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Influence of Design Parameters in the Brake Squeal in Electric Cars

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ABSTRACT

Electric cars have many advantages over vehicles that require fossil fuels for their use. However, both types of cars are subject to problems related to brake squeal, which is a noise that causes discomfort for those inside the vehicle. In particular, this problem is more significant in electric cars because the motor operates silently compared to internal combustion engines, making the noise more obvious. In this context, the aim of this study is to perform a parametric analysis to verify the influence of design parameters on squeal in a cooled disc brake of an electric car. Therefore, finite element methods were used to analyse the brake geometry, which was subjected to static and modal analyses to extract complex eigenvalues and identify unstable modes. The simulation results indicated that the friction coefficient, stiffness of the components of system, and brake pressure are significant to the performance of the brake system, being the Young's modulus of the back plate the best parameter to minimize the brake squeal.

KEYWORDS: brake noise; complex modal analysis; finite elements.

INTRODUCTION

It is a common knowledge that the transport sector is one of the main agents in the generation of environmentally harmful waste, because most vehicles still rely on fossil fuels for propulsion. In this regard, the advancement of technology has allowed the creation of electric cars, which are more economical and efficient vehicles capable of reducing the emissions of polluting gases on the planet (VERMA; DWIVEDI; VERMA, 2022). In particular, these cars run on engines that use energy stored in batteries and super capacitors to drive the machine instead of an internal combustion engine.

Although electric cars have many advantages over conventional vehicles, brake noise is a sound problem that affects both types. However, it has greater relevance in modern cars because electric motor operate silently compared to internal combustion engines, which generate more noise than the brake system (AWE, 2019).

Among all the noises caused by the brakes, the squeal stands out, which is a phenomenon that causes discomfort for those inside the vehicle, especially in the audible frequency range of 0-20 kHz according to Akay (2001). In the literature, several theories have been developed to explain this phenomenon and, therefore, it is known to be highly dependent on modal coupling, which occurs when the vibration modes are close. In addition, two other secondary mechanisms, however necessary for the squeal to really happen, are stick-slip and sprag-slip (DIAS et al., 2022).

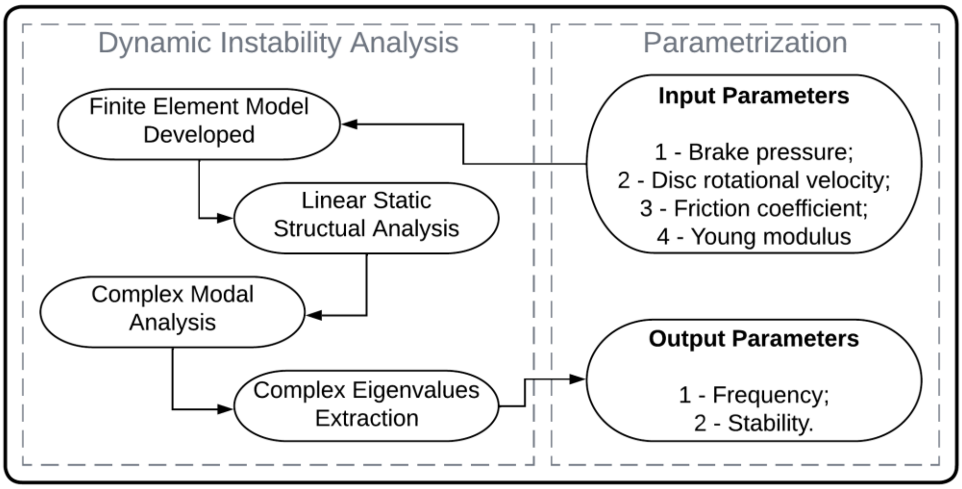
Because of the great advances in computing, developers and scientists are increasingly using finite element methods to characterize the vibrational behavior of brake systems (MIRANDA et al., 2020). For instance, Chen (2019) created a model to suppress the brake squeal in a light train disc brake system. Based on this finite element model, it was observed that the calculated unstable frequency was consistent with the experimentally measured frequency. Furthermore, Balaji (2021) developed a numerical model from the Abaqus platform to identify the main causes of brake squeal in a disc brake system. Thus, it was observed that the effects of friction and stiffness of the system components are significant.

Thus, operational aspects (velocity, brake pressure) as well as material properties (Young’s modulus, coefficient of friction) can result in instability conditions conducive to squeal and, hence, discomfort for those inside the vehicle. Therefore, the aim of this work is to perform a parametric analysis using the commercial software of finite element Ansys to verify the influence of material and operational parameters on the brake squeal in a cooled disc brake of an electric car and, thus, identify the main characteristics that the parameters must have in this type of car. To achieve this goal, a car disc brake model was developed based on the finite element method, and the geometry was subjected to both static and modal analyses for the extraction of complex eigenvalues to verify the occurrence of brake squeal. Additionally, parametric analysis of the variables was carried out through the Design of Experiments (DOE).

PARAMETRIZATION

Figure 1 illustrates the process followed in this study. The first step is the development of the finite element model of the disc brake system. In particular, six variables were considered for parameterization, and their values are described in **Table 1**. Subsequently, the following processes are carried out: 1) a static structural analysis of the brake operation, with the disc rotating, and thus obtain a state of pre-tension and 2) a frequency calculation using complex eigenvalues to obtain the unstable modes, that is, eigenvalues with a positive real part. Therefore, the output parameters are obtained, and these data are kept for further analysis.

Figure 1 – Flowchart of the sensitivity analysis model



Source: Own Authorship.

To perform the parameterization, lower and higher values of the parameters were extrapolated so that it was possible to verify the sensitivity of each one of them in the occurrence of instability points within the vibration modes of the system. It is worth emphasizing that the nominal values were used to validate the finite element model.

Table 1 – Input parameter values range of the disc brake

Parameter name (Unit)	Nominal value	Range
Brake pressure (MPa)	1.84	[1.4;2.5]
Disc rotational velocity (rad/s)	10	[4.5;44.7]
Friction coefficient	0.3	[0.3;0.5]
Back plate Young's modulus (GPa)	210	[193;220]
Disc Young's modulus (GPa)	125	[100;140]
Friction material Young's modulus (GPa)	2.60	[2.0;3.0]

Source: Own Authorship.

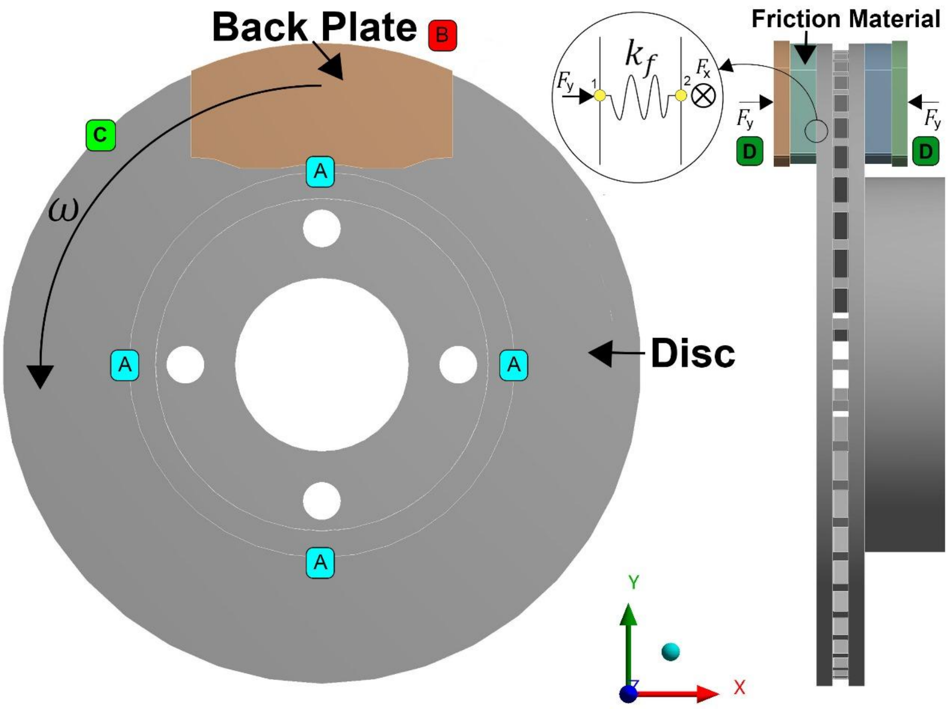
It is important to note that the Young's modulus values are set based on values presented in the bibliography. The disc, which is made of gray iron because it has acceptable thermal properties, mechanical strength and wear resistance, can be manufactured with Young's modulus ranging from below 100 GPa to approximately 140 GPa (BELHOCINE, 2015). With respect to pads, they are divided into two parts: the back steel plate, which receives the pressure to activate the system, and the front part, which is made of composite material and suffers wear due to friction. Steel does not present much variation in the modulus and usually has a range of values set to 200 and 220 GPa. However, the determination of the properties and modelling of the friction material are made using complex processes (MIRANDA et al., 2021).

FINITE ELEMENT MODEL

A car disc brake system consists of a disc coupled to the wheel, a calliper piston assembly where the first slides inside the second, and a pair of brake pads. Typically, the system is activated hydraulically when the driver presses the pedal, applying pressure to the pads. The pads are then pressed against the rotating brake disc. This contact generates a frictional force that reduces disc speed. However, the model created in this work considers only the main elements that contribute to the brake squeal, which are the disc and pads (JUNIOR; GERGES; JORDAN, 2008).

The chosen geometry was designed in Computer Aided Design (CAD) in the Design Modeller software according to the measurements of a real system available in the work of Oehlmeyer (2008) and is shown in **Figure 2**.

Figure 2 – CAD model of the disc brake system used in the simulation with the boundary condition setup for static structural analysis in Ansys

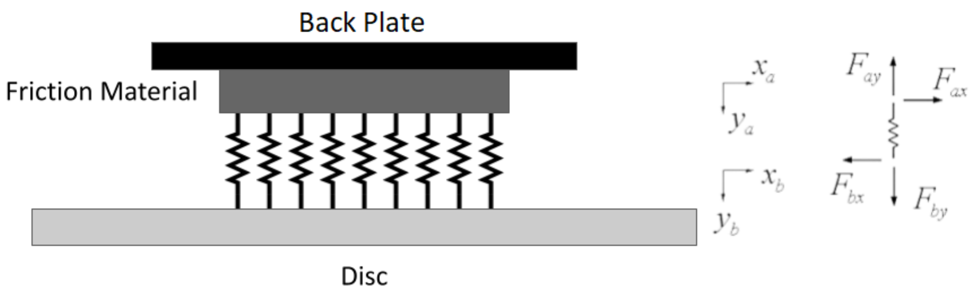


Source: Own Authorship.

Before the static analysis, the contacts between the surfaces are defined according to their behavior. Therefore, the contacts between the disc and the friction material were defined as frictional because of the sliding that occurs in these interfaces, and the contact between the friction material and the back plate was defined as bonded because they are glued together and should not move with each other (ANSYS Inc., 2017).

In simulations of brake squeal, the friction between the surfaces is usually modelled according to geometric coupling, where a spring is used to connect the corresponding pair of nodes on the disc surface and the contact pad, as shown in **Figure 3** (OUYANG, 2005). In addition, the friction coefficient is considered constant. Thus, the generation of dynamic instability must be proven through complex eigenvalue analyses, because the inconstant friction force comes from a changeable normal force (MIRANDA et al., 2021).

Figure 3 – Contact interface elements graphical representation



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The stiffness matrix of the interface element, considering normal (F_x) and frictional forces (F_y), can be expressed by **Equation 1**, where k_f represents the stiffness of the contact elements, μ is the coefficient of friction, and x and y denote horizontal and vertical displacements, respectively. To facilitate convergence, an Ansys Parametric Design Language (APDL) script was used to implement asymmetric element matrices in the system solution. It is worth noting that APDL functions as a language specific to Ansys, allowing the insertion of commands through lines of code (ANSYS Inc., 2017).

$$\begin{bmatrix} F_{ax} \\ F_{ay} \\ F_{bx} \\ F_{by} \end{bmatrix} = \begin{bmatrix} 0 & \mu k_f & 0 & \mu k_f \\ 0 & k_f & 0 & -k_f \\ 0 & \mu k_f & 0 & -\mu k_f \\ 0 & -k_f & 0 & k_f \end{bmatrix} \begin{bmatrix} x_a \\ y_a \\ x_b \\ y_b \end{bmatrix} \tag{1}$$

To initiate the static structural analysis, the following boundary conditions were defined, as illustrated in **Figure 2**: (a) the inner faces of the disc screw holes were considered as fixed supports, (b) the external faces of the back plate had their movement restricted in all degrees of freedom, except for translation in a direction normal to the contact surface, (c) disc rotation was simulated, and (d) distributed pressure was applied to the external back plate surfaces. Additionally, the rotation of disc was introduced using the CMROTATE command in an APDL script. This method is highly recommended for wear and brake squeal analysis because it induces artificial rotation in the system elements.

The static analysis consists of a single stage, with a total duration of 1.0 s, configured with the solver in interactive mode, and Large Deflections are turned on to consider changes in stiffness due to the deflection of system structures. In the Nonlinear Controls, the Newton-Raphson option was set to unsymmetric.

Additionally, the weak springs option was activated to facilitate simulation convergence. It is essential to note that this option creates an artificial set of springs linked to the model's geometry to stabilize its operation and prevent calculation errors (ANSYS Inc., 2017). In particular, the applied force is constant throughout the process of 1.0 s, which facilitates analysis.

Modal analysis, which starts in a pre-stressed state calculated during static analysis, aims to extract the complex eigenvalues and eigenvectors that indicate the unstable modes of vibration. This is possible by solving the equation of motion for an FE model of a damped system with several degrees of freedom in free vibration (**Equation 2)**, where [*M*], [*C*], [*K*] and [K_f] are the mass, damping, stiffness and contact stiffness matrices, respectively (DIAS et al., 2021).

$$[M]\ddot{x} + [C]\dot{x} + \left[K + K_f\right]x = 0 \tag{2}$$

The extraction of eigenvalues, where the matrices [*M*], [*C*] and [$K+K_f$] can be real or complex, symmetric or asymmetric, can generate complex eigenvalues defined by **Equation 3** (OEHLMEYER, 2008; MIRANDA et al., 2020; MIRANDA et al., 2021; DIAS et al., 2021; DIAS et al., 2022).

$$\lambda_{1,2} = \alpha_i \pm i\omega \tag{3}$$

In **Equation 3**, λ represents the eigenvalue, which is divided into two parts: α, the real part; and ω, the imaginary part, which represents the natural frequency of the system. If α has a negative value, the system is typically stable, however, if this value is positive, the range of motion will increase with time, which describes an unstable system (DA SILVA, 2013).

Furthermore, experimental data demonstrate that systems with a large real part (α) have a greater tendency to generate noise in the brake system, so it is a measure that indicates the probability of occurrence of squeal in a given unstable mode (TRICHES, 2004). These unstable modes occur in frequency ranges from 0 to 20 kHz (OEHLMEYER, 2008), as a result the modal analysis was configured to extract only 150 modes, because they represent frequencies up to 20 kHz.

The Ansys software has the QRDAMP method that enables the use of non-symmetric matrices and the extraction of complex eigenvalues. This functionality is incorporated into an APDL script. This method is a procedure designed to determine complex eigenvalues and the corresponding eigenvectors of a linear system, allowing non-symmetric stiffness [*K*] and damping [*C*] matrices. Therefore, **Equation 4** describes the extraction of complex eigenvalues and eigenvectors (OEHLMEYER, 2008; MIRANDA et al., 2021; DIAS et al., 2022).

$$\{\psi\} = [\phi]\{z\} \tag{4}$$

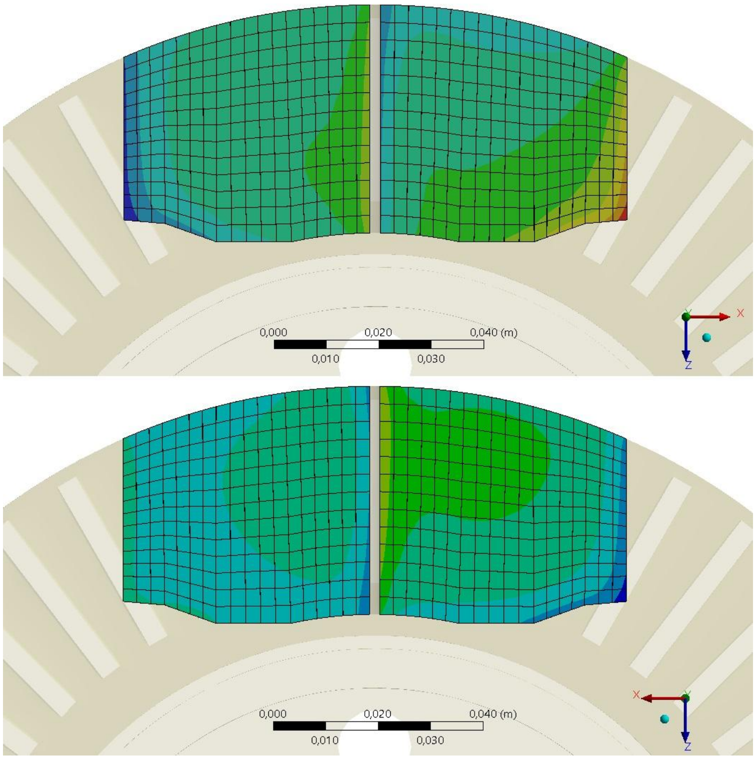
In **Equation 4**, ψ represented complex eigenvalues and eigenvectors, ϕ, the eigenvector normalized, and *z* is column matrix of 2 lines, where in the first position there is *y* and in the second *y'*.

Furthermore, an APDL script was incorporated into the modal analysis with the objective of computing the first unstable frequency, the highest eigenvalue found and the total number of unstable modes. These variables will serve as the output for the parameterization analysis.

RESULTS AND DISCUSSION

To validate the finite element model, the pressure distribution characteristics in the brake pads were examined, as illustrated in **Figure 4**. In the figure, the red color indicates points with higher pressure, followed by yellow, light green, dark green, cyan, light blue and dark blue, in descending order of pressure. Thus, the maximum pressure was observed at the leading edge of the contact, where the frictional contact begins. However, the minimum pressure occurs at the opposite end of the contact area, demonstrating a behavior consistent with that expected for a disc brake pad. Additionally, the distribution profile between the two pads is not symmetrical due to the asymmetrical mass distribution on the disc and, therefore, there is a higher contact pressure on the side with greater mass.

Figure 4 – Contact pressure results of the automotive brake static structural

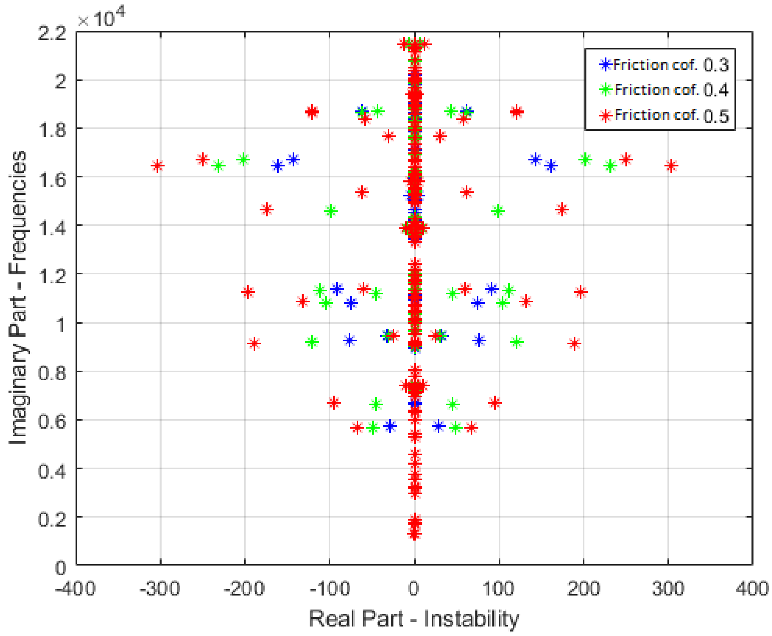


Source: Own Authorship.

Another method used for validation involved plotting the real part of the complex eigenvalues against their corresponding natural frequencies associated (**Figure 5**). In the graph, the x-axis is the real part and the y-axis is the imaginary part of the eigenvalues, which is related to frequency. This visualization allows for a clear understanding of the overall occurrence of instability, making the assessment more accessible.

The main parameter that tends to increase the occurrence of unstable modes is the friction coefficient (OUYANG, 2005), as demonstrated in **Figure 5**. In the graph, blue stars represent the results for a friction coefficient of 0.3, green stars for a coefficient of 0.4, and blue stars again for a coefficient of 0.5. These friction values resulted in 10, 14 and 22 unstable frequencies, respectively. Thereby, there is an unstable modal density, that leads to brake squeal, and the increase in thecoefficient, this density tends to intensify.

Figure 5 – Real part of complex eigenvalue as a function of the natural frequency of the associated model



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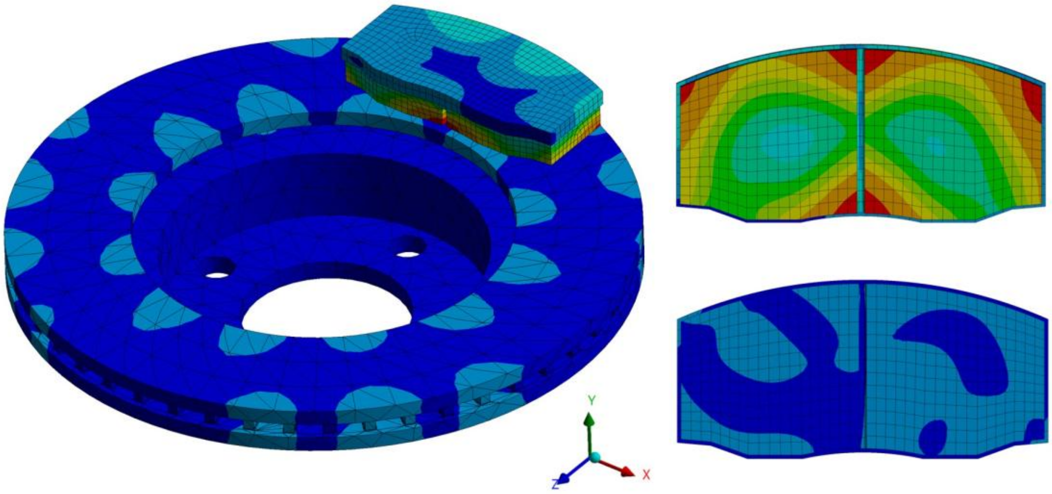
Furthermore, **Figure 6** illustrates the deformation of the unstable mode with the maximum real part. This mode exhibits consistent deformation profiles for all friction values, maintaining similar patterns across different coefficients. However, these variations in the friction coefficient not only lead to higher values of the real part but also increase the total deformation. In fact, the maximum deformation reached 295.96, 315.6, and 330.68 mm, respectively.

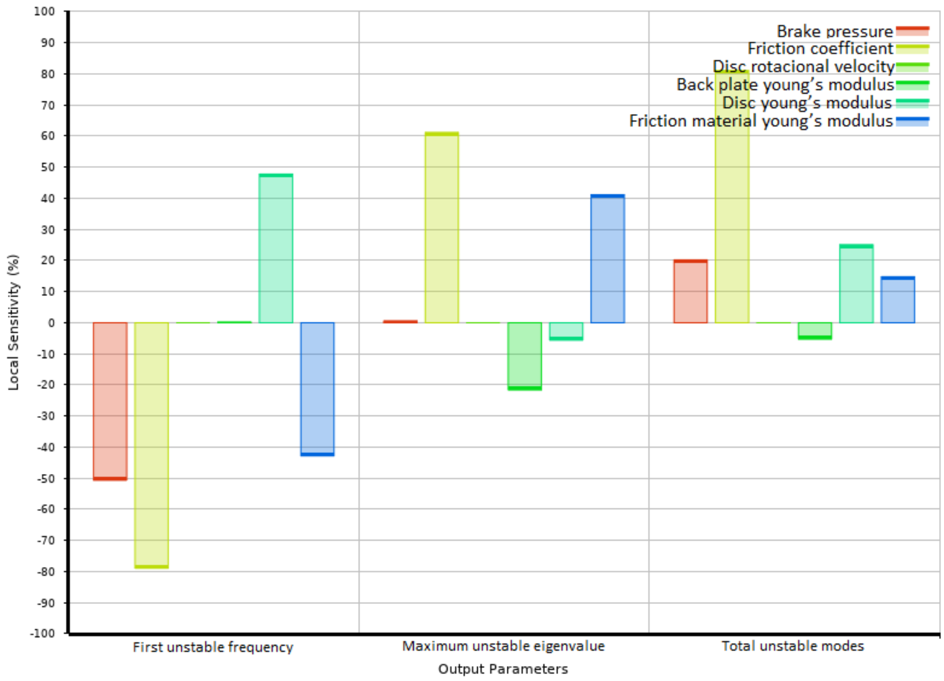
Figure 6 – Representation of unstable mode with the maximum real part of the automotive brake system



Source: Own Authorship.

Figure 7 summarizes the results obtained from 80 simulations of the parameterization process and it indicates the degree of influence that each parameter has on the output variable. Thus, if it is positive, then the input parameter has a direct correlation with the output parameter, i.e., there is an increase in the response if this input value increases. Analogously, the negative value indicates an inverse correlation with the output, i.e., if the input parameter grows, the output will have a smaller response.

Figure 7 – Disc brake sensitivity bar chart



Source: Own Authorship.

Based on **Figure 7**, an increase in the Young's modulus of the disc tends to raise the first unstable frequency and decrease the tendency to occur unstable modes, although it increases the total number of modes. In addition, it is observed that a high value of the back plate Young's modulus stabilizes the brake system because an increase in this value reduces the total number of unstable modes and their likelihood of occurrence. On the other hand, the Young's modulus of the friction material needs to be low because it elevates the first unstable frequency and reduce the total number of unstable modes as well as the chance of their occurrence. These results are consistent with the findings of Miranda et al. (2020).

It is widely known that a higher coefficient of friction signifies better braking performance; however, it is evident that this also signifies poorer noise performance, as depicted in **Figure 7**. In contrast, the rotational velocity of the disc exhibits little or no influence on brake squeal. Furthermore, brake pressure, a parameter under the driver's control, significantly impacts the occurrence of brake squeal. In particular, it is observed that increases in braking pressure lead to an increase in the first unstable frequency. Hence, a high braking pressure causes the frequency where the noise occurs to be higher, however this generate more unstable modes. This behavior is also visualized in the work of Junior, Gerges and Jordan (2008), who created a model to analyse the instability of disc brake systems in cars.

In summary, minimizing brake squeal can involve decreasing the friction coefficient, although this action may compromise brake performance, affecting the reliability and safety of the system. Another measure often cited is ensuring well-spaced natural frequencies of components to prevent modal coupling (OEHLMEYER, 2008). Thereby, design modifications can be applied to the system or its materials, however implementing geometric chances is challenging due to design constraints. Consequently, replacing materials proves to be a more effective approach. For instance, Dunlap, Riehle and Longhouse (1999) replaced gray cast iron (material for the disc) with cast iron possessing greater damping, successfully eliminating noise and altering the natural frequency of the disc. Based on the results, it is advisable to increase the Young's modulus of the disc and back plate while decreasing the Young's modulus of the friction material. Furthermore, drivers are recommended to apply low brake pressure to minimize the presence of additional modes and prevent squeal.

CONCLUSION

Brake squeal is a problem that affects any type of car, including electric cars. Therefore more resources should be available for researchers with the aim of minimizing the instability of the brake system and, hence, noise discomfort. Based on that, in this work, we sought to identify the main characteristics that the material and operational parameters must have in a refrigerated disc brake system of an electric car.

Through simulation with finite elements, it was possible to parameterize and verify the influence of material and operational parameters on the brake squeal, obtaining the following results:

1) High coefficients of friction increase the degree of system instability.
2) Regarding the material, electric cars must have a high Young's modulus for the disc and back plate. However, friction material should have a low Young's modulus.
3) The back plate has the function of damping the vibration of the disc and pads.
4) For a high braking pressure, the first frequency where the noise occurs tends to be higher and the total number of modes is increased.
5) Disc rotation velocity has little or no impact on brake squeal generation.

For future work, we intend to test other automotive system geometries, such as solid discs and explore changes in the contact area of the pad. In this way, it will be possible to compare different geometries to identify the best values for the parameters of the brake systems and, thus, minimizing noise discomfort in electric cars. Furthermore, the strategy also involves incorporating wear due to friction into the model to analyse its influence on squeal.

Influência de Parâmetros de Projeto no *Brake Squeal* de Carros Elétricos

RESUMO

O carro elétrico tem muitas vantagens em relação aos veículos que requerem combustíveis fósseis para seu uso. Porém, ambos os tipos de carro estão sujeitos a problemas relacionados ao *brake squeal*, o qual é um ruído que causa desconforto sonoro para todos dentro do veículo. Em particular, isso é um problema mais significante nos carros elétricos porque o motor funciona de forma silenciosa quando comparada aos motores de combustão interna e, assim, esse ruído é mais aparente. Nesse contexto, o objetivo deste trabalho é realizar uma análise paramétrica para verificar a influência dos parâmetros de projeto na geração de *squeal* em um disco de freio refrigerado. Com isso, métodos dos elementos finitos foi utilizado em uma geometria de freio, o qual foi sujeita a uma análise estática e modal para extrair os autovalores complexos e, assim, identificar os modos instáveis. Os resultados da simulação indicaram que o coeficiente de atrito, as rigidez dos componentes do sistema e a pressão nas pastilhas são significantes para a performance do sistema de freio, sendo o módulo de Young da placa traseira da pastilha o melhor parâmetro para minimizar o *brake squeal*.

PALAVRAS-CHAVE: ruído de freio; análise modal complexa; elementos finitos.

Influencia de los Parámetros de diseño en el Brake Squeal de los coches elétricos

RESUMEN

El coche eléctrico tiene muchas ventajas frente a los vehículos que requieren de combustibles fósiles para su uso. Sin embargo, ambos tipos de automóviles están sujetos a problemas relacionados con el *brake squeal*, que es un ruido que causa molestias sonoras a todos los que están dentro del vehículo. En particular, este es un problema más importante con los autos eléctricos porque el motor funciona de manera más silenciosa en comparación con los motores de combustión interna y, por lo tanto, este ruido es más evidente. En este contexto, el objetivo de este trabajo es realizar un análisis paramétrico para verificar la influencia de los parámetros de diseño en la generación de *squeal* en un disco de freno refrigerado. Así, se utilizaron métodos de elementos finitos en una geometría de freno, la cual fue sometida a un análisis estático y modal para extraer los autovalores complejos y así identificar los modos inestables. Los resultados de la simulación indicaron que el coeficiente de fricción, la rigidez de los componentes del sistema y la presión sobre las pastillas son significativos para el rendimiento del sistema de frenos, siendo el módulo de Young de la placa trasera de la pastilla el mejor parámetro para minimizar el *brake squeal*.

PALABRAS CLAVE: ruido de freno; análisis modal complejo; elementos finitos.

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