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PARAMETRIC STUDIES OF DISC BRAKE SQUEAL USING FINITE ELEMENT APPROACH

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ABSTRACT

Disc brake squeal noise is a very complicated phenomenon, which automobile manufacturers have confronted for decades due to consistent customer complaints and high warranty costs. In recent years, the finite element method (FEM) has become the preferred method due to high hardware costs of experimental methods. In this study, a simplified model for the disc brake is presented using the ABAQUS/Standard finite element software. The analysis process uses a nonlinear static simulation sequence followed by a complex eigenvalue extraction to determine the squeal propensity. The effect of the main operational parameters (braking pressure, and friction coefficient) on the squeal propensity is performed. The influence of changing the rotor stiffness and back plates stiffness under different operation condition are investigated. The results of this analysis show that the squeal noise can be reduced by increasing the rotor stiffness and decreasing the back plate stiffness of the pads.

Keywords: *Disc brake, squeal, finite element, complex eigenvalue, parametric study*

1.0 INTRODUCTION

Disc brake noise, in general, is one of the major contributors to the automotive industry’s warranty costs. In most cases, this type of noise has little or no effect on the performance of brake system. However, most customers perceive this noise as a problem and demand that their dealer’s fix it. Customer complaints result in significant yearly warranty costs. More importantly, customer dissatisfaction may result in the rejection of certain brands of brake systems or vehicles. The automotive industry is thus looking for new ways to solve this problem [1].

In general, brake noise has been divided into three categories, in relation to the frequency of noise occurrence. The three categories presented are low-frequency noise, low-frequency squeal and high-frequency squeal. Low-frequency disc brake noise typically occurs in the frequency range between 100 and 1000 Hz. Typical noises that reside in this category are grunt, groan, grind and moan. This type of noise is caused by friction material excitation at the rotor and lining

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interface. The energy is transmitted as a vibratory response through the brake corner and couples with other chassis components [2].

Low-frequency squeal is generally classified as a noise having a narrow frequency bandwidth in the frequency range above 1000 Hz, but below the first in-plane mode of the rotor. The failure mode for this category of squeal can be associated with frictional excitation coupled with a phenomenon referred to as ''mode locking'' of brake corner components. Mode locking is the coupling of two or more modes of various structures producing optimum conditions for brake squeal [2].

High-frequency brake squeal is defined as a noise which is produced by friction induced excitation imparted by coupled resonances (closed spaced modes) of the rotor itself as well as other brake components. It is typically classified as squeal noise occurring at frequencies above 5 kHz. Since it is a range of frequency which affects a region of high sensitivity in the human ear, high-frequency brake squeal is considered the most annoying type of noise. Brake squeal is a concern in the automotive industry that has challenged many researchers and engineers for years. Considerable analytical, numerical and experimental efforts have been spent on this subject, and much physical insight has been gained on how disc brakes may generate squeal, although all the mechanisms have not been completely understood [3].

This study attempts to present a simplified finite element model to examine the squeal propensity of a disc brake system for a range of operational parameters like friction coefficient, and braking pressure. The evaluation of the effect of material properties (the rotor Young's modulus and the back plates of the pads young's modulus) on the squeal propensity is performed. The simulations performed in this study present a guideline to reduce squeal noise by using design modification, which dependent on the modified material prosperities of disc brake components.

2.0 BRAKE NOISE GENERATION MECHANISMS

Disc brake squeal occurs when a system experiences vibrations with very large mechanical amplitude. It is supposed that there are two occurrence mechanisms of a squeal noise. The first mechanism is a phenomenon resulting from the "stick-slip" of a friction side [4]. The second mechanism is a phenomenon resulting from geometric instabilities of the brake assembly [3]. Both mechanisms, however, attribute the brake system vibration and the accompanying audible noise to variable friction forces at the pad–rotor interface. Regarding the squeal noise caused by geometric instability of system, if two neighboring vibration modes are close to each other in the frequency range and have similar characteristics, they may merge if the coefficient of friction between the pad and disc increases. When these modes coupled at the same frequency, one of them becomes unstable. The unstable mode can be identified during complex eigenvalue analysis [5-12] because the real part of the eigenvalue corresponding to an unstable mode is positive.

3.0 METHODOLOGY AND NUMERICAL MODEL

3.1 Problem Formulation

The mass matrix and stiffness matrix of engineering structures can be assumed to be symmetric, respectively, positive definite and semi-positive definite in general. The eigensolutions of such structures are extensively studied and the vibration of such systems is stable. There are, however, engineering problems whose stiffness matrices are asymmetric. Usually the asymmetry is produced not by the structure itself, but by some external loads interacting with the structure [15], such as friction in brake noise problems [13].The equation of motion for a vibrating system is

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\{F\} \tag{1}$$

where M, C and K are mass, damping and stiffness matrices, respectively, and u is the generalized displacement vector.

For friction induced vibration, it is assumed that the forcing function F is mainly contributed to by the variable friction force at the pad-rotor interface. The friction interface is modeled as an array of friction springs. With this simplified interface model, the force vector becomes linear:

$$\{F\}=\lfloor K_f \rfloor\{u\} \tag{2}$$

where, K_f is the friction stiffness matrix.

A homogeneous equation is then obtained by combining Eqs. (1), and (2) by moving the friction term to the left-hand side

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K-K_f]\{u\}=\{F\} \tag{3}$$

Eq. (3) is now the equation of motion for a free vibration system with a pseudo forcing function in the stiffness term. The friction stiffness acts as the so-called ‘‘direct current’’ spring [1] that causes the stiffness matrix to be asymmetric.

3.2 Complex Eigenvalue Analysis

The complex eigenvalue analysis made available in ABAQUS is utilized to determine disc brake assembly stability. The essence of this method lies in the asymmetric stiffness matrix that is derived from the contact stiffness and the friction coefficient at the disc/ pads interface [6]. In order to perform the complex eigenvalue analysis using ABAQUS, four main steps are required [7].
They are given as follows:

i. Nonlinear static analysis for applying brake-line pressure.
ii. Nonlinear static analysis to impose rotational speed on the disc.
iii. Normal mode analysis to extract natural frequency of undamped system.
iv. Complex eigenvalue analysis that incorporates the effect of friction coupling.

In this analysis, the complex eigenproblem is solved using the subspace projection method; thus, the natural frequency extraction analysis must be performed first to determine the projection subspace. The complex eigenvalue problem can be given in the following form:

$$(\lambda^2 M + \lambda C + K)y = 0 \tag{4}$$

Where M is the mass matrix, which is symmetric and positive definite; C is the damping matrix, which can include friction-induced damping effects as well as material damping contribution; K is the asymmetric (due to friction contributions) stiffness matrix; λ is the eigenvalue; and y is the eigenvector. Both eigenvalues and eigenvectors may be complex. In the third step stated above, this system is symmetrized by dropping the damping matrix C and asymmetric contributions to the stiffness matrix K_s to find the projection subspace. Therefore, the eigenvalue, λ becomes a pure imaginary where $\lambda = i\omega$ and the eigenproblem can be written as follows:

$$(-\omega^2 M + K_s)Z = 0 \tag{5}$$

This symmetric eigenvalue problem is solved using the Lanczos iteration eigensolver. Next, the original matrices are projected onto the subspace of real eigenvectors. z and given as follows:

$$\begin{aligned} M^* &= [z_1, z_2, \dots\dots\dots z_n]^T M[z_1, z_2, \dots\dots\dots z_n], \\ C^* &= [z_1, z_2, \dots\dots\dots z_n]^T C[z_1, z_2, \dots\dots\dots z_n], \\ K^* &= [z_1, z_2, \dots\dots\dots z_n]^T K[z_1, z_2, \dots\dots\dots z_n], \end{aligned} \tag{6}$$

Now the eigenvalue problem is expressed in the following form:

$$(\lambda^2 M^* + \lambda C^* + K^*)y^* = 0 \tag{7}$$

The reduced complex eigenvalues problem is then solved using the QZ method for a generalized nonsymmetrical eigenvalue problem. The eigenvectors of the original system are recovered by the following:

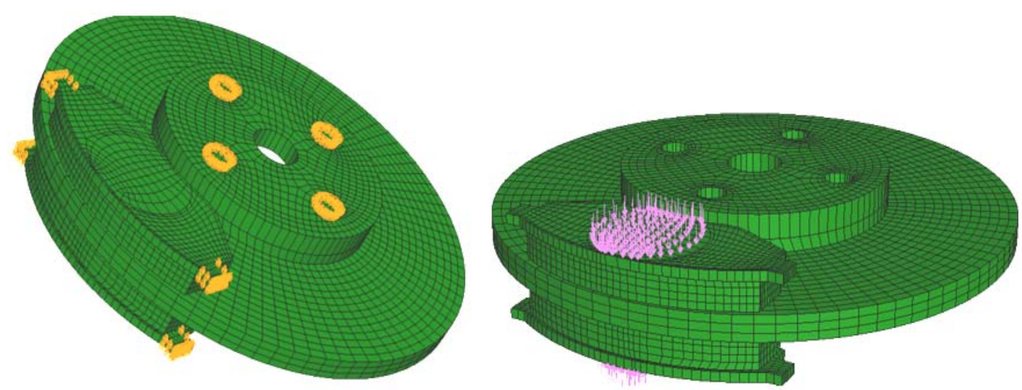
$$Y^k = [z_1, z_2, \dots\dots\dots z_n]y^{*k} \tag{8}$$

where y^k is the approximation of the k-th eigenvector of the original system.

For more detailed description of the formulation and the algorithm, refer to [16]. The complex values λ , can be expressed as $\lambda = \alpha \pm i\omega$ where α is the damping coefficient (real part of λ) and ω is the damped natural frequency (imaginary part of λ) describing damped sinusoidal motion. If the damping coefficient is negative, decaying oscillations typical of a stable system result. A positive damping coefficient, however, causes the amplitude of oscillations to increase with time. Therefore the system is not stable when the damping coefficient is positive. By examining the real part of the system eigenvalues the modes that are unstable and likely to produce squeal are revealed. An extra term, damping ratio, is defined as $-2\alpha/|\omega|$. If the damping ratio is negative, the system becomes unstable, and vice versa.

3.3 Finite Element Analysis Approach

A commercial front disc brake system consists of a rotor that rotates about the axis of a wheel, a caliper–piston assembly where the piston slides inside the caliper, which is mounted to the vehicle suspension system, and a pair of brake pads. When hydraulic pressure is applied, the piston is pushed forward to press the inner pad against the disc and simultaneously the outer pad is pressed by the caliper against the disc. Numerical simulations using the ABAQUS finite element software package were performed in this study for a simplified version of a disc brake system which consists of the two main components contributing to squeal: the disc and the pads as shown in Figure 1. The disc has a diameter of 280 mm and a thickness with typical value of 10 mm and is made of cast iron. The pair of brake pads, which consist of contact plates and back plates, are pressed against the disc in order to generate a friction torque to slow the disc rotation. The contact plates are made of an organic friction material and the back plates are made of steel.



(a) (b)

Figure 1: The simplified disc brake model

The FE mesh is generated using three-dimensional hexahedral element (C3D8 and C3D8I) for the disc and pads. There are about 21,700 nodes and 16,200 elements are used. The surface based contact interactions are defined between both sides of the disc as master surface and the contact plates of the pads as slave surface without the need for matching meshes.

Figure 1a, present the boundary conditions used for the model are as follows, the ears of the back plate were constrained in all degree of freedom (DOF) except the friction surface normal direction, and the rotor was constrained in all DOF at the bolt holes. The calliper–piston assembly is not defined in the simplified model of the disc brake system, hence the hydraulic pressure is directly applied to the back plates at the contact regions between the pads and the pistons as shown in Figure 1b, and it is assumed that an equal magnitude of force acts on each pad.

In the first step of the brake squeal analysis the contact between the pads and the rotor is established by applying pressure of a cross the back plate and a non-linear static analysis was performed that included both a preload step and a rotational velocity on the rotor step following the same methodology. Next, the lanczos method extracted the real eigenvalues and mode shapes of the model. Finally the complex modes analysis is performed on brake system model based on the real frequency calculated by Lanczos method.

4.0 ANALYSIS OF STABILITY FOR DISC BRAKE

4.1 Description of Unstable Modes of Disc Brake

To demonstrate the squeal propensity of the disc brake, the 100 eigenvalues extracted between zero and 13 kHz for the base brake system with $\mu = 0.5$ are plotted on the complex plane in Figure 2. In the baseline case no other sources of damping are specified. All of the modes have zero damping (lie on the imaginary axis) except where pairs of modes have become coupled and formed a stable/unstable pair. These result in the eigenvalue that occur in conjugate pairs that are symmetrically located about the imaginary axis. In this case nine unstable modes can be seen. An alternative way to express these results is to plot damping ratio vs. frequency as shown in Figure 3. The nine modes with positive real parts now appear with negative damping values. While there is no direct proportionality between squeal propensity and the level of damping coefficient, it has been suggested that higher values tend to be associated with modes that are most likely to squeal [6].

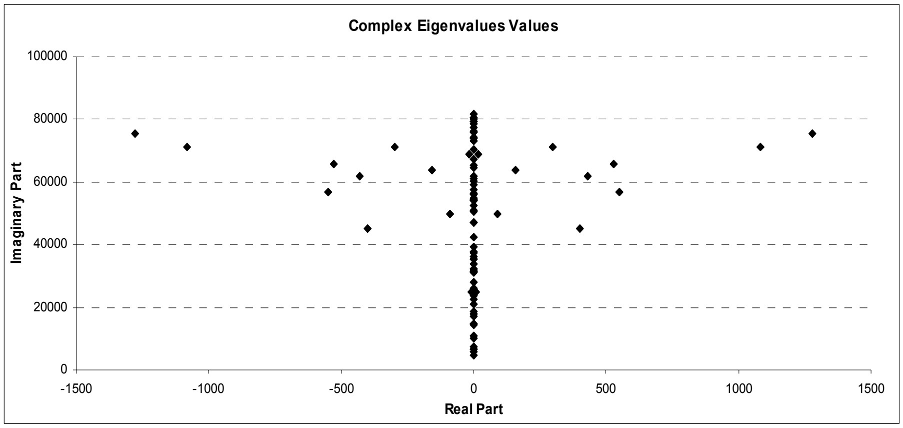


Figure 2: Eigenvalues extracted from the disc brake model plotted on the complex plane.

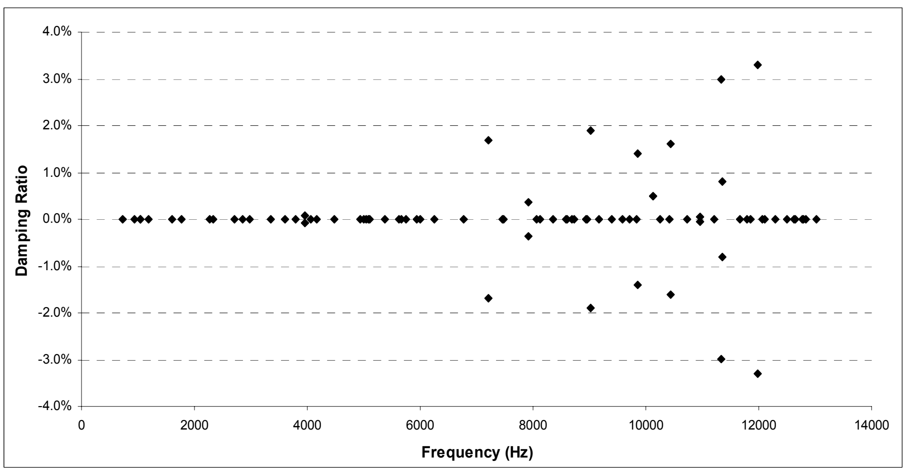


Figure 3: Damping ratio vs. frequency for the disc brake model

The results show that higher damping coefficient is approximately at unstable frequency 12 kHz. There is a significant pad bending vibration for these cases. Figure 4, gives an example of the vibration mode of the disc brake system at a frequency of 12 kHz.

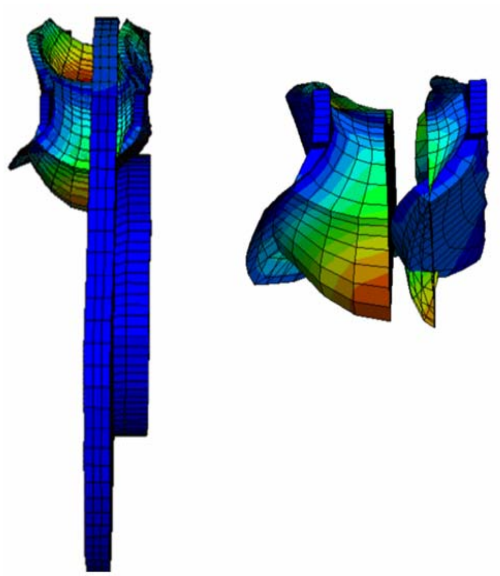


Figure 4: Vibration mode of the disc brake model at 12 kHz.

It can be seen that the pads have serious out-of-plane modes as shown in Figure 4 this suggests that the brake pads may be the source of the disc brake squeal. Except the unstable vibration modes which occur at frequency 12 kHz and are caused mainly by the pads vibration, the other unstable vibration modes are caused mainly by the disc vibration. Figure 5 give an example of the unstable vibration mode of the disc brake system, where all the system parameters are the typical values. It can be seen that the disc has significant out-of-plane vibration compared with the vibration of pads.

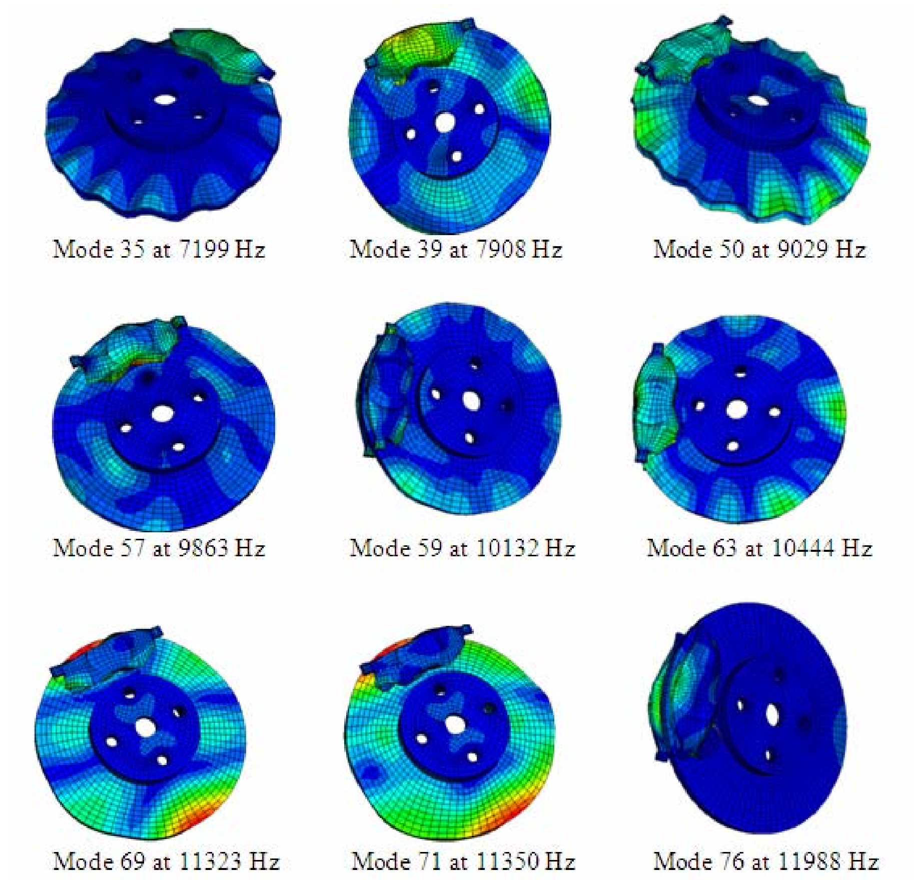


Figure 5: Example of unstable modes of the disc brake model.

5.0 EFFECT OF PARAMETERS FOR DISC BRAKE SQUEAL

5.1 Variation of Friction Coefficient

The effect of friction coefficient of the pad-rotor interface is performed. Usually, the analysis is performed for varying the friction coefficients from 0.1 to 0.7. With the low friction coefficient all of the modes of the system will be stable. As the friction coefficient is increased, modes can be driven closer to one another in frequency. At some critical friction value, a sudden change occurs (called a bifurcation), and a new mode exists that contains the original modes as a coupled pair. Figure 6a, shows results in the form of the damping coefficient as a function of frequency for different friction coefficients. It can be seen that the major squeal frequency is approximately 12 kHz. The value of the damping coefficient is increase significantly with an increase of the friction coefficient as shown in Figure 6b, at a frequency of 12 kHz.

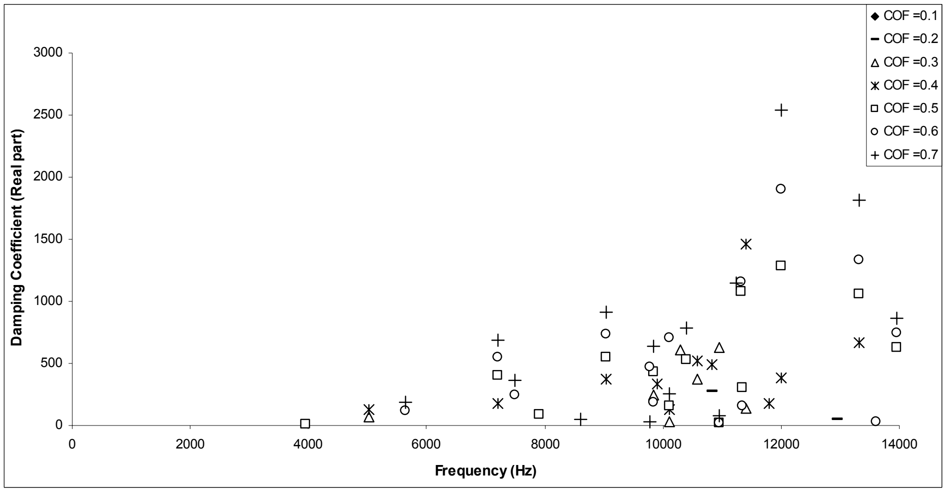


Figure 6a: Unstable modes with friction coefficient varied from 0.1 to 0.7.

It is understandable that with an increase in the friction coefficient, there is an accompanying increase in the instability of the system, thus an increase in the damping coefficient. This means that the most fundamental method of eliminating brake squeal is to reduce the friction between the pads and the disc. However, this obviously reduces braking performance and is not a preferable method to employ.

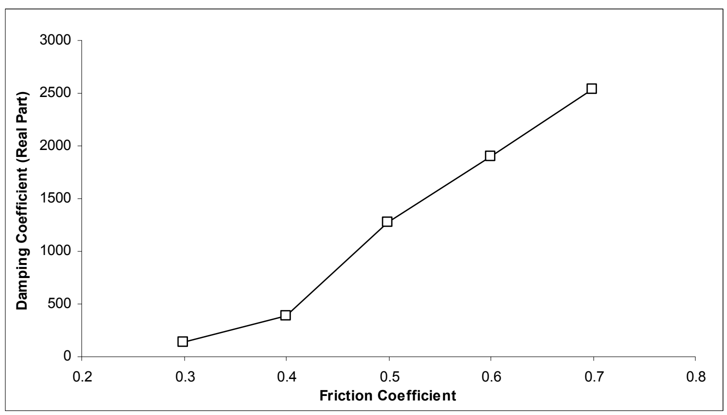


Figure 6b: Variation of the damping coefficient with friction coefficient at frequency 12 kHz.

5.2 Variation of Braking Pressure

The effect of the braking pressure on the squeal propensity is studied by varying the applied pressure from 0.1 MPa to 1.5 MPa. An initial comparison of the eigenvalue values were performed for braking pressure. Figure 7a, shows the change of the damping coefficient with frequency for different braking pressure.

The main unstable frequency at 12 kHz was chosen for a deeper analysis, which has a significant effect in terms of defining the squeal propensity. Figure 7b, shows the variation of the damping coefficient with braking pressure at frequency 12 kHz. Basically, the increase in braking pressure leads to a linear increase in the main unstable frequency. Thus the squeal propensity is increased, due to a large braking pressure leading to high values for contact stiffness between the pads and the rotor [12].

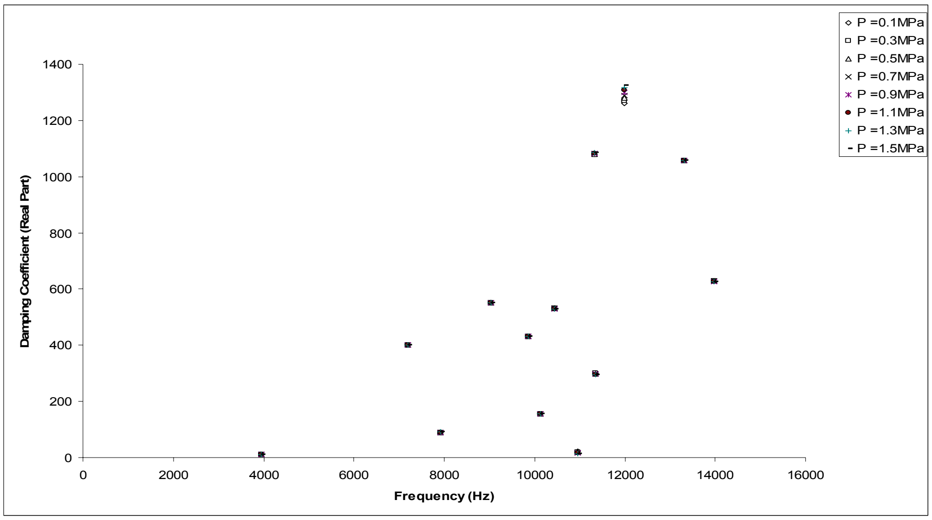


Figure 7a: Unstable modes with braking pressure varied from 0.1 to 1.5 MPa.

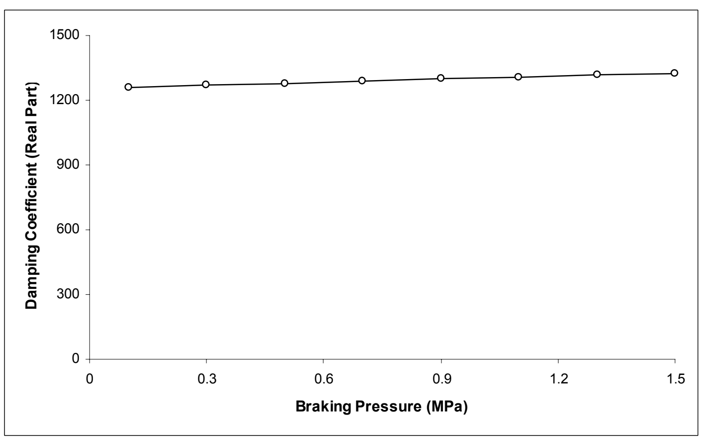


Figure 7b: Variation of the damping coefficient with braking pressure at frequency 12 kHz.

5.3 Variation of Stiffness of the Disc

The effect of rotor stiffness in terms of Young's Modulus is performed. The rotor is made of grey cast iron. The elastic modulus of cast irons varies from below 100 GPa through to the values close to that of steel at approximately 200 GPa. Grey cast iron is particularly variable in properties depending upon its carbon and, to a lesser degree, silicon content [14]. The stiffness of the disc brake is performed by varying Young's modulus of the disc from 85 GPa to 135 GPa. Where the baseline young's modulus of the disc is 105 GPa. Figure 8a, Shows results of the damping ratio versus frequency for different Young's modulus 85 GPa, 95 GPa, 105 GPa, 115 GPa, 125 GPa and135 GPa.

It can be seen that the major squeal frequency does not change for different disc Young's modulus. The value of the major squeal frequency is approximately 12 kHz. As Young's modulus is increased and hence as the stiffness of the disc is increased, the value of the damping coefficient decreases. Similar evaluations have been carried out by Liu et al [11]. Figure 8b, presents the damping coefficient versus Young's modulus of the disc at a frequency of 12 kHz.

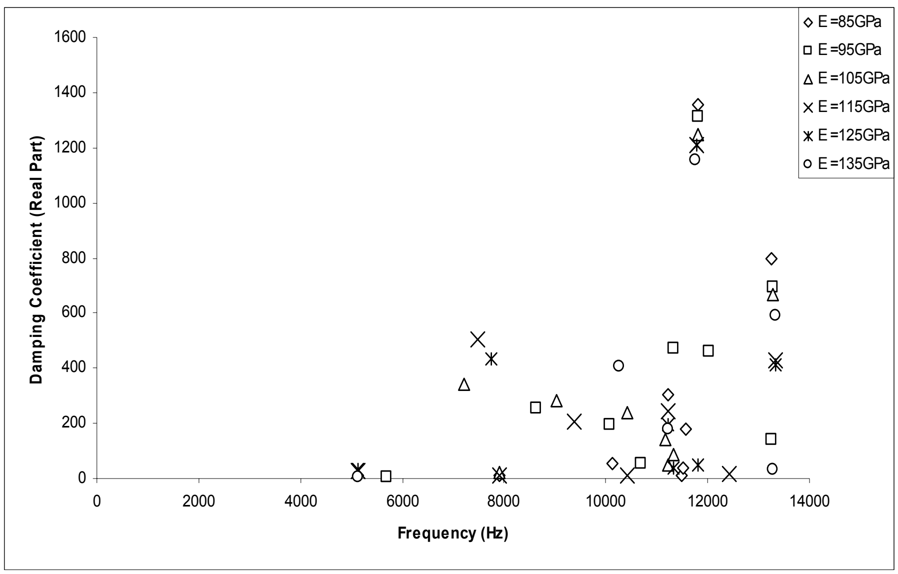


Figure 8a: Variation of the damping coefficient with frequency for different Young's modulus of the disc.

It is found that larger disc stiffness can reduce the squeal propensity of the disc system. This can be looked upon as increasing the mechanical impedance of the rotor and therefore making it more resistive in responding to input forces and reduce the vibration magnitude; as a result, the squeal propensity of the disc system can be reduced.

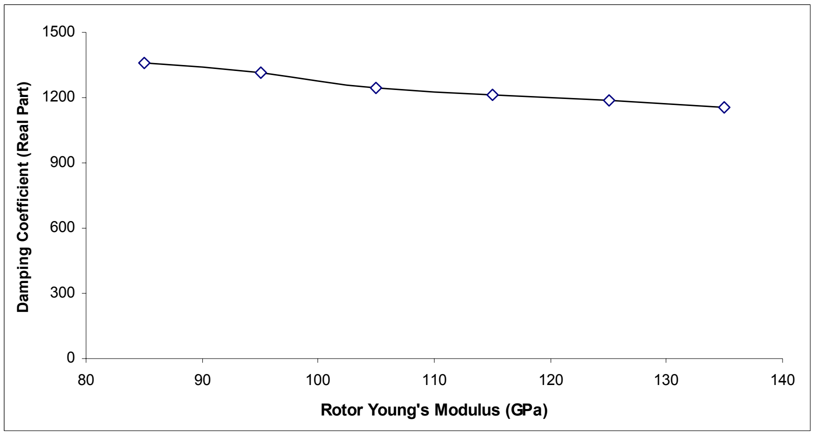


Figure 8b: Variation of the damping coefficient with young's modulus of the disc at frequency 12 kHz

5.4 Variation of Stiffness of the Back Plates of the Pad

The disc brake pads consist of two parts, friction plates which are made of organic material and back plates made of steel. In this study, the baseline Young's modulus of the back plates of the pads is 210 GPa was varied from 190 GPa to 230 GPa. To perform the variation of back plate stiffness on the disc squeal. Figure 9a, shows results of the damping coefficient versus frequency for different Young's modulus.

It can be seen that the dominant squeal occurs at a frequency of approximately 12 kHz. As Young's modulus is increased, corresponding to an increase in stiffness of the back plates of the pads, the value of the damping coefficient increases significantly as shown in Figure 9b, the variation of the damping coefficient with Young's modulus of the back plates at the main frequency is shown. This important observation implies that the stiffer back plates of pads cause a higher squeal propensity. This is so since the friction material connected to the back plates is very soft compared with the back plate material. Hence the higher the stiffness of the back plates, the greater the uneven deformation and vibration magnitude of the pad, and hence the higher the damping coefficient. Thus, an effective method to reduce squeal propensity of disc brake system is to use a soft material for the back plates of the pads.

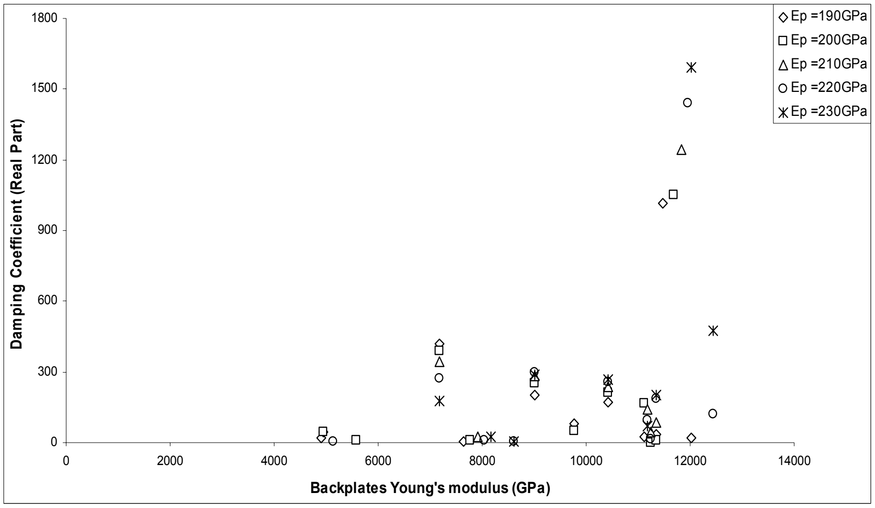


Figure 9a: Variation of the damping coefficient with frequency for different young's modulus of the back plates of the pads.

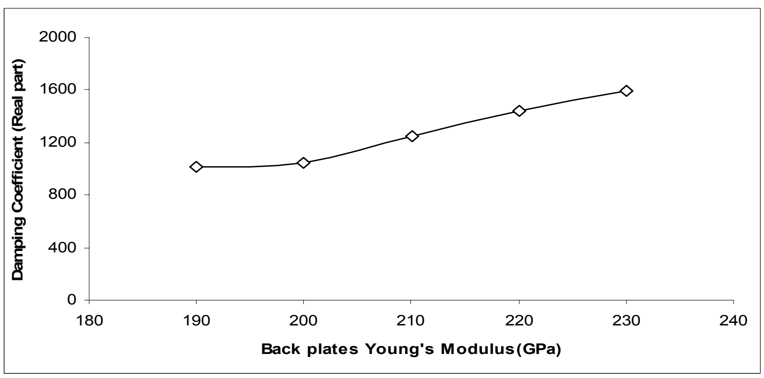


Figure 9b: Variation of the damping coefficient with young's modulus of the back plates of the pads at frequency 12 kHz.

6.0 CONCLUSION

Friction-induced disc brake squeal is investigated using the ABAQUS/Standard finite element software, which combines a nonlinear static analysis and a complex eigenvalue extraction method. The nonlinear effects can be taken into account in the preloading steps in order to more accurately friction-induced damping taken into account at which a complex eigenvalue analysis is performed. The parametric analysis shows that significant pad bending vibration may be responsible for causing the disc brake squeal and the major squeal frequency is approximately 12 kHz for the present disc brake system. The effects of the friction between the pads and the disc, the stiffness of the disc, and the stiffness of the back plates of the pads on disc squeal are significant, but the effects of the hydraulic pressure on disc squeal are not obvious. Parametric study shows that, if the Young's modulus of the disc is larger, the system is more stable, and, if the Young's modulus of the back plate of the pads is larger, the system is more unstable.

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