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A Numerical Investigation into the Brake Squeal Propensity Using Finite Element Method

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Abstract Brake squeal noise which is caused by friction-induced vibrations has been considered as one of the most difficult problems in the automobile industry. In recent years, finite element (FE) simulations are widely used for simulating disc brake squeal and the main objective of this research is to provide new insights into the squeal generation the importance of the friction coefficient using the FE model. A detailed 3D-FE model of a commercial disc brake assembly is created in order to predict squeal behavior. The FE model is validated using the results of the components and assembly's experimental measurements, followed by using a complex eigenvalue analysis to assess the braking stability. It is found that the validated FE model by experimental modal testing of the brake components and assembly can predict the disc brake squeal with satisfactory accuracy. It is also found that the probability of squeal noise Propensity increases with increasing the coefficient of friction between the disc and the brake pads.

Keywords: *disc brake squeal, finite element analysis, experimental modal analysis, coefficient of friction*

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1. Introduction

The squealing phenomenon is presented in the form of an absolute percentage of noise occurrences relative to the sound pressure level, from 70 to 94 dB [1]. Brake squeal has been studied by experimental, analytical and numerical methods in an attempt to understand, predict and prevent squeal occurrence. A recent review stated that experimental methods are expensive due to hardware costs and long turnaround time for design iterations. In addition, discoveries made on a particular type of brake are not always transferable to other types of brake and quite often product developments are based on a trial-and-error basis. Furthermore, a stability margin is frequently not found experimentally [2].

Recent developments in computer-aided engineering (CAE), especially the finite element analysis (FEA) method, have resulted in an increased capability to design brakes that perform better from a noise point of view. A 3-dimensional finite element model of brake pads is developed in [3]. Ripin [4] only considered brake pads and a rigid surface of the disc in his 3-dimensional model. While a deformable disc is adopted in [5]; however the caliper and the mounting bracket were not included in this model. A deformable disc with more brake components is included in [6,7]. Works that considered all disc brake components and used deformable-to-deformable surfaces of the disc and pads are in [8,9,10].

In the first attempt, a complex eigenvalue analysis was incorporated with the finite element method by building a detailed 3-dimensional disc brake model and validated each of the components over the experimental data by FE modal analysis. The refining process was introduced to bring close frequencies between predicted and experimental results [11]. Some FE models were validated, but only at the components level. In addition, a few researchers have used FE models that were validated at the components and assembly level based on modal testing data, for example, in [11,12].

From the previous studies, it is observed that just a few of them validate the FE disc brake model at both component and assembly stages. This paper introduces an approach to predict disc brake squeal noise. To this end, the finite element model for the disc brake is first carried out then-experimental modal analysis (EMA) is performed to validate the FE model at two stages, i.e. component and assembly levels. After that, a complex eigenvalue analysis made available in ABAQUS is performed, In order to assess the stability of the disc brake assembly. The positive real parts of complex eigenvalue indicate an unstable system. Finally, the effect of friction coefficient disc and the brake pads is evaluated.

2. Development of Finite Element Model

Since brake squeal is a very complex phenomenon because of its strong dependence on many parameters including materials and geometry of brake components, component interaction and many operating and environmental condition. In this research, efforts are focused on the development of an improved FE model, as an extension of the earlier FE brake models, in which a 3-dimensional FE model of the disc brake is developed and validated at both component and assembly level. Hence, the complex eigenvalue results can be used with a higher confidence level for prediction squeal noise for the disc brake system. The schematic flow diagram for the proposed methodology is given in Figure 1 to validate the FE model.

In this study, the FE model consists of a ventilated rotor, a piston, a floating caliper, a mounting bracket, piston and finger pads, two bolts and two guide pins, as shown in Figure 2. The finite element model has up to 27,200 solid elements using a combination of element types C3D4 and C3D8 and has approximately 92,000 degrees of freedom. The validation of the FE model of individual components and the complete assembly is evaluated in comparison with the modal testing results. All the connections between components have been modeled, especially contacts which have been taken into account through surface interactions.

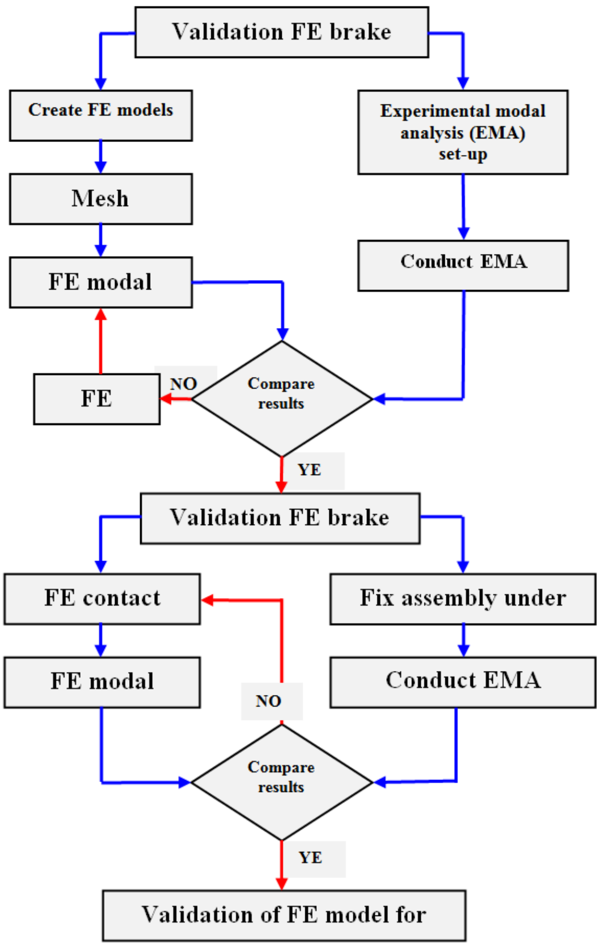


Figure 1. Schematic flow diagram for the validation FE Model

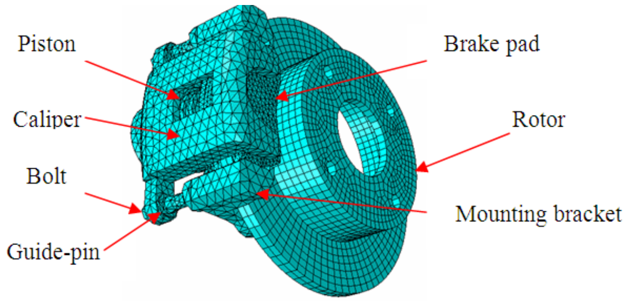


Figure 2. Finite element model of disc brake assembly

2.1. Experimental Modal Analysis of the Disc Brake

In this study, two levels are conducted to validate the FE model using experimental modal analysis (EMA). The first level is to obtain dynamic characteristics of the individual disc brake components with free-free boundary conditions. The second level is to perform dynamic characteristics of the complete assembly with boundary conditions. The type of EMA known as the Frequency Response Function (FRF) method, which measures the input excitation and output response simultaneously is examined for the individual disc brake components and the complete assembly. The FRF was measured by exciting each structure with a small impact hammer with a hard tip. The acceleration response was measured with a light small accelerometer through dynamic signal analyzer type DEWE-41-T-DSA. The FRF measurements were recorded for each structure. Then, by using DEWE FRF software the curve fitting process was performed on the transfer function spectrums obtained to extract the natural frequencies, damping ratios, and mode shapes. Figure 3 shows the experimental modal analysis set-up at free-free conditions. The individual components were fixed on a brake test rig under applied pressure using a hydraulic pump and pressure gauge as shown in Figure 4. The modal characteristics of disc brake assembly are investigated experimentally. The accelerometer is placed at the outer surface of the disc as it has a more regular shape than the other components and the measurement is taken using the impact hammer in the normal direction. From Figure 5, it can be seen that for the frequency range of interest up to 10 kHz, the coherence function is one for all the trials conducted. Hence the measured response is taken as the result of given input excitation.

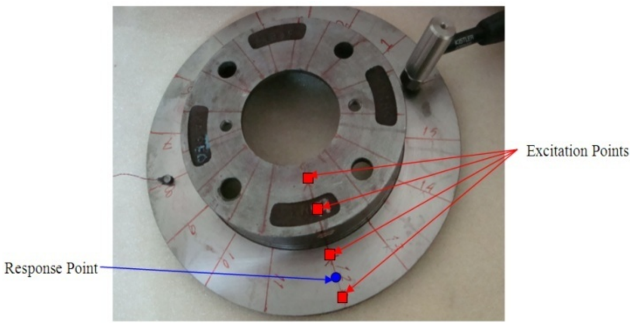


Figure 3. EMA for disc brake at free- free conditions

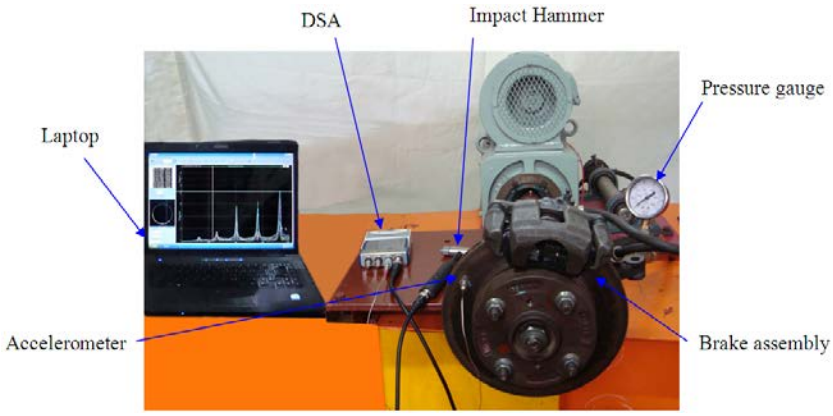


Figure 4. Experimental modal analysis for disc brake assembly

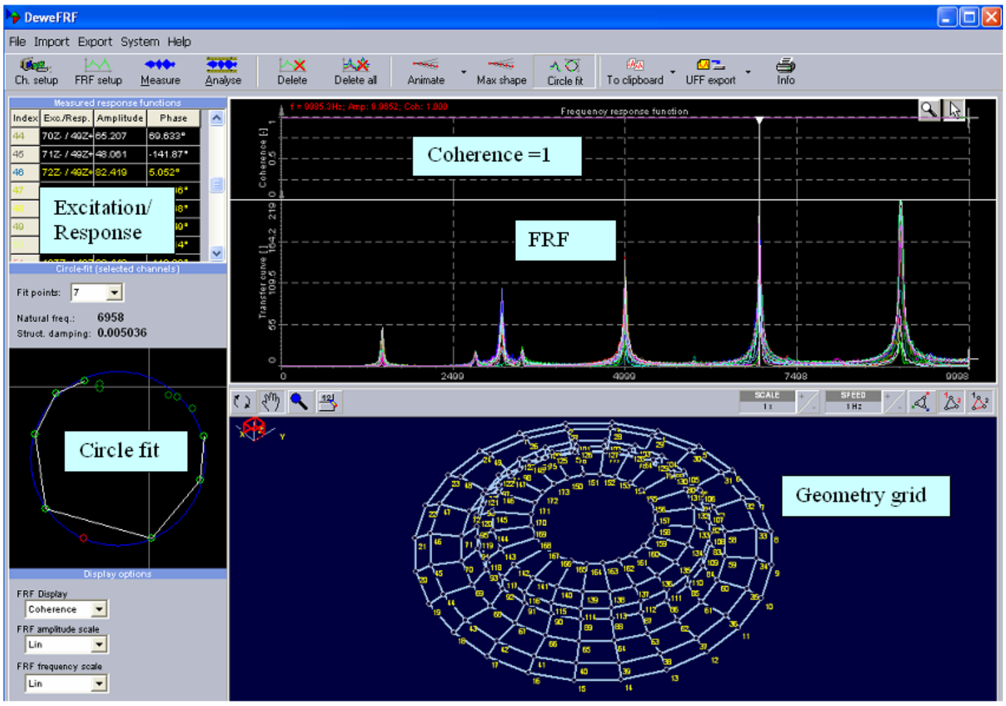


Figure 5. Overview of DEWE/FRF during EMA

2.2. Validation of FE Model

To obtain dynamic properties of disc brake components or assembly, it is common practice to perform experimental modal analysis from which natural frequencies and its graphical mode shapes can be captured easily. Natural frequencies up to 10 kHz are considered since this study takes into account squeal frequencies between 1 to 10 kHz. Each component of the disc brake assembly was examined through the EMA with free-free boundary conditions. The free-free condition allows the structure to vibrate without interference from other parts, making the visualization easier of mode shapes associated with each natural frequency, and easy for FE model validation. Using standard material properties, the predicted frequencies are not well correlated with the experimental results. Hence tuning of the density and Young's modulus for the material properties the predicted results are close to the measured data as shown in Table 1. These results are based on the material properties given in Table 2. To correct the predicted frequencies with the experimental results a FE updating was used to reduce relative errors between the two sets of results by tuning material. This validation of brake components is done to ensure the dynamical properties agree with those of the physical component. The results showed very good correlation between predicted and measured natural frequencies for all brake components.

2.3. Validation at Assembly Level

For validation brake assembly, all FE models of brake components are used to create assembly brake model with the same material properties and FE meshing. A similar experimental condition is also applied to the FE brake assembly model. All boundary conditions and component interfaces were considered. A surface-to-surface contact is used to represent contact interaction between disc brake components as shown in Table 3. As reported by Lee et al. 2019 [13], the contact conditions between rotor disc and friction materials are significant parameters deciding the occurrence of squeal noise. In the experimental modal analysis, a brake-line pressure of 0.7 MPa is imposed to the stationary disc brake assembly. A similar condition is also applied to the FE brake assembly model. In this validation, measurements are taken on the disc as it has a more regular shape than the other components. As shown in Table 4 a good agreement is found between the predicted results and the measured data.

Table 1. Comparison of natural frequencies at free-free boundary condition

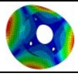
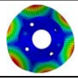
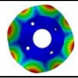
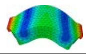
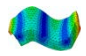
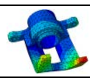
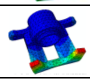
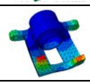
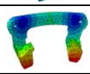
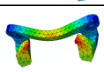
Components	Mode	Mode shape	FE (kHz)	Experimental (kHz)
Rotor	1		1.47	1.40
	2		2.71	2.75
	3		3.78	3.85
Brake Pad	1		3.31	3.25
	2		8.18	8.10
Caliper assembly	1		2.98	2.93
	2		5.58	5.78
	3		8.35	8.40
Mounting bracket	1		0.93	0.92
	2		1.50	1.45
	3		3.41	3.35

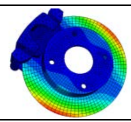
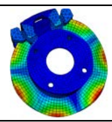
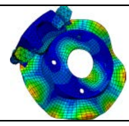
Table 2. Material properties of disc brake component

Components	Modulus Young (GPa)	Density (kgm^{-3})	Poisson's ratio
Disc	110	6880	0.3
Caliper	110	6200	0.3
Piston	210	7890	0.3
Friction material Back pad	2 210	2350 7780	0.3 0.3
Bracket	157	7760	0.3
Bolt	52	9680	0.3
Guide pin	70	7900	0.3

Table 3. Contact interaction between components

Components	Interaction
Disc-brake pad	Surface-to-surface
Caliper-brake pad	Surface-to-surface
Piston-brake pad	Surface-to-surface
Mounting bracket-brake pad	Surface-to-surface
Guide pin-mounting bracket	Surface-to-surface
Piston-caliper	Surface-to-surface

Table 4. Modal results of the brake assembly

Mode	1	2	3
Mode shape			
FEA (Hz)	1585	3389	5980
Exp.(Hz)	1592	3410	6030

3. Predict Brake Squeal by Complex Eigenvalue Analysis

The complex eigenvalue analysis available in ABAQUS is commonly utilized to determine instability of the disc brake assembly. The positive real parts of the complex eigenvalue indicate the degree of instability of the disc brake assembly and are thought to indicate the likelihood of squeal occurrence. 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/pad interface. The disc brake assembly is defined to be unstable or squeal if the real parts of the complex eigenvalue are positive. To perform the complex eigenvalue analysis using ABAQUS, four main steps are required.

They are given as follows:

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

The simulation results revealed that three unstable modes (squeal propensity) can be seen in the frequency range up to 10 kHz with a friction coefficient μ=0.45, the unstable modes are summarized in Figure 6.

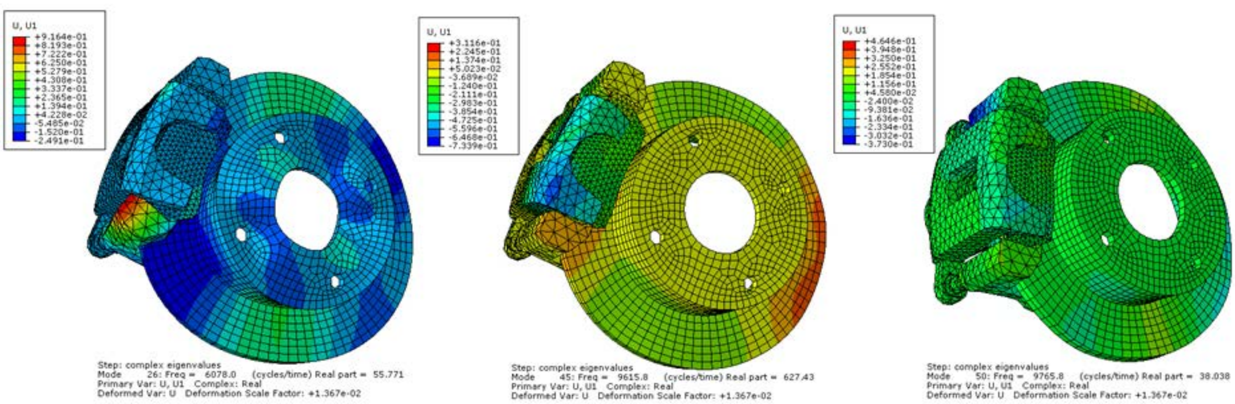


Figure 6. Unstable modes of the disc brake model at μ=0.5

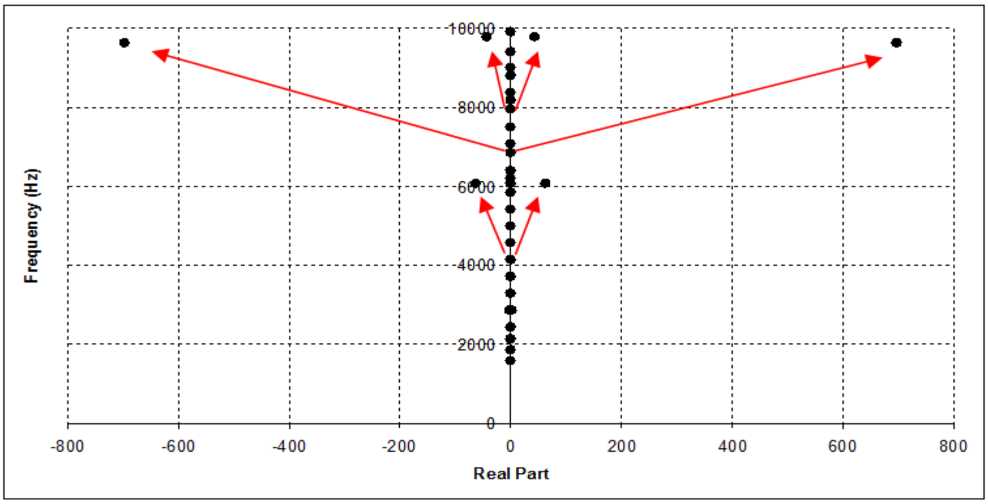


Figure 7. Eigenvalues extracted from the disc brake model plotted on the complex plane

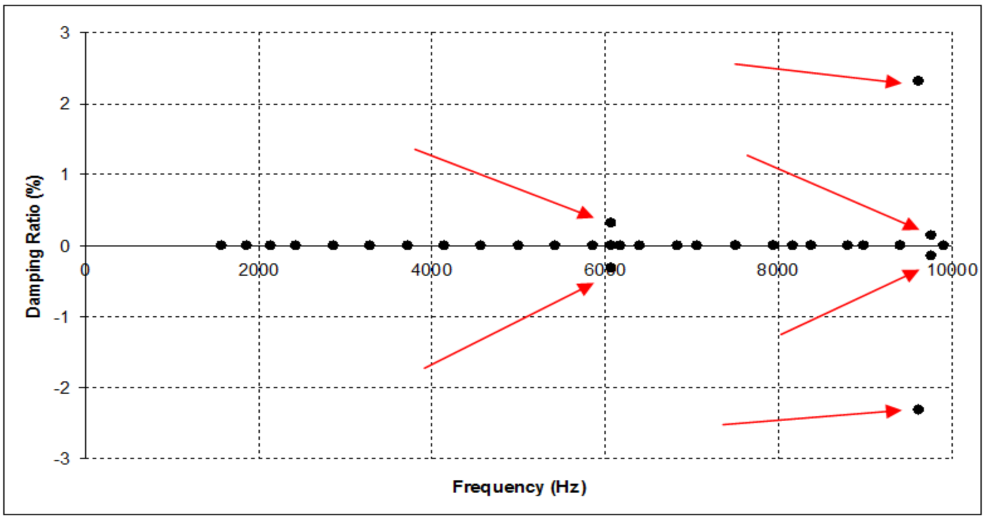


Figure 8. Damping ratio vs. frequency for the disc brake model

To demonstrate the squeal propensity of the disc brake, the complex eigenvalues extracted between zero and 10 kHz for the brake assembly with μ=0.5 are plotted on the complex plane in Figure 7. 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 three unstable modes can be seen at 6075 Hz, 9619 Hz and 9764 Hz. An alternative way to express these results is to plot damping ratio versus frequency as shown in Figure 8. The three modes with positive real parts now appear with negative damping values.

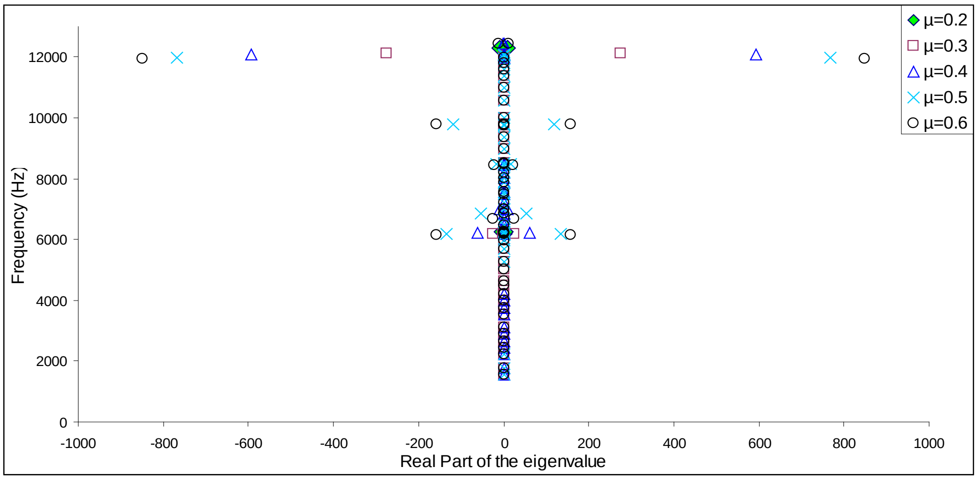


Figure 9. Complex eigenvalue analysis with variation of friction coefficient (μ)

4. Effect of Friction Coefficient

The real and imaginary parts of the complex eigenvalues are, respectively, responsible for the stability and the frequency of the corresponding modes. Since friction is the main cause of instability, a complex eigenvalue analysis has been undertaken to assess the brake stability as the friction coefficient values. It was observed that high values for friction coefficient tend to facilitate two modes merging to form an unstable complex mode. Also, an increase in the friction coefficient leads to an increase in the unstable frequency. Figure 9, shows the results of a complex eigenvalue analysis with variation of the friction coefficient (μ). As predicted in the complex eigenvalue analysis, as the friction coefficient further increases, real parts of eigenvalues, the values that can be used to gauge the degree of instability of a complex mode, increase further, as well, and more unstable modes may emerge. This is because the higher coefficient of friction causes the variable frictional forces to be higher resulting in the tendency to excite greater number of unstable modes.

5. Conclusions

This paper presents numerical analysis of disc brake squeal using a detailed three dimensional finite element model of a real disc brake assembly. To predict squeal frequency from complex eigenvalue analysis the FE model needs to be validated. In this work the model is validated at component and assembly level using experimental modal analysis. Good agreement is achieved at both levels between predicted and measured data. From the complex eigenvalue analysis there are three unstable frequencies predicted at 6075 Hz, 9619 Hz and 9764 Hz. It is also found that the probability of squeal noise Propensity increases with increasing the coefficient of friction between the disc and the brake pads.

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