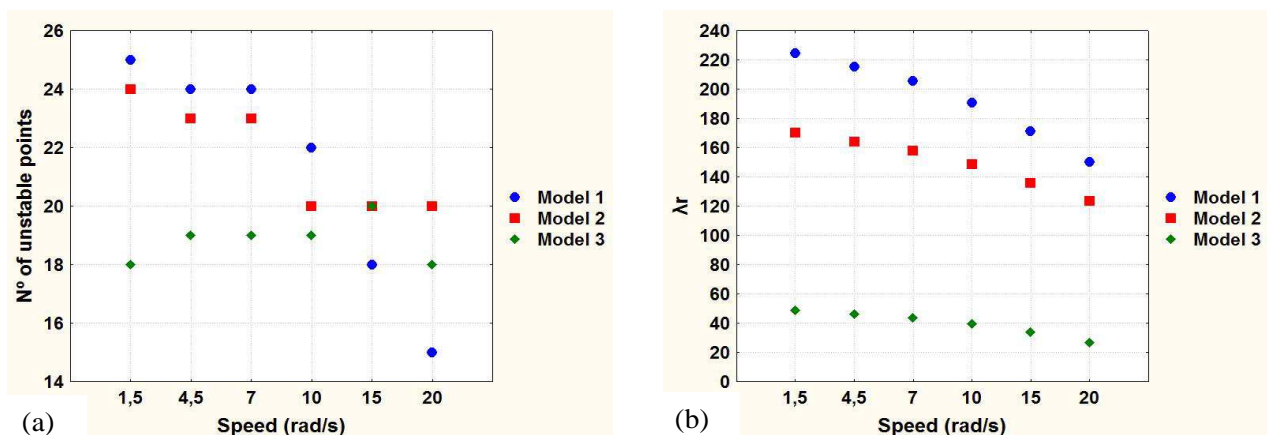
A similar analysis was conducted varying speed; also in this case, for all pad models, the system instability points are found in the frequency range of low squeal and model 3 has its unstable points in the

frequency range of 4000-6000 Hz (figure 10). As opposite to what happened with load variation, in this case a decrease in the number of unstable points and in the maximum value of the real part were related to an increase of the speed (figure 11).

The results presented for the variation of load and speed appear quite consistent given that instability is related to the friction coefficient and that an increase of friction promotes instability [11]. In fact, as we have a positive and negative variation of friction, respectively, with the variation of load and speed (figure 1), it is reasonable to obtain a similar variation with respect primarily to the maximum value of the real part.



**Figure 10.** Effect of speed on the brake system stability for each model.



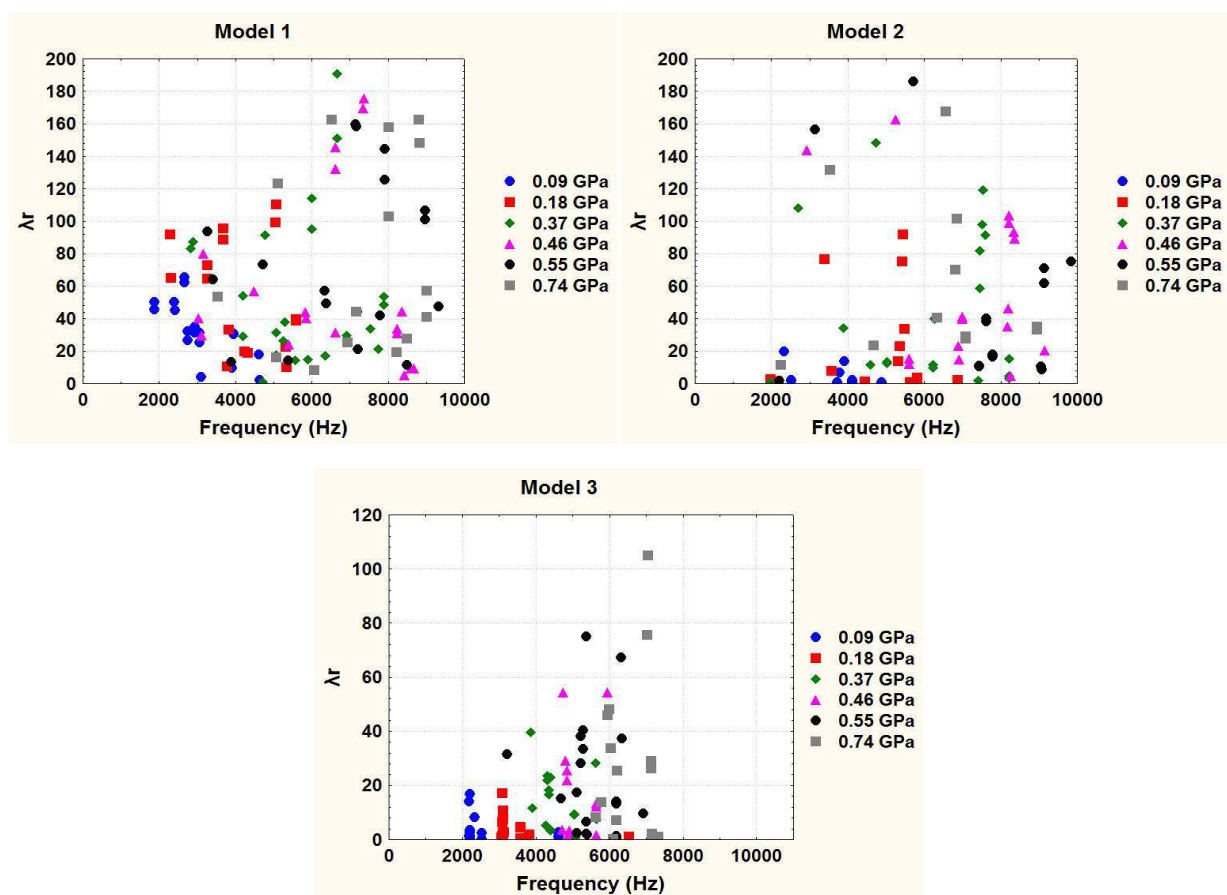
**Figure 11.** Effect of speed: (a) no. of instability points; (b) maximum positive real part.

The next variable to be changed was the Young's modulus of the friction material because, as observed in section 2, the Young's modulus could decrease for increasing temperature, due to the composite nature and

microstructure of the friction materials. The effect of the pad Young's modulus was studied multiplying its reference value ( $E_p=0.37$  GPa) by factors ranging from 0.25 to 2. Looking at figure 12, one can see that the unstable points are more scattered when compared with the results obtained for load and speed variations. However, most unstable points are still within the frequency zone of low squeal for all models, although there are some points of model 1 and 2 that are outside this frequency zone. Model 3 again shows less dispersed results in frequency with respect to the other models.

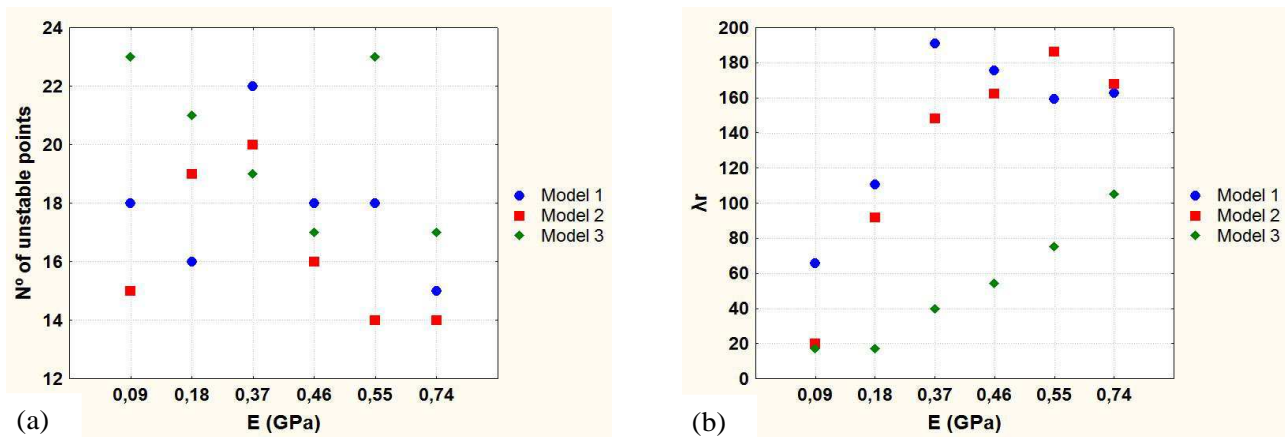
The influence of the pad Young's modulus on the number of unstable points of system (figure 13 a) is not definite. Except for one value, only model 3 shows a clear trend with a decrease of unstable points for increasing stiffness. Other authors [17,18,19] reported that increasing pad Young's modulus reduces the overall number of unstable modes and indicate that overall system stability would be enhanced by higher modulus pads. Figure 14 b shows a general increasing trend of the maximum positive real part with increasing pad Young's modulus for all models.

Finally it can be observed that in almost all the reported results the maximum positive real part of the eigenvalues of pad model 3 was always smaller than the others, independent of the simulated conditions. This behavior shows that pad model 3 has a lower tendency to instability when compared with the other two models.



**Figure 12.** Effect of pad Young's modulus on the brake system stability for each model.





**Figure 13.** Effect of pad Young's modulus: (a) no. of instability points; (b) maximum positive real part.

#### 4. Conclusions

In the present work a finite element (FE) complex eigenvalue parametric analysis was performed on the brake assembly to evaluate the propensity to dynamic instability of brakes with multiple pads as a function of the number of pads and material parameters, obtaining useful indications for the design. The following conclusions can be drawn:

- the initial simulation with all pad models showed that the system instability points, if any, are found in the frequency range of low squeal (2 - 8 kHz);
- the same initial simulations showed that in the natural mode of vibration corresponding to the maximum eigenvalue positive real part for each model, the disc doesn't have significant out-of-plane displacements while the pads have considerable out-of-plane displacements;
- as load, speed and pad Young's modulus vary, most unstable points are still found in the frequency range of low squeal (2 - 8 kHz); however, for variations of the pad Young's modulus there are some points of model 1 and 2 that are outside this frequency zone;
- increasing the load generally produced an increase of the number of unstable points and of the maximum eigenvalue positive real part;
- increasing the disc speed generally produced a reduction of the number of unstable points and of the maximum eigenvalue positive real part;
- increasing the elastic modulus of the friction material showed a general increasing trend of the maximum eigenvalue positive real part;
- in all the results the pad with the highest number of contact elements had the maximum positive real part of the eigenvalues always smaller than the others, independent of the simulated conditions and thus a lower tendency to instability.

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