Uncertainty Analysis for the Prediction of Disc Brake Squeal Propensity

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ABSTRACT

Since brake squeal was first investigated in the 1930s, it has been a noise, vibration and harshness (NVH) problem plaguing the automotive industry due to warranty-related claims and customer dissatisfaction. Accelerating research efforts in the last decade, represented by almost 70% of the papers published in the open literature, have improved the understanding of the generation mechanisms of brake squeal, resulting in better analysis of the problem and better development of countermeasures by combining numerical simulations with noise dynamometer tests. However, it is still a challenge to predict brake squeal propensity with any confidence. This is because of modelling difficulties that include the often transient and nonlinear nature of brake squeal, and uncertainties in material properties, operating conditions (brake pad pressure and temperature, speed), contact conditions between pad and disc, and friction. Although the conventional Complex Eigenvalue Analysis (CEA) method, widely used in industry, is a good linear analysis tool for identifying unstable vibration modes to complement noise dynamometer tests, it is not a predictive tool as it may either over-predict or under-predict the number of unstable vibration modes. In addition, there is no correlation between the magnitude of the positive real part of a complex eigenvalue and the likelihood that the unstable vibration mode will squeal. Transient nonlinear simulations are still computationally too expensive to be implemented in industries for even exploratory predictions. In this paper, a stochastic approach, incorporating uncertainties in the surface roughness of the lining, material properties and the friction coefficient, is applied to predict the squeal propensity of a full disc brake system by using CEA on a finite element model updated by experimental modal testing results. Results compared with noise dynamometer squeal tests illustrate the potential of the stochastic CEA approach over the traditional deterministic CEA approach.

Keywords: Brake Squeal, Complex Eigenvalue Analysis, Uncertainty analysis
I-INCE Classification of Subjects Number(s): 51.4

1. INTRODUCTION

Brake squeal, a frequent source of customers complaints and warranty claims, is caused by friction-induced vibrations and generally occurs in the frequency range of 1 kHz – 16 kHz. It is fugitive and may not be repeatable even under apparently very similar operating conditions. Despite being studied since the 1930s (1, 2) and attracting significant research efforts (3-5) as represented by almost 70% of research papers published in the English Language in the last decade (Figure 1), reliable prediction of brake squeal remains as elusive as ever because of primarily four reasons. Firstly, although brake squeal is an acoustical phenomenon, the traditional practice is to predict unstable vibration modes by applying the complex eigenvalue analysis (CEA) to a finite element (FE) brake model in the frequency domain (6). Acoustic radiation calculations from predicted unstable structural modes are very rarely made (7). There is no guarantee that every predicted

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unstable vibration mode will produce audible squeal as it depends on the radiation efficiency, radiating surface area and amplitude of vibration. Hence, CEA results generally tend to be over-predictive (8, 9). Secondly, nonlinearities in material properties, friction mechanism, boundary conditions and contact conditions between pads and disc have been known to play an important role in brake squeal (10, 11). However, CEA, being a linear analysis tool, has been shown to under-predict the number of unstable vibration modes when nonlinearities are important (10, 12). Thirdly, while most CEA calculations have been conducted using the simplest Coulomb friction model with a constant friction coefficient, more realistic models such as velocity-dependent friction models and state dependent friction models (eg, LuGre model) appear to improve the agreement between predictions and squeal tests (6, 13). However, a friction model that can characterize friction mechanism in both micro- and macro-scales and has been validated against experimental measurements is still lacking. Fourthly, boundary conditions, contact conditions between pads and disc, material properties and operating conditions are not known precisely and instabilities could be sensitive to small variations in these conditions (14, 15). Consequently, the current industry practice still has to rely on extensive squeal tests in brake noise dynamometers or in-situ in vehicles complemented by CEA, modal tests and operating deflection shape measurements for the analysis of squealing structural modes.

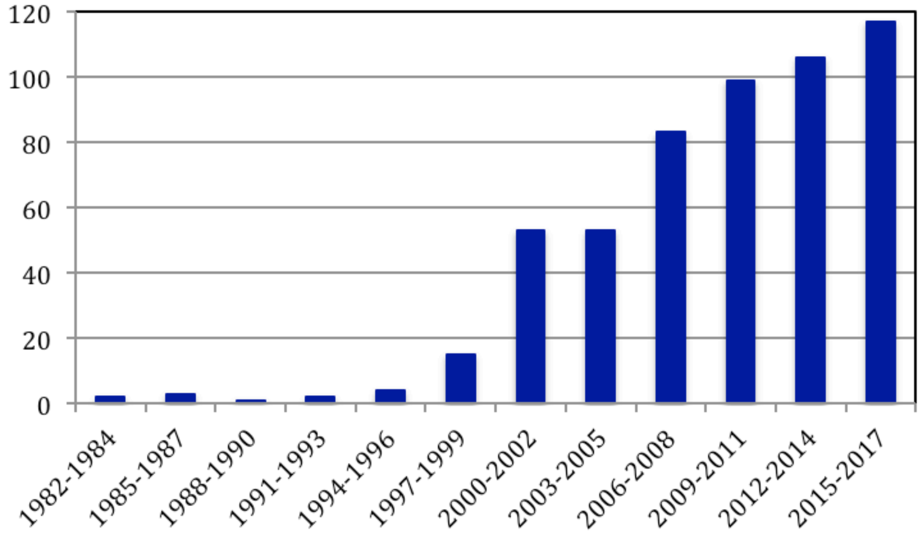


Figure 1 – Number of English Language papers listed under the keywords “brake squeal” by ISI Thomson Web of Science for the period 1982-2017 as at 9 May 2017.

While transient and nonlinear simulations in the time domain (11, 16) for the prediction of unstable vibration modes in a full brake system directly address the transient and nonlinear nature of brake squeal, they are computationally too expensive to be adopted in industries even with the current state-of-the-art computer hardware and software. In addition, issues with regard to friction modeling and uncertainties in material properties, boundary conditions, contact conditions and operating conditions still need to be addressed. Thus, there have been a few recent studies using uncertainty analysis to improve the prediction of unstable vibration modes with some success (14, 15, 17). Both Tison et al (15) and Renault et al (14) found that uncertainties in the pad surface topography were important in improving the agreement between squeal predictions and brake noise dynamometer tests.

By employing an analytical model of 3x3 friction oscillators coupled to a sliding rigid plate with three different friction models (Amonton-Coulomb model, velocity-dependent model and LuGre model), Zhang et al (18) incorporated uncertainties in the friction coefficient parameters and conducted Monte Carlo simulations to examine their effects on the prediction unstable vibration modes by the CEA. They found some vibration modes to be unstable with a high probability of occurrence greater than 70%, independent of the friction model used. These unstable vibration modes, termed “robustly unstable modes”, are, therefore, most likely to cause squeal. By employing an analytical viscously damped self-excited 4-dof friction oscillator coupled with nonlinear stiffness to a sliding belt, Zhang et al (12) showed that the linear CEA under-predicts the instability region compared with a full nonlinear analysis. When uncertainties in the non-linear contact stiffness were incorporated, an uncertainty analysis using the CEA was able to significantly improve the prediction of the instability region compared with the deterministic CEA, hence demonstrating the potential of uncertainty analysis for instability predictions of a nonlinear system.

This study was, therefore, aimed at extending the findings of the analytical models of Zhang et al (12, 18) to a full brake system in predicting squeal propensity by identifying robustly unstable vibration modes using uncertainty analysis that incorporates uncertainties in material properties of pad lining, pad lining surface roughness and friction coefficient parameters for two friction models: Amonton-Coulomb model and the velocity-dependent model with an applied contact pressure of 1 MPa. The linear CEA was used for both uncertainty and deterministic analyses. Results from the uncertainty analysis are compared with deterministic analysis as well as brake noise dynamometer tests to illustrate its potential in improving the reliability of prediction of brake squeal by the CEA.

2. The Updated Finite Element Brake Model

The brake system, manufactured by Chassis Brakes International (Australia) Pty Ltd for Ford SUV 2013 Everest, consists of a rotor, two pad-subassemblies (lining, backplate, shim), a single piston calliper and a bracket. The FE model of the full brake system developed using ABAQUS 6.14 is shown in Figure 2.

Experimental modal testings of all brake components (rotor, calliper and bracket), sub-assemblies (shim-backplate-lining which forms the pad, and the pad-bracket) and the fully assembled brake system were conducted by using a B&K 4809 shaker to excite the structure and by measuring the driving point response with a B&K 8001 impedance head and the response at all other points with a Polytec PSV-400 scanning laser vibrometer in the frequency range of 50 Hz to 10 kHz.

The FE model of the fully assembled brake was built from FE models updated at components level, sub-assembly level and fully assembled level (19, 20) by using experimental modal testing results with the commercial model updating software FEMTOOLS version 3.81. Twelve springs were used to simulate the four abutment clips and connect the backplate ears to the bracket. All components were meshed with 10-nodes nonlinear tetrahedral elements except for the linings which are meshed with 20-nodes nonlinear hexahedral elements in order to facilitate the incorporation of uncertainties in the lining surface roughness. In total, 204,105 elements were considered to be adequate after a mesh-independent study(19). Table 1(a) displays the predicted mode shapes of the fully assembled brake system for 12 modes with their corresponding modal characteristics (frequencies and damping) listed in Table 1 (b). From Table 1(b), the averaged difference in modal frequencies between measurements and predictions is 1.76% and the averaged modal assurance criterion between predicted and measured mode shapes is 82.9%.

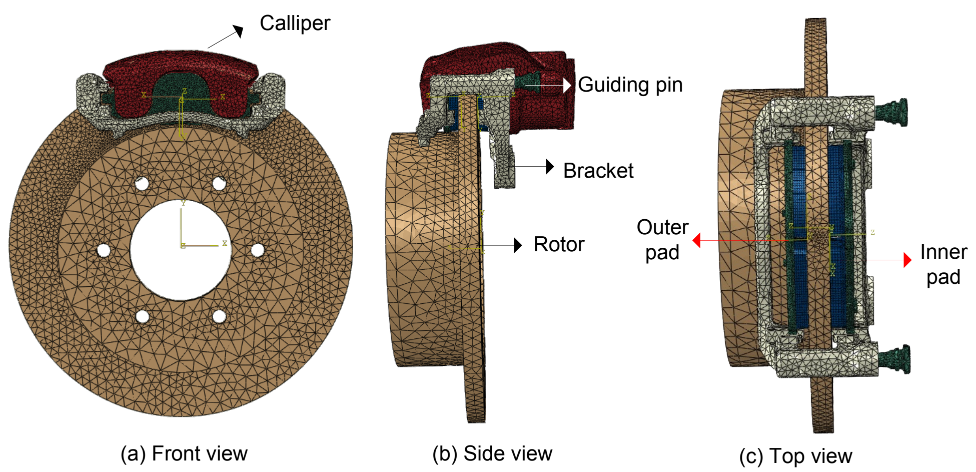
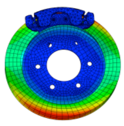
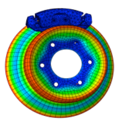
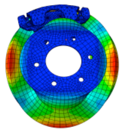
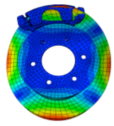
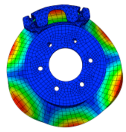
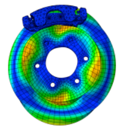
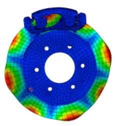
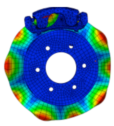
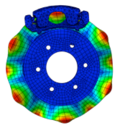
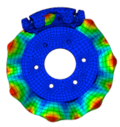
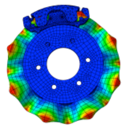
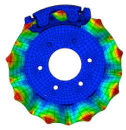


Figure 2 - FE model for a full brake assembly model (a) front view, (b)side view, (c) top view with callipers removed to show the two pads.

Table 1 – Modal Characteristics of the full brake system

(a) Predicted Mode Shapes expressed in (m,n) where m is the number of nodal diameters and n is the number of nodal circles

 (0,1)	 (0,2)	 (1,2)	 (2,0)	 (3,0)	 (3,1)
 (4,0)	 (5,0)	 ‘(6,0)	 (7,0)	 (8,0)	 (9,0)

(b) Modal frequencies and damping and Modal Assurance Criterion (MAC) between measurements and predictions.

Mode	Measured frequency (Hz)	Predicted frequency (Hz)	Frequency difference (%)	MAC between measured and predicted mode shapes (%)	Modal damping (%)
(2,0)	1136	1109	-2.37	95.1	1.27
(0,1)	1466	1550	5.72	84.8	2.23
(0,2)	1615	1743	7.92	85.5	1.10
(1,2)	1854	1853	-0.05	88.0	0.25
(3,0)	1882	1919	1.97	87.9	1.76
(4,0)	2843	2768	-2.64	81.0	1.49
(5,0)	3892	3872	-0.51	75.9	0.67
(6,0)	4588	4685	2.11	79.2	0.45
(3,1)	6261	6387	2.01	88.1	0.43
(7,0)	6688	6712	0.35	86.5	0.49
(8,0)	8207	8563	4.33	80.6	0.62
(9,0)	9099	9372	4.14	78.5	0.05

3. UNCERTAINTY ANALYSIS

In conducting the uncertainty analysis, suitable candidate parameters to be modelled as random variables need to be considered. As the properties of the friction contact have been found to be important factors in initiating brake squeal (6, 14, 21, 22), the friction coefficient, the roughness of the pad lining, the Young's modulus of the pad lining in the *x*- and *z*-directions (tangential and normal direction to the contact interface respectively) and the shear modulus of the pad lining are considered as random variables in this study. On the other hand, the material properties of all other brake components take on fixed values (determined by model updating) as they are much less sensitive to contact pressure compared with the material properties of the brake lining (23). The statistical distributions and the range used for these random variables are described below followed by the procedure for conducting the uncertainty analysis.

3.1 Friction Model

Two friction models are considered here: the Amonton-Coulomb and the velocity-dependent model. For the Amonton-Coulomb model, the friction coefficient μ is constant, as given in equation (1):

$$\mu = \mu^{c} \tag{1}$$

For the velocity-dependent model (24), the friction coefficient $\mu(v)$ is dependent on the velocity v such that it decreases from the static friction coefficient μ_v^s to the kinetic friction coefficient μ_v^k at a rate governed by the parameter v_v^s as the relative sliding velocity $(v-\dot{x})$ between the disc and the pad increases:

$$\mu(v) = \mu_v^k + (\mu_v^s - \mu_v^k)\exp[-(\frac{v-\dot{x}}{v_v^s})^2] \tag{2}$$

The friction coefficient is extracted from brake dynamometer tests and the determination of the statistical distributions of its parameters has been given by Zhang et al (18). Figure 3(a) shows that the friction coefficient is in the range of 0.3 to 0.55 and slightly negatively skewed. The statistical distributions and ranges of the friction coefficient for the Amonton-Coulomb model μ^c, and the static friction coefficient μ_v^s, the kinetic friction coefficient μ_v^k, as well as v_v^s for the velocity-dependent model are given in Table 2.

Table 2 –Statistical distributions and the range of parameters for Amonton-Coulomb and velocity-dependent friction models

	μ^c	μ_v^s	μ_v^k	v_v^s
Distribution	Cauchy(3.9,0.34)	Beta (319.89,517.23)	Beta (362.26,443.66)	Cauchy (7.62, 0.45)
Range	[0.30, 0.55]	[0.39,0.55]	[0.33,0.39]	[5.83, 8.51]

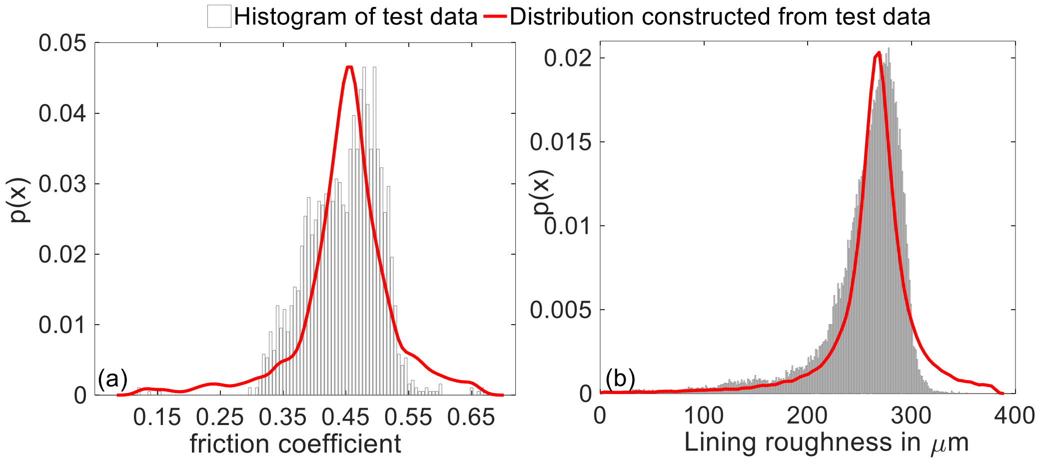


Figure 3 – Histogram and Probability Density Function for (a) friction coefficient; (b) pad lining roughness

3.2 Pad Lining Surface Roughness

The pad lining roughness was measured using a Nanovea profilometer PS50 with a P1-OP3500 measurement pen (30 μm step size in the *x*- and *y*-directions, 1 μm resolution in the *z*-direction and

400 Hz with 5 averages). The x-y plane is in the lining surface with the z-axis normal to it. In contrast to many reported studies assuming a flat lining surface (25-27), the statistical Beta (35.79, 15.39) distribution constructed from the measured roughness in Figure 3(b) shows that the roughness of the lining surface varies from 0 to 360 µm. Here the surface roughness is defined as the difference between the height at each location and the minimum height. The Beta distribution of the roughness in Figure 3(b) is used to generate the z-coordinate of the nodes of pad lining elements in the FE model to inverstigate the influence of the lining roughness on instability prediction.

3.3 Pad Lining Material Properties

The Young's modulus of the pad lining in the x- and z-directions (E_x, E_z) and the shear modulus G_{xz} are assumed to be log-normally distributed (28). The means of the lognormal distributions for E_x, E_z and G_{xz} are taken from the updated material properties as 10.66, 2.99 and 2.48 GPa respectively while the corresponding variances are taken as 20%, 20% and 50% respectively (23).

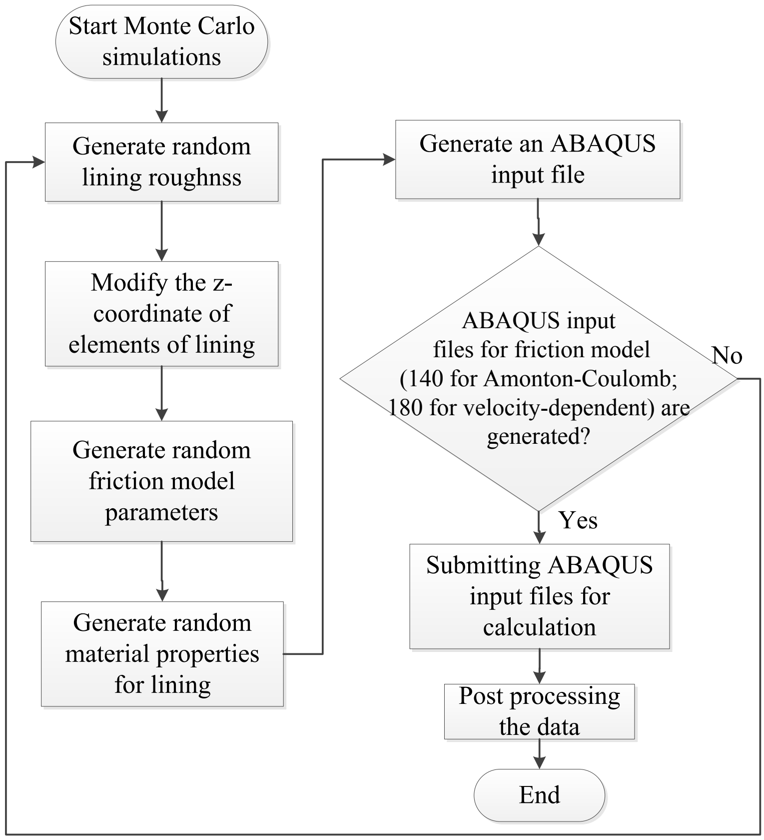


Figure 4 – Flow Chart for conducting uncertainty analysis using Monte Carlo simulations

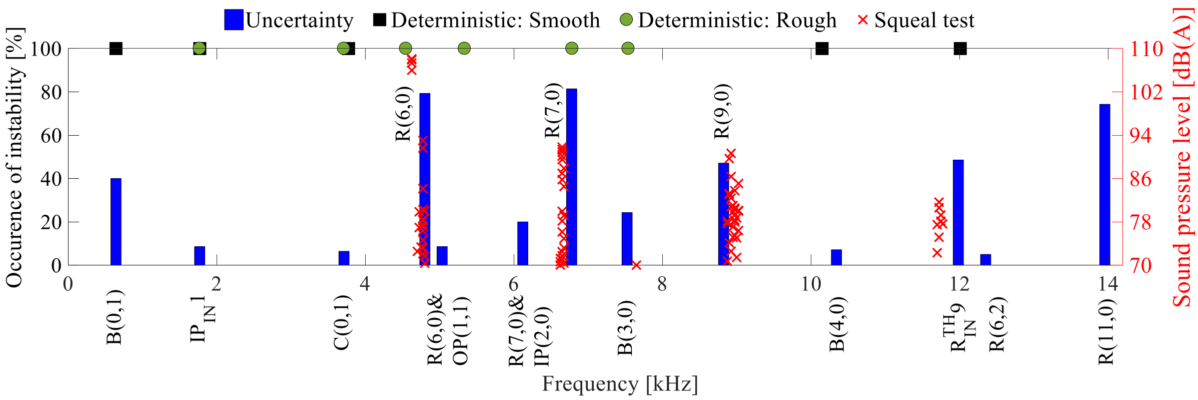
3.4 Procedure for Uncertainty Analysis

Figure 4 displays the flow chart of conducting the uncertainty analysis using Monte Carlo simulations for the prediction of unstable vibration modes. In order to model the lining surface roughness, a FE brake model with a smooth lining surface (Figure 2) is established in ABAQUS with the nodes on the lining surface grouped into a node set. The Beta distribution of the lining roughness given in Figure 3(b) is used to generate random numbers for all the nodes in the node set and the z-coordinate (z-direction is the contact normal direction) for each node is modified by subtracting the generated random number for that node from its original z-coordinate. For each friction model, the values of the parameters for the friction coefficient are generated using the statistical distributions listed in Table 2. This is followed by generating the pad lining material properties from lognormal distributions. For each simulation (ie, a set of random values for the surface roughness, friction coefficient parameters and pad lining material properties), the surface-to-surface contact and the finite sliding contact formulation is applied in ABAQUS to the interface between the pads and the rotor (29) and the instability of the full brake FE model is predicted by the CEA in ABAQUS by applying a contact pressure of 1 MPa for a forward vehicle speed of 10 km/h. In total, 140 and 180 simulations are conducted for the Amonton-Coulomb and the velocity-dependent model respectively.

In order to evaluate the performance of the uncertainty analysis, two deterministic finite element models are employed: a model with a smooth pad lining surface, referred to as "deterministic smooth"; and a model with a rough pad lining surface based on profilometer measurements, referred to as "deterministic rough". For both deterministic models, the pad lining material properties and friction model parameters are taken as the median values of the corresponding statistical distributions.

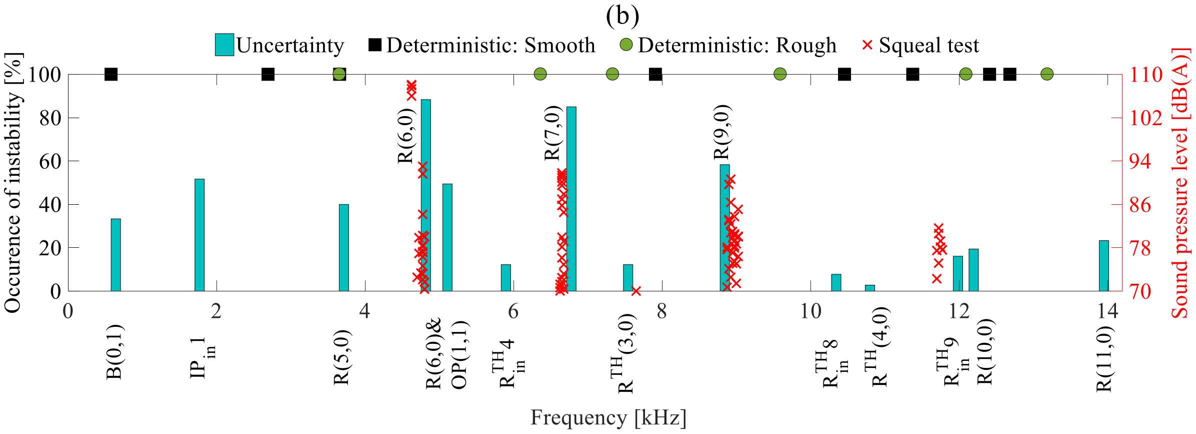
4. RESULTS AND DISCUSSIONS OF SQUEAL PREDICTION

The occurrence of each unstable vibration mode predicted by the uncertainty analysis, the deterministic smooth model and the deterministic rough model is compared with the results of brake noise dynamometer tests in Figure 5 (a) and (b) for the Amonton-Coulomb and velocity-dependent friction models respectively. In Figure 5, the dominant component modes in the predicted unstable vibration modes are identified by B, C, IP, IN, OP, R, IN and TH for Bracket, Calliper, Inner Pad, In-Plane, Outer Pad, Rotor and Top Hat respectively. The results show that none of the unstable vibration modes predicted by the deterministic smooth model is close to any recorded squeal frequency. On the other hand, six unstable vibration modes have been predicted by the deterministic rough model for the Amonton-Coulomb friction model with two of these unstable modes close to the squeal frequencies of 4,751 and 6,672 Hz, clearly performing better than the deterministic smooth model. However, none of the 6 unstable modes predicted by the deterministic rough model with the velocity-dependent model is close to the four recorded squeal frequencies, highlighting the sensitivity of the deterministic rough model to the friction model used.



(a) Amonton-Coulomb Friction Model

(b)



(c) Velocity-dependent Friction Model

Figure 5 – Occurrence of instability for predicted unstable vibration modes with brake line pressure at 1 MPa

For the uncertainty analysis results, Figure 5 shows that with both the Amonton-Coulomb and the velocity-dependent friction models, three predicted unstable modes near the three recorded squeal frequencies of 4,751, 6,672 and 8,974 Hz all have an occurrence of instability of over 50% and are virtually insensitive to the friction model used, hence robustly unstable according to the findings of Zhang et al (18) and most likely to produce squeal. Although there are many more unstable modes predicted by the uncertainty analysis, the occurrence of instability for these other unstable modes is very sensitive to the friction model used and hence they are unlikely to squeal.

5. CONCLUSIONS

A FE model of a fully-assembled brake system updated with experimental modal testing results has been used to test the performance of the uncertainty analysis in improving the reliability of squeal prediction using the complex eigenvalue analysis by comparisons with brake noise dynamometer tests and predictions made by deterministic smooth (lining surface) and deterministic rough models. Statistical distributions constructed from the measured pad lining surface roughness and friction coefficient are used to model the surface roughness and friction coefficient parameters as random variables. The material properties of the pad lining surface (Young's modulus and shear modulus) are also modelled as random variables. In order to test the sensitivity of instability predictions to the friction model used, two friction models are employed: Amonton-Coulomb friction model (ie, constant friction coefficient) and the velocity-dependent friction model. Results show that none of the unstable vibration modes predicted by the deterministic smooth model occurs at any of the recorded squeal frequency. The deterministic rough model correctly predicts two squeal frequencies with the Amonton-Coulomb friction model but none with the velocity-dependent friction model. On the other hand, the uncertainty analysis predicts three unstable modes occurring at three of the four squeal frequencies for both the Amonton-Coulomb and the velocity-dependent friction models while the probability of all other predicted unstable modes is very sensitive to the friction model used. Although CEA is a linear tool, these results demonstrate the potential of the uncertainty analysis (stochastic CEA) in significantly improving the reliability of squeal prediction using deterministic CEA for a full brake system. In this study, the use of two different friction models identifies the robustly unstable modes (ie, most likely to squeal) and eliminates the over-prediction issue often associated with both deterministic and stochastic CEA.

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