Understanding Mode-Coupling Mechanism of Brake Squeal Using Finite Element Analysis

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Abstract

Brake squeal is widely accepted by scientists and engineers as a noise which is caused by friction-induced vibrations and it frequently occurs at frequency above 1 kHz. It is one of the most difficult problems and is a big issue in the automobile industry. In recent years, squeal noise prediction methodologies using finite element analysis (FEA) have widely been investigated. Extensive research effort has been undertaken on understanding of brake squeal generation.

This paper is concerned with the FEA and modal testing of a commercial disc brake assembly. The goal is to study squeal noise prediction at early stage of design development using a more realistic FE model. Firstly, the FE model of the disc brake was developed using a 3D solid element. Modal testing using hammer excitation measuring techniques was used to measure the physical vibration properties of the disc brake. The FE model is validated by comparing experimental results of the brake components and assembly with the results obtained from simulation. It is found that good correlation is achieved in the dynamic properties of the brake components and assembly. Next, the validated FE model was used to analyse brake squeal using a complex eigenvalue analysis. Finally, a mode-coupling mechanism that leads to identify brake squeal generation is discussed.

1. Introduction

Disk Brake Squeal noise is a problem that continues to confront automobile manufacturers. Customer complaints result in significant warranty costs yearly. Furthermore, customer dissatisfaction can cause a loss of future business. Physically, squeal noise occurs when the friction coupling between the rotor and pad creates a dynamic instability. This leads to vibration of the structure, which radiates a high frequency noise in the 1- 16 kHz range.

Over the years, a large amount of research work has been carried out to explain and predict the brake squeal phenomenon. (Ibrahim, 1994) in his literature reviewed many theorems and mechanisms. Four typical mechanisms are stick–slip, sprag–slip, negative friction velocity slope, and mode coupling of structures. It was proved theoretically that stick–slip mechanism, sprag–slip, and negative friction velocity slope can cause chaos and instability of the systems, but these theoretical models cannot explain all events related to the squeal noise. (Chen et al., 2003) found that there is no variable correlation between negative friction–velocity slope and the occurrence of squeal noise in reciprocating sliding systems. Squeal noise can occur in regions with both negative and positive friction–velocity slopes. In spite of all of this, mode coupling is generally acknowledged to be one of the most important mechanisms leading to self-excited vibration in relative sliding systems with friction (Hoffman et al., 2003). In modeling of squealing brake systems, according to (Kinkaid et al., 2003) there are over 15 different models for squealing disc brakes in the literature. These models involve 2–14 degrees of freedom. Specially, a finite element model of a squealing brake system includes up to thousands of degrees of freedom. Many understandings of squealing friction systems have been obtained through modeling these friction systems. Some theories for brake squeal generation partially originated from the modeling of friction systems such as the decreasing coefficient of friction with increasing relative sliding speed, sprag-slip and modal coupling (Kinkaid et al., 2003).

In recent years, the finite element (FE) method has gained wide acceptance for modelling brake vibrations and noise problems. Although successful case studies have been reported from time to time, no real reliable brake noise prevention tool truly exist yet (Liles, 1989; Ripin, 1995; Lee et al., 1998; Blaschke et al., 2000; Bajer et al., 2003; AbuBakar et al., 2006; Liu et al., 2007; Mario et al., 2008; Dai et al., 2008; Nouby et al., 2009).

Recent literature reviews (Kinkaid et al., 2003; Papinniemi et al., 2002) reported the complexity and lack of understanding of the brake squeal problem. (Ouyang et al., 2005) reported that experimental methods are expensive due to hardware costs and long turnaround time for design iterations.

Due to a general lack of confidence in FE models, the dynamic testing of structures has become a standard procedure for model validation and updating. Over the past thirty years, modal testing and analysis have become a fast-developing technique for the experimental evaluation of the dynamic properties of structures (Ewins, 2001). One of the earliest researchers who attempted to incorporate the complex eigenvalue analysis with a large finite element model and used modal analysis to compare natural frequencies and its mode shapes for each of disc brake components was (Liles, 1989). Also many researchers validated their models only at the components stage for example, in (Ripin, 1995) and (Lee et al., 1998). For complex eigenvalue analysis of disc brake squeal, some researchers have used FE models that were validated at the components and assembly level based on modal testing data, for example, in (Dom et al., 2003; Ouyang et al., 2003; Goto et al., 2004). Material properties were adjusted in the tuning process to reduce relative errors in natural frequencies between predicted and experimental results. From the previous studies, it is found that just a few of them validate FE disc brake model at both component and assembly level.

The motivation of this paper is to develop a finite element model for a commercial disc brake assembly to predict squeal occurrence. To this end, the finite element model for the disc brake is first carried out then experimental modal analysis (EMA) is performed to correct FE model at two stages, i.e. individual component and assembly levels. After that a complex eigenvalue analysis made available in Abaqus/standard is performed, in order to assess stability of the disc brake assembly. The positive real parts of complex eigenvalue indicate squeal propensity. Finally a mode-coupling mechanism that leads to identify brake squeal generation is discussed. It has been found that the validated FE model by modal testing of the brake components and assembly, can predict the disc brake squeal with satisfactory accuracy.

2. The Finite Element Modelling of Disc Brake

A disc brake system consists of a rotor, 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.

A detailed three-dimensional FE model of a commercial front brake assembly is developed. The FE model consists of a rotor, a piston, a floating caliper, a mounting bracket, piston and finger pads, two bolts and two guide pins, as shown in Fig. 1. The finite element model has up to 27,200 solid elements using combination of element types C3D4 and C3D8 and has approximately 92,000 degrees of freedom. The validation of the FE model of individual component and complete assembly is evaluated in comparison with the modal testing results. Details of these test steps are discussed next. All the connections between components have been modeled, especially contacts which have been taken into account through surface interactions.

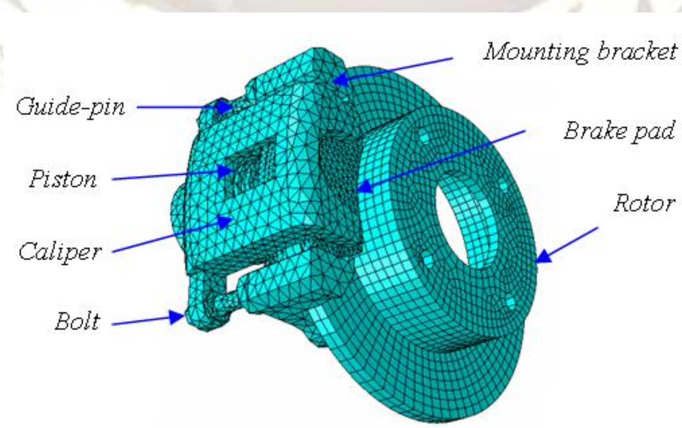


Figure 1. Finite element model of disc brake assembly

3. Experimental Investigation for the Dynamic Characteristics of the Disc Brake

When a FE model is to be validated, the requirement on the convergence of the model must be satisfied. This means that the dynamic properties predicted by the model can be trusted in the frequency range of interest. In this study two stages were used to validate the FE model using experimental modal analysis (EMA). The first stage is to obtain dynamic characteristics of the individual disc brake components with free-free boundary conditions. The second stage 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. Analysis of the experimental data was conducted with DEWE FRF, a commercial modal analysis software package. The Frequency Response Functions were measured by exciting each structure with a small impact hammer with sensitivity of 10mV/N and a hard tip. The acceleration response was measured with a light small accelerometer with sensitivity of 10mV/g 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. Fig. 2 shows the experimental modal analysis set-up.

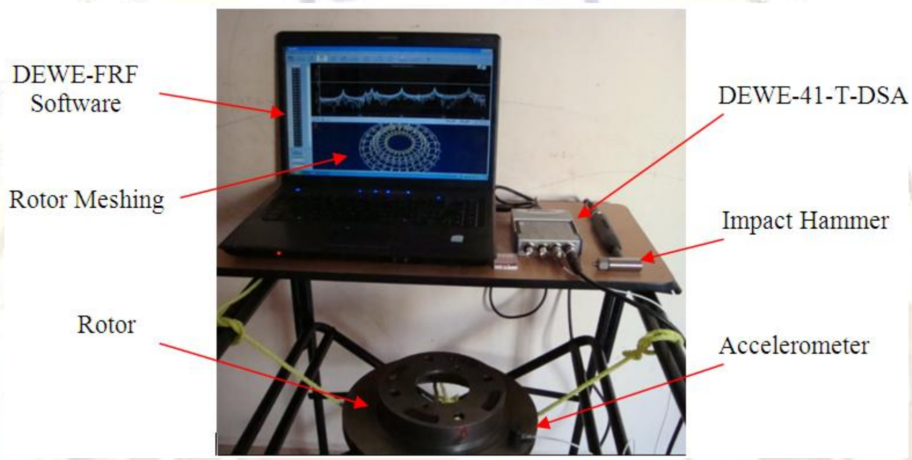


Figure 2. Experimental Modal Analysis set-up

3.1 Dynamic Characteristics of the Brake Components

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. For the process of FE model validation, there must be an initial FE model. Although this model may not be reliable enough to predict the dynamic properties of the structure accurately, it must contain some useful information about the structure's dynamic properties (density and Young's modulus). The density is determined straight forward the volume of the structure is known from the CAD geometry, so the density is adjusted to the measured mass. The Young's modulus is adjusted as standard values. The values for Poisson's ratio were held constant throughout. The mesh refinement technique is considered to determine a suitable mesh density for all disc brake components. In order 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. After tuning process the final material properties of disc brake components are shown in table 1, and the details of FE model validation will be mention below.

Table 1. Material properties of disc brake component

Components	Density ($kg.m^{-3}$)	Young's modulus (GPa)	Poisson's ratio
Disc	6900	110	0.3
Caliper	6100	110	0.3
Bracket	7850	157	0.3
Piston	7918	210	0.3
Guide pin	7850	70	0.3

Bolts	9720	52	0.3
Friction material	2500	2	0.3
Back pad	7800	210	0.3

3.1.1 Modal Analysis of Brake rotor

The modal analysis of rotor is perhaps the most important process to understand the disc brake squeal problem. Dynamic properties like natural frequencies and mode shapes of brake rotor are crucial in defining which type of brake squeal problem may occur. The rotor was suspended by two slender cables and tested in a free-free condition as shown in Fig. 2. The experimental data were obtained via hammer testing using a DEWE-41 analyser connected to a laptop. One piezoelectric accelerometer was attached to response point on the rotor by using wax pee. A total of 80 points arranged a long 16 lines radiating from the centre of the rotor was constructed. Each line featured four excited points and one response point as shown in Fig. 3.

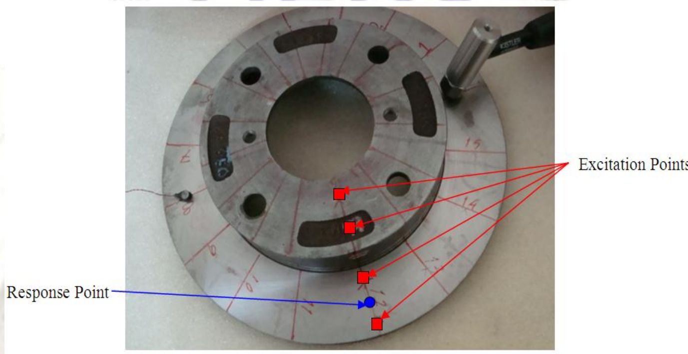
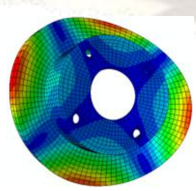
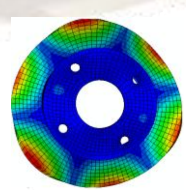
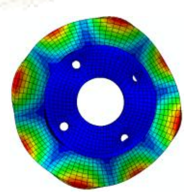


Figure 3. Brake rotor experimental mesh geometry.

The excitation provided by an impact hammer in the out-of-plane direction. As a result, only the bending modes were obtained by this modal testing. The FE model of the rotor is created and modal analysis is predicted using standard material properties. The model consists of 2195 solid elements and 3968 nodes. Then an attempt was made to minimise the variation between finite element analysis results and experimental modal analysis results by tuning of material properties. The rotor is made of cast iron, the Young's modulus of cast iron depending on its carbon, and, to a lesser, silicon content, so the most reliable way to set the material properties is to tune them to the experimental determined modal properties. Based on the tuning process for the rotor the predicted results are close to the measured as shown in Table 2.

Table 2. Modal results of the rotor at free-free boundary conditions

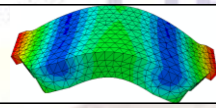
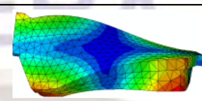
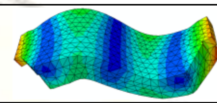
Mode	1	2	3
Exp.(Hz)	1559	2868	4224
FEA (Hz)	1562	2833	4098
Error (%)	0	-1.2	-3
Mode shape			

3.1.2 Modal Analysis of Brake Pad

The brake pad consists of two sections, back plate section made of steel and friction material section made of composite material. The modal testing of the pad was carried out on the backing plate of the pad. The mesh consisted of 17 points, the accelerometer fixed at middle point and the excitation in out-of-plane direction is applied to the rest

of points. Only two bending modes were identified over the frequency range 10 kHz. The finite element model of brake pad consists of 6909 solid elements and 2195 nodes.
For validation, the standard values of steel properties are used for back plate, and tuning material is examined for friction material. A comparison of the brake pad frequencies from FE model compared to those found in modal testing are listed in table 3.

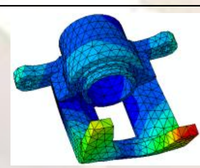
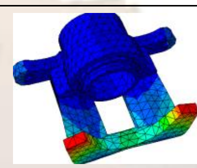
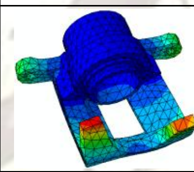
Table 3. Modal results of the brake pad at free-free boundary conditions

Mode	1	2	3
Exp.(Hz)	3159	-	8303
FEA (Hz)	3336	5297	8203
Error (%)	5.1	-	-1.2
Mode shape			

3.1.3 Modal Analysis of Caliper

The floating caliper is a complex geometry. It houses the sliding piston within the brake system. The mesh consisted of 30 points at different position on the outer surface in all three coordinates. The accelerometer is fixed at the middle surface of the calliper housing and excitation is conducted on all points to capture as much of the vibration characteristics as possible. The FE model of the caliper consists of 6169 solid elements and 1846 nodes. By using the same steps for validation, the results of the predicted results show that a good agreement with the measured data as shown in table 4.

Table 4. Modal results of the Caliper at free-free boundary conditions

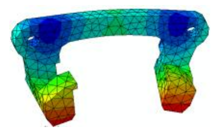
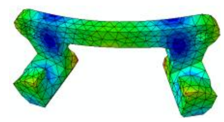
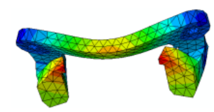
Mode	1	2	3
Exp.(Hz)	3601	6774	8471
FEA (Hz)	3710	6599	8344
Error (%)	2.9	-2.6	-1.5
Mode shape			

3.1.4 Modal Analysis of Mounting bracket

The FE model of the mounting bracket is created and modal analysis is predicted. The FE model consists of 3482 solid elements and 1034 nodes. The modal testing for mounting bracket is performed for 22 points, as a calliper the measurement were conducted in all three coordinates. Tuning material properties for the bracket, which is made of cast iron, is performed. A comparison of the mounting bracket frequencies from FE modal compared to those found in modal test are listed in table 5. It was observed that the experimental and FEA modal frequencies agree to within 5%.

Table 5. Modal results of the mounting bracket at free-free boundary conditions

Mode	1	2	3
Exp.(Hz)	878	1565	3405
FEA (Hz)	880	1638	3486
Error (%)	0	4.4	2.3

Mode shape			

3.2. Dynamic Characteristics Of The Disc Brake Assembly

When the brake system works under pressure the dynamics of the brake components are changed significantly. This behavior is more pronounced for the pad than for the disc, because of the friction material. The disc is made of cast iron while the brake pad has a significant proportion of friction material, which can be compressed by the brake pressure. Due to this compression, the stiffness of the pad is increased, moving the resonances to higher frequencies. On the other hand, the brake pressure has little influence over the disc. Therefore, the resonance frequencies of these two components change at different rates when brake pressure is applied (Triches et al., 2002).

The individual components were fixed on a brake test rig under applied pressure using hydraulic pump and pressure gauge as shown in Fig. 4. Measurements were taken using the majority of the rotor mesh points. The caliper assembly partially obscured the rotor surface meaning only 60 measurements were taken instead of 80. The measurements were confined to the rotor surface. The experimental set up otherwise was the same as for the rotor. The excitation was applied with the impact hammer in the normal direction.

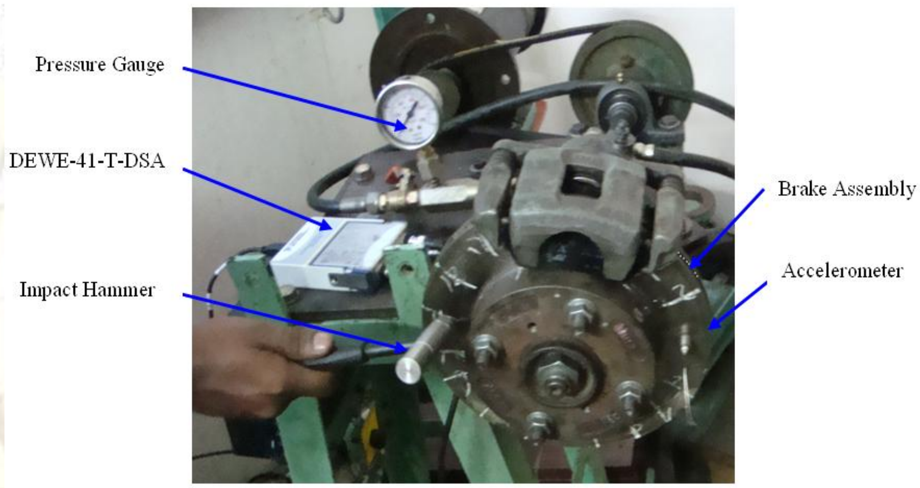


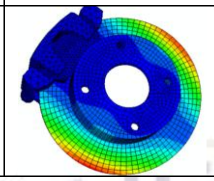
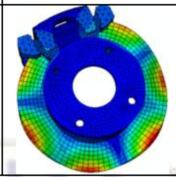
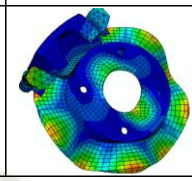
Figure 4. Experimental modal analysis of disc brake assembly.

For the purpose of 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. There are many different methods can be used to represent contact interaction between disc brake components with ABAQUS software. In this section, a surface-to-surface contact element which allows some level of relative motion between sliding components is used between disc brake components as shown in table 6. As shown in Table 7 a good agreement is found between the predicted results and the measured data.

Table 6. 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 7. Modal results of the brake assembly

Mode	1	2	3
Exp.(Hz)	1574	3534	6140
FEA (Hz)	1601	3425	6026
Error (%)	1.6	-3.1	-1.9
Mode shape			

4. Complex Eigenvalue Analysis

Complex eigenvalue analysis (CEA) has become a common tool for investigating the stability of brake system modes. The positive real parts (damping coefficient) of the complex eigenvalues indicate the degree of instability of the disc brake assembly and reflect the likelihood of squeal occurrence. To further gain insight into the behaviour of brake system, a CEA was performed. Using the subspace projection method, the natural frequencies are calculated up to 10 kHz. A description of CEA is beyond the scope of this paper, so the reader is referred to any of the large number of papers have been published on the subject, see for example, (Liu et al., 2007; Mario et al., 2008; Dai et al., 2008; Nouby et al., 2009).

To demonstrate the squeal occurrence of the disc brake, the complex eigenvalues is performed between 1 kHz and 10 kHz for the brake assembly under applied pressure 0.4 Mpa, rotation velocity 5 rad/sec and coefficient of friction 0.45. The CEA result at previous operation conditions is plotted on the complex plane as shown in Fig. 5. In the baseline case no sources of damping are specified. All of the modes have zero damping 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 Y-axis. In this case, there are three unstable frequencies predicted at 6078 Hz, 9615 Hz and 9765 Hz as shown in Fig. 6.

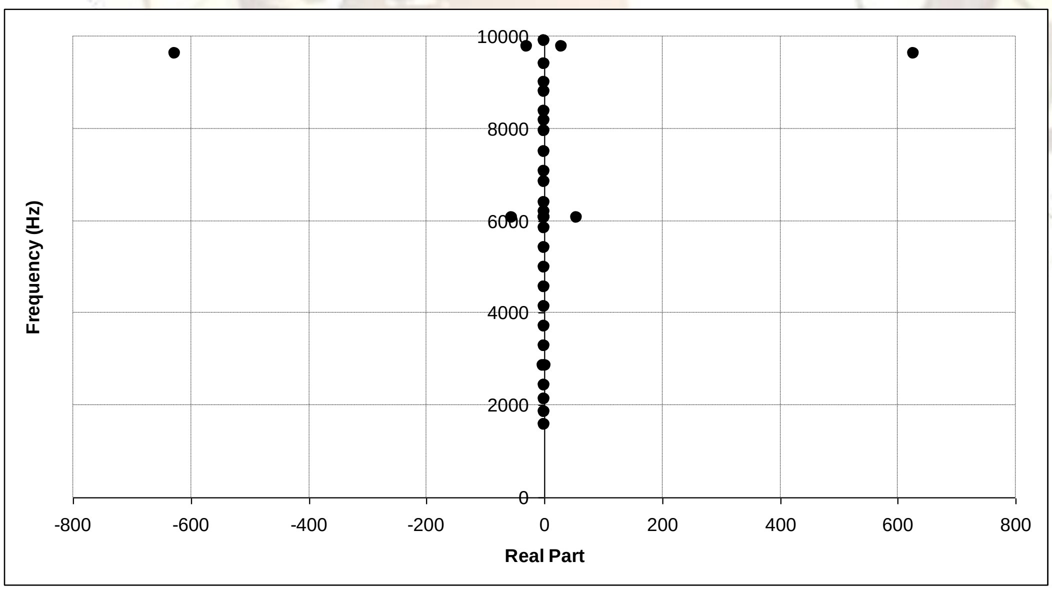


Figure 5. Eigenvalues extracted from the disc brake model plotted on the complex plane at μ =0.45.

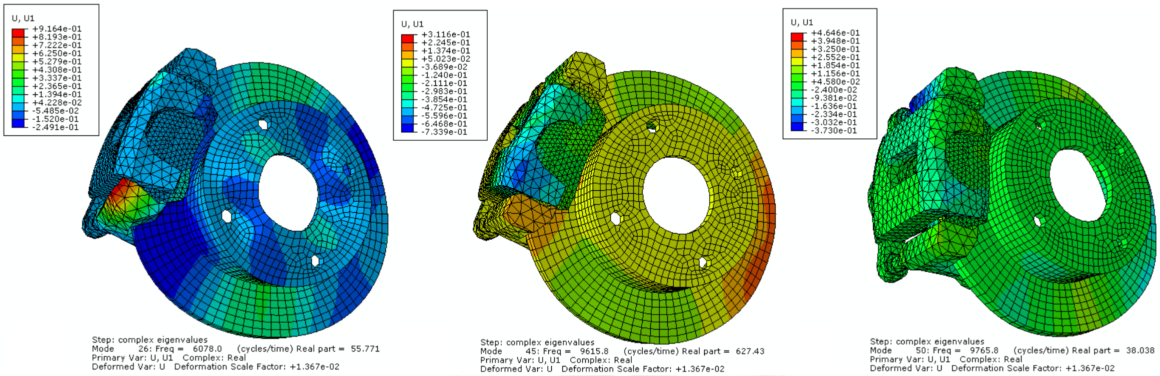


Figure 6. Unstable modes of the disc brake model at μ=0.45.

5. Variation Of Friction Coefficient

Brake squeal is generally defined as a friction induced instability phenomenon. 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. The effect of friction coefficient of the pad-rotor interface is performed. The unstable modes for varying μ from 0.2 to 0.6 are plotted as real parts versus frequency in Fig. 7 to illustrate how the instability increases with friction level. 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 7, shows results in the form of the real part as a function of frequency for different friction coefficients. It was observed that with μ equal to 0.2 one unstable mode is predicted at high frequency 9622 Hz, with increasing friction coefficient values up to 0.6 a numbers of unstable modes are seen to appear. It was observed that the propensity for squeal increases with higher coefficients of friction. 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.

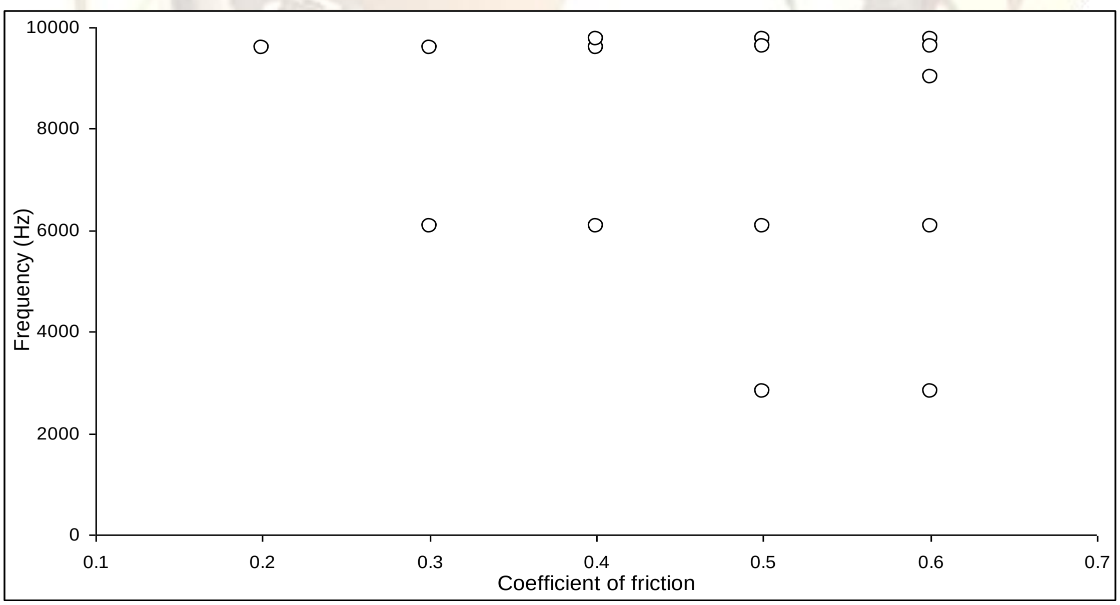
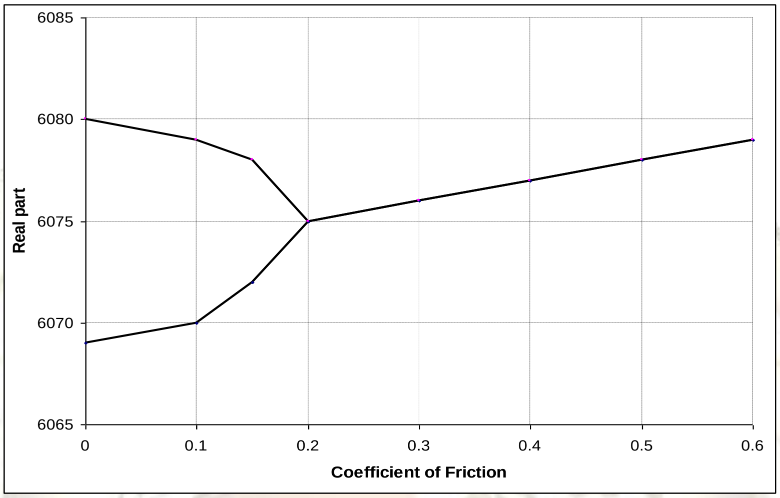


Figure 7. Prediction of unstable frequencies with variation of friction coefficient

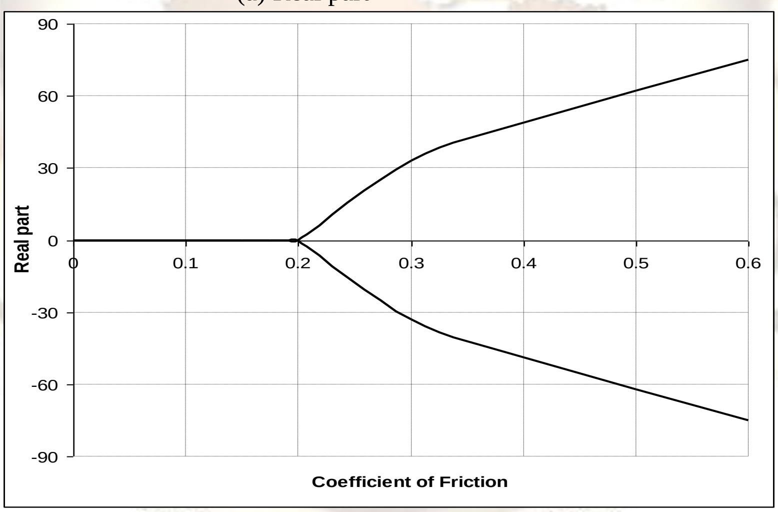
6. Mode-Coupling Mechanism

(Kinkaid et al., 2003) obtained the knowledge of the mode-coupling theory from a simple model with two degrees of freedom. Then they extended it to complicated models with multiple degrees of freedom and finite element models. This FE model will help us to further understand the significance of mode-coupling and extend insights into the brake squeal generation. The FE analyses by complex modes indicate that when two modes close to each other in the frequency range coalesce under the influence of friction (become coupled), the system becomes unstable. To explain coupled mode analysis of the FE model included varying the friction coefficient and observing whether system modes become coupled and unstable.

The influence of friction coefficient on the stability of the 6075Hz mode is examined. Fig. 8 shows real and imaginary part of the eigenvalue pair as functions of friction coefficient μ. It is seen that when μ is less than 0.2 there are two distinct modes at different frequencies. As friction reaches 0.2 some of the adjacent modal frequencies start to merge towards each other and form a complex conjugate pair. At this point, imaginary parts of eigenvalues converge to one value and real parts start to diverge. The meaning of this result is that the coefficient of friction of 0.2 is a critical value of the squeal at 6075Hz. Hence, the merging process gives rise to an unstable mode.



(a) Real part



(b) Imaginary part

Figure 8. Complex eigenvalues of the 6075Hz mode as functions of friction coefficient μ.
(a) Real part, (b) imaginary part.

7. Conclusion

This paper presents numerical analysis of floating caliper disc brake using a detailed three dimensional finite element model of a realistic disc brake system. The FE model is corrected using modal testing data. The modal testing is divided into two stages. The first stage is to obtain dynamic characteristics of the individual disc brake components with free-free boundary conditions. The second stage is to perform dynamic characteristics of the complete assembly. The results showed that good agreement between the FE model using solid elements and measured natural frequencies. The complex eigenvalue analysis has been undertaken as a function of the friction coefficient in order to assess squeal propensity. The simulation result showed that with an increase in the friction coefficient, there is an accompanying increase in the instability of the system. Also the friction coefficient has been investigated to assist the understanding of the mode-coupling mechanism which leads to identify brake squeal occurrence.

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