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Instability prediction of a fully updated brake system with uncertainty in contact conditions

Zhi Zhang, Sebastian Oberst, Joseph C.S. Lai

Acoustics and Vibration Unit, School of Engineering and Information Technology,
The University of New South Wales, Canberra, Australia

zhi.zhang3@student.adfa.edu.au
s.oberst@adfa.edu.au
j.lai@adfa.edu.au

Abstract

Brake squeal, as friction-induced audible noise above 1 kHz to 20 kHz, is a major concern to automotive manufacturers because of noise, vibration and harshness performance and warranty-claim related costs. The prediction of brake squeal is as difficult as ever due to nonlinearities, uncertainty boundary and operating conditions. The exact contact conditions between pad linings with rotor are often not known and difficult to model. The friction contact has been found to be multi-scaled, inhomogeneous, and highly dependent on operating conditions. In this study, the surface roughness of a pad lining is measured using a Nanovea profilometer with a P1 – OP3500 measurement pen. Statistical distributions are generated for describing the surface roughness. A FE full brake model is updated by modal testing of individual components, subassemblies and full assembled brake. The variation in roughness of pad lining is implemented into the FE model for analysing its effect on brake instability prediction using the conventional complex eigenvalue analysis. The instability predictions are compared with noise dynamometer tests. Results are discussed with a view to applying the methodology to quantify the reliability of squeal prediction against the uncertainty in contact conditions.

Keywords: Brake squeal, full brake system testing, model updating, uncertainty in contact condition, instability prediction

1 Introduction

Brake squeal remains a major concern to the automotive industry owing to noise-related warranty claims incurring significant costs in the budgets for Noise, Vibration and Harshness (NVH) departments [1]. The prediction of squeal occurrence using numerical models of brake systems is as difficult as ever owing to the variability in the material properties, operating conditions and nonlinearity in a brake system [2, 3]. The linear complex eigenvalue analysis (CEA) method as the industrial standard to predict instabilities often either over- or under-predicts the number of instabilities [4]. Further, squeal prediction is difficult due to the many interacting mechanisms that may trigger instabilities such as mode-coupling, instantaneous modes, stick-slip, sprag – slip, negative gradient of friction coefficient with sliding velocity, hammering effect [1, 5, 6]. Also, the simulated dynamics using the finite element (FE) brake model does not accurately represent the true physics of a real brake application. Hence, experimental model updating techniques have been employed to better approximate mechanical properties and their dynamic responses in FE brake models [7-9]. Renault, et al. [10] and Williams et al. [8] applied model updating to improve their FE brake model but only on individual brake components such as rotor and pad, rather than on the assembled system. Abu-Bakar and Ouyang [11] reduced the averaged difference between simulated and measured natural frequencies to 5.3 % for frequencies up to 9 kHz by applying model updating to a full brake system. Tison, et al [12] used a minimum modal assurance criterion of 70% to perform a full brake model updating up to 4 kHz. Oberst et al. [13] and Zhang et. [9] considered experimentally validated roughness in the contact conditions as well as joint stiffness and effects of smearing in subassemblies of pad-bracket and lining-backplate-shim for frequencies up to 20 kHz to better model the connection between individual components in subassemblies for a full brake system. A complete updating process should also consider the vibration response of the full brake system and damping for better brake squeal analysis [14].

Apart from model updating, Zhang et al. [15] showed that an uncertainty analysis applied to an analytical model of a friction oscillator could improve predictions of instabilities using linear CEA in the presence of nonlinearities. Tison et al. modelled the pad's surface topography using a stochastic field method and found CEA's instability prediction better correlated with squeal dynamometer test [12]. Renault, et al. [6] incorporated uncertainties using Monte Carlo method in pad material properties, the pad surface topography, and the operating conditions to conduct local sensitivity and uncertainty simulations. They found the uncertainties due to the pad surface topography important in improving squeal predictions. However, the pad's material properties, its surface topography as well as the friction forces and contact pressure interact in the contact interface and it would be difficult to determine which of these parameters is more important by using a sensitivity analysis which only considers linear effects but not its interactions. To reveal the combined effects and to evaluate their importance on instability generation, an uncertainty analysis should be combined with a global sensitivity analysis [16].

Therefore, this study is aimed at extending the study of Oberst et al. [13] to a fully assembled and updated brake model to determine how the probability of instability occurrence is affected by changes of three parameters: the pad's material properties, the pad's surface topography and the friction coefficient. Similar to Zhang et al. [15], a global sensitivity analysis combined

with an uncertainty analysis is used. The prediction is compared with brake dynamometer tests of a squealing brake system to evaluate the instability prediction quality.

2 A full brake model and model updating

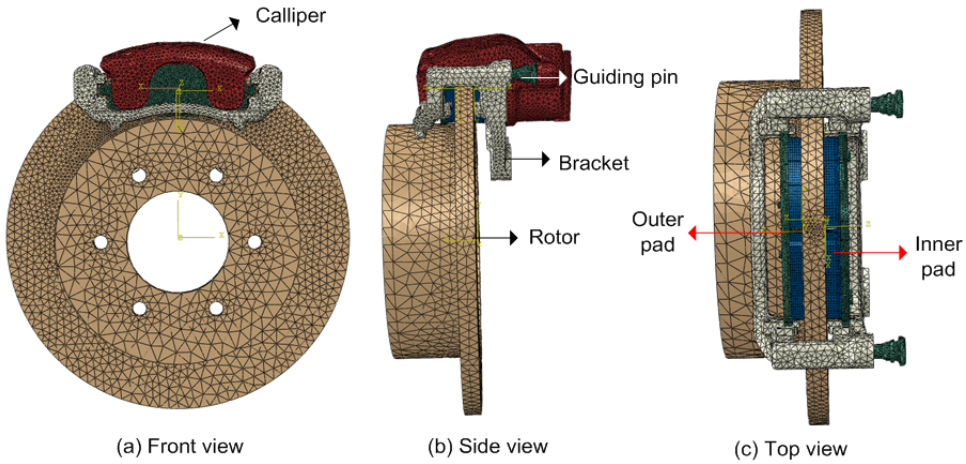


Figure 1. A FE full brake assembly model for brake squeal prediction (a) top view, (b)side view, (c) top view with the calliper not present to show the two pads

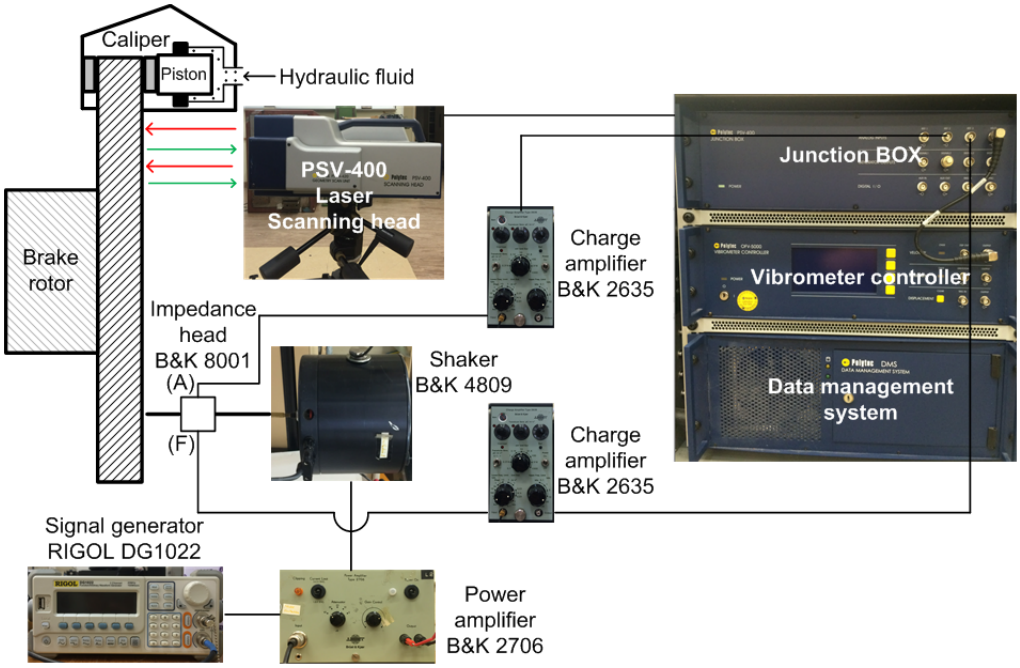


Figure 2. Schematic of the experimental set-up

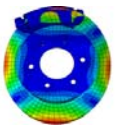
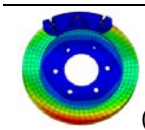
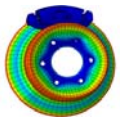
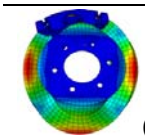
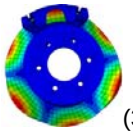
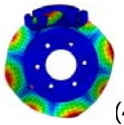
A full brake FE model (**Figure 1**) consisting of a rotor, two pad- assemblies (lining, backplate, shim), a single piston calliper and a bracket is set up using Abaqus 6.14. To include the stiffness of the abutment clips (details [9, 13]), twelve springs connect the backplate ears to the bracket in the *x*-, *y*-, and *z*- directions respectively. All components are meshed with 10-nodes nonlinear tetrahedral elements except for the linings which are meshed with 20-nodes nonlinear

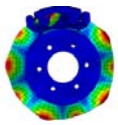
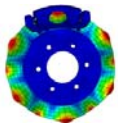
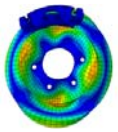
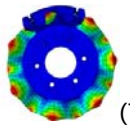
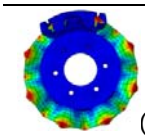
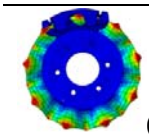
hexahedral elements. In total, 204,105 elements are used. The surface-to-surface contact with Amonton-Coulomb type friction and finite sliding contact formulation with the possibility of separation is applied to the interface between pads and the rotor. In our previous study, the model has been successively updated through mesh-independent study [8], model updating on component [8] and sub-assembly level [9], and application of correct pressurisation area [9].

2.1 Modal testing of the full brake system

Modal testing was conducted on the full brake system using the equipment set up in **Figure 2**. A shaker was used to excite the brake system with the impedance head (B&K 8001) measuring the driving point accelerance. The modal testing was conducted by using a Polytec Scanning Laser Vibrometer PSV-400 in a frequency range of 50 Hz to 10 kHz which is the frequency limit of the impedance head. The scanning surface was sprayed with white primer to improve the signal to noise ratio and to reduce the effect of speckle noise. The mechanical properties of the FE brake model are updated using the commercial model updating software FEMtools 3.8.1. The updated results in Table 1 for 12 modes indicate the reduction of the averaged relative error of frequency from 2.59% to 1.76%, with an increase in the averaged modal assurance criterion (MAC) between the predicted and measured mode shape from 79.3% to 82.9%.

Table 1. Correlation of FE with modal testing results of full brake system, (B: baseline, U: updated)

Mode	Pressure MPa	Measured frequency (Hz)	Simulated frequency (Hz)		Relative error of frequency (%)		MAC between measurement with FE (%)		Modal damping (%)
			B	U	B	U	B	U	
 (2,0)	0.5	1124	1079	1086	-4.01	-3.38	81.3	86.1	1.99
	1.0	1136	1082	1109	-4.75	-2.37	90.2	95.1	1.27
	1.5	1065	1080	1082	1.41	1.59	89.1	90.3	1.09
	2.0	1099	1084	1092	-1.36	-0.63	88.4	91.4	1.21
 (0,1)	0.5	1459	1556	1562	6.64	7.05	70.4	79.7	1.42
	1.0	1466	1557	1550	6.20	5.72	75.6	84.8	2.23
	1.5	1476	1521	1518	3.04	2.84	79.3	80.1	2.63
	2.0	1465	1508	1501	2.93	2.45	84.5	84.5	2.41
 (0,2)	0.5	1610	1759	1750	9.25	8.69	76.4	83.3	1.16
	1.0	1615	1766	1743	9.34	7.92	79.1	85.5	1.10
	1.5	1626	1742	1728	7.13	6.27	80.6	82.3	1.16
	2.0	1630	1733	1721	6.32	5.58	78.7	80.1	0.90
 (1,2)	0.5	1843	1860	1858	0.92	0.82	80.1	86.9	0.66
	1.0	1854	1862	1853	0.43	-0.05	82.2	88.0	0.25
	1.5	1860	1864	1860	0.21	0.00	79.4	79.9	0.78
	2.0	1865	1865	1866	0.00	0.05	81.7	81.8	0.82
 (3,0)	0.5	1902	1938	1927	1.89	1.31	82.4	87.0	1.27
	1.0	1882	1937	1919	2.92	1.97	83.3	87.9	1.76
	1.5	1902	1939	1935	1.94	1.74	85.4	85.7	0.72
	2.0	1923	1941	1940	0.93	0.88	83.4	88.1	0.82
 (4,0)	0.5	2770	2780	2783	0.36	0.47	80.4	81.6	0.41
	1.0	2843	2779	2768	-2.25	-2.64	80.9	81.0	1.49
	1.5	2861	2745	2801	-4.05	-2.09	81.3	82.0	1.91
	2.0	2853	2688	2789	-5.78	-2.24	82.6	82.4	1.94

 (5,0)	0.5	3785	3788	3786	0.07	0.02	80.1	80.7	0.41
	1.0	3892	3834	3872	-1.49	-0.51	75.7	75.9	0.67
	1.5	3918	3851	3839	-1.71	-2.01	78.8	80.1	0.87
	2.0	3907	3860	3867	-1.20	-1.02	90.8	91.0	0.65
 (6,0)	0.5	4566	4716	4889	3.28	7.07	80.5	81.6	0.58
	1.0	4588	4699	4685	2.41	2.11	80.1	79.2	0.45
	1.5	4637	4705	4724	1.46	1.87	85.4	88.1	0.77
	2.0	4702	4887	4810	3.93	2.29	77.6	80.1	0.35
 (3,1)TH	0.5	6132	6145	6179	0.21	0.76	81.5	91.2	0.30
	1.0	6261	6373	6387	1.79	2.01	85.4	88.1	0.43
	1.5	6375	6471	6472	1.50	1.52	78.4	85.5	0.56
	2.0	6324	6378	6475	0.85	2.38	79.1	79.2	0.44
 (7,0)	0.5	6720	6741	6789	-0.58	0.13	81.0	90.6	0.39
	1.0	6688	6734	6712	0.68	0.35	85.4	86.5	0.49
	1.5	6631	6721	6709	1.35	1.17	83.7	86.1	0.43
	2.0	6716	7017	6904	4.48	2.79	79.3	83.7	0.32
 (8,0)	0.5	8217	8324	8387	1.30	2.06	82.2	83.5	0.39
	1.0	8207	8524	8563	3.86	4.33	81.3	80.6	0.62
	1.5	8271	8614	8576	4.14	3.68	71.6	79.2	0.33
	2.0	8866	8880	8829	0.15	-0.41	67.1	70.5	0.16
 (9,0)	0.5	9018	10377	9281	15.06	2.91	73.0	71.9	0.02
	1.0	9099	10383	9372	15.37	4.14	75.4	78.5	0.05
	1.5	9089	10368	9288	14.07	2.18	72.4	77.1	0.02
	2.0	9146	10389	9401	13.59	2.78	71.2	75.3	0.04

2.2 Squeal noise dynamometer testing

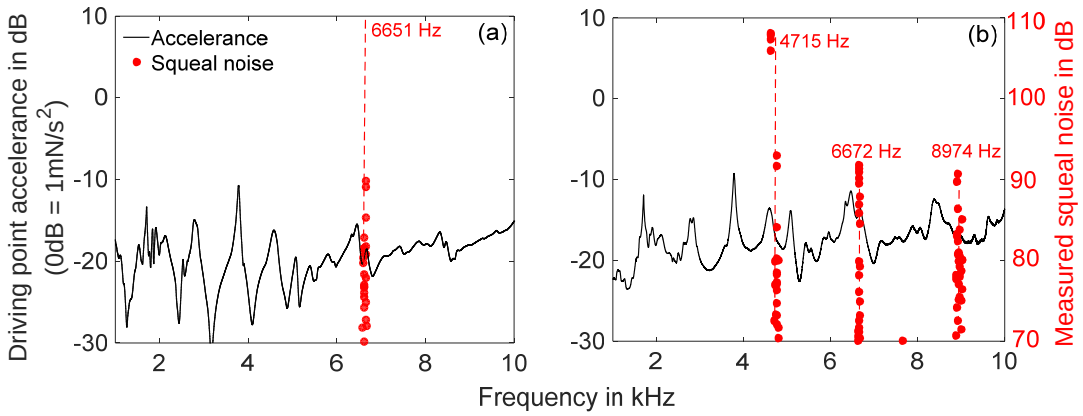


Figure 3. Driving point accelerance vs squeal test (SPL>70 dB) (a) 0.5 MPa, (b) 1.0 MPa

The squeal noise of the brake system was tested in a computer-controlled industrial noise dynamometer. All squealing events of the cold test section with a sound pressure level (SPL) higher than 70 dB was recorded according to SAE J2521 [17]. The frequencies of the squealing events are compared with the measured driving point accelerance in Figure 3 for 2 different contact pressures (0.5 and 1.0 MPa) to determine which mode is dominant at each squeal frequency. For a contact pressure of 0.5 MPa, Figure 3 (a) shows that the mode dominated by the rotor (7, 0) mode, (7 and 0 represent the number of nodal diameters and nodal circles respectively), is the most likely contributor to the instability because its mode frequency (6680

Hz, in Table 1) is closest to the squeal frequency (6651 Hz). Siimilarly, by analysing the results in Figure 3 (b), other modes dominant for squeal are identified and the results are summarised in Table 2.

3 Uncertainty quantification

Table 2. Modes responsible for triggering squeal

Pressure (MPa)	Squeal frequency (Hz)	Mode's frequency (Hz)	Mode shape	Entry in Table 1
0.5	6651	6680	Rotor (7,0)	10th
1.0	4715	4588	Rotor (6,0)	8th
	6672	6688	Rotor (7,0)	10th
	8974	9018	Rotor (9,0)	12th

As highlighted by Oberst and Lai [3], the properties of frictional contact are most important factors influencing the occurrence of brake squeal (see also [10, 12]). Therefore, the friction coefficient, the roughness of the lining material in contact, and the elasticity of pad in the *x*-direction (tangential) are considered as random parameters in this study. The friction coefficient is extracted from dynamometer tests. The friction coefficient measurement is detailed in [18] and its statistical distribution is displayed in **Figure 4** (a) indicating the friction coefficient is slightly negatively skewed and in the range of 0.3 to 0.55.

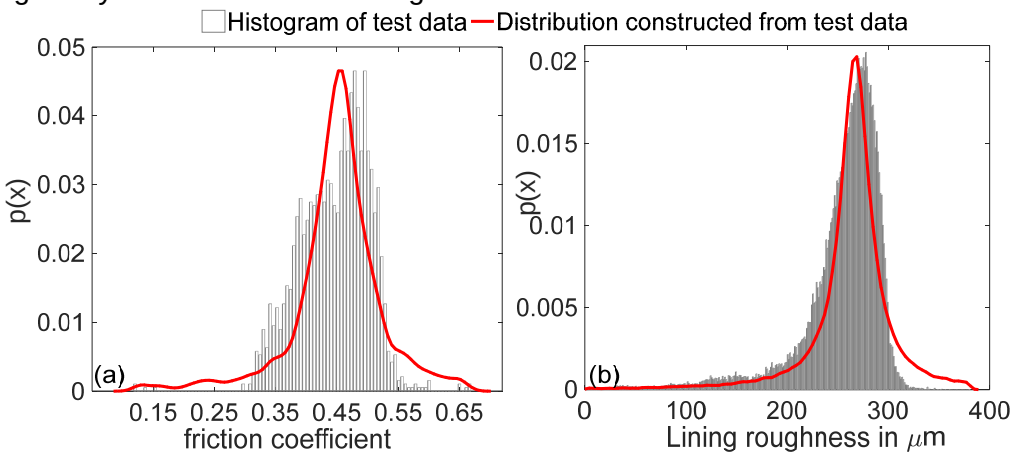


Figure 4. Histogram of pdf density function for (a) friction coefficient, (b) roughness of pad lining

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*-direction, 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 [19-21], the measured roughness and its statistical distribution in **Figure 4** (b) both show that the roughness of the lining surface varies from 0 to 360 µm. Therefore constructed statistical distributions are used to generate random friction coefficient and lining roughness for the uncertainty analysis, (more details [12]). The elasticity of two mating objects significantly affects their contact [22]. Therefore, the variability of the Young's modulus of the rotor and pad needs to be considered in the uncertainty analysis. As shown by Yuhas et al [23] and Oberst and Lai [24], there is much less variation in the Young's modulus with contact pressure for the rotor than for the pad and the longitudinal and shear modulus of lining can be varied by 10% when the pressure increases from 1 MPa to 3 MPa. Thus, the longitudinal modulus in the contact normal direction E_x and shear modulus G_{xz} are taken as lognormally distributed (c.f. [25]) random parameters. The

mean and variance of the lognormal distributions for E_x (G_{xz}) are 3.1 GPa (2.0 GPa), and 0.096 GPa (0.04 GPa) respectively. Other material properties are taken as constant values [13].

4 Numerical brake squeal prediction using uncertainties

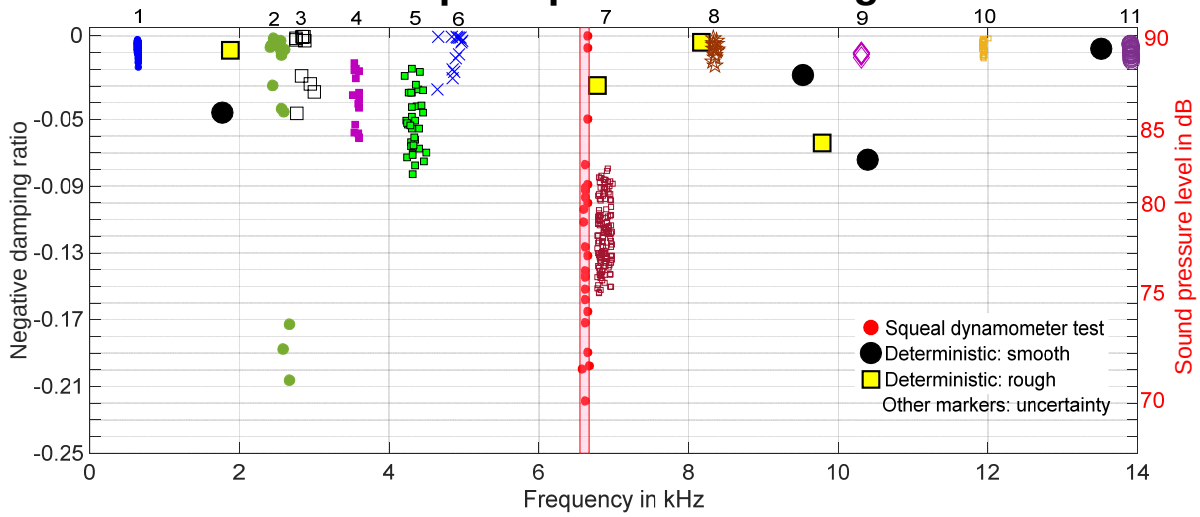


Figure 5. Comparison of instability prediction of deterministic and uncertainty analysis with squeal dynamometer tests (Pressure = 0.5 MPa)

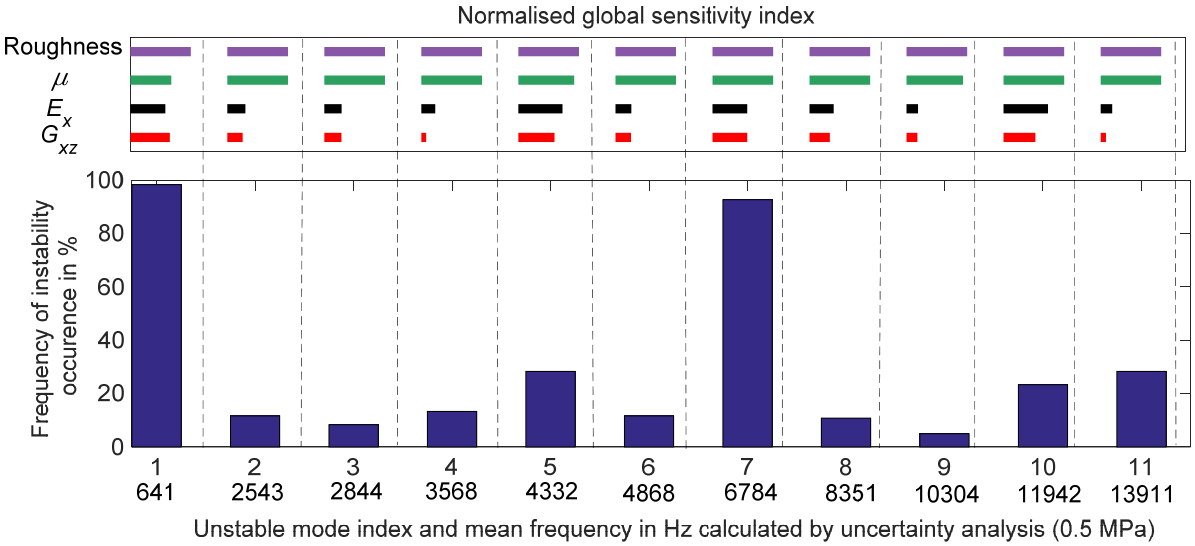


Figure 6. Frequency of occurrence and global sensitivity index for predicted unstable mode (Pressure = 0.5 MPa)

To account for the uncertainty in the surface roughness, the coordinate of each node on the pad lining contact surface is adjusted using the statistical distribution in **Figure 4 (b)**. The uncertainties in the friction coefficient and elasticity of the pad lining are implemented by adjusting the corresponding parameters in the Abaqus INP file. For each contact pressure, 120 simulations of the complex eigenvalue extraction were conducted. **Figure 5** shows the instability predictions for a full brake model from the deterministic analysis (for pads with smooth and rough surface respectively with nominal parameter values) and the uncertainty analysis compared with the noise dynamometer test results for a contact pressure of 0.5 MPa. The

damping ratio $-2\lambda/|\omega|$ is used here as the instability indicator. Figure 5 shows that none of the 4 predicted unstable modes from the smooth pad deterministic model is close to the squeal frequency and mode (6651 Hz, dominated by the rotor mode (7,0) in Table 2) but one of the 4 predicted unstable modes from the rough pad deterministic model is very close to the squeal frequency. In contrast, the uncertainty analysis predicts 11 unstable modes, and the 7th unstable mode's mean frequency (6784 Hz) is close to the measured squeal frequency and its mode shape is dominated by the rotor mode (7,0). **Figure 6** shows the frequency and global sensitivity index [16] of occurrence of the eleven unstable modes predicted by the uncertainity analysis, indicating the highest frequency of instability occurrence for the 1st and the 7th mode and the importance of the roughness and friction coefficient. The first unstable mode's frequency of 641 Hz, below the threshold of 1 kHz for brake squeal, is dominated by a calliper mode. As shown in **Figures 5** and **6**, both the deterministic and the uncertainty analyses over-predict the number of instabilities but the frequency of occurrence of all unstable modes predicted by the uncertainty analysis is significantly lower than that of the squealing mode. For a contact pressure of 1.0 MPa, four squealing frequencies at 4715Hz, 6672Hz, 8974 Hz, and 11632 Hz identified in noise dynamometer tests are shown in **Figure 7**. The squealing events of 11632 Hz are not given in **Figure 3** because the upper frequency limit of the impedance head is 10000 Hz. Five instabilities were predicted by the deterministic analysis for smooth pads with only 1 predicted unstable mode's frequency (12011 Hz) close to the measured squeal frequency of 11632 Hz. Two of the 5 predicted unstable modes from the deterministic analysis for rough pads are close to the squealing frequencies near the 5th and 7th modes. In contrast, 12 unstable modes were predicted by the uncertainty analysis with the 5th, 7th, 9th unstable modes correctly predicting the corresponding test squealing frequencies and squealing modes (dominated by rotor (6,0), rotor (7,0) and rotor (9,0), referred to Table 2). The 11th unstable mode's mean frequency, 11580 Hz, is also close to the squealing frequency 11632 Hz, but the squealing mode is not identified in the modal testing. Different from the results in **Figure 6,** the correctly predicted unstable modes do not have the highest frequency of occurrence in **Figure 8** while the 4th and 12th unstable modes occur most frequently, suggesting the over-prediction of the uncertainty analysis for a contact pressure of 1.0 MPa is greater than for 0.5 MPa.

5 Conclusion

A global sensitivity analysis combined with an uncertainty analysis and the complex eigenvalue analysis method has been applied to an experimentally validated FE full brake model to predict brake squeal propensity. The roughness of the pad lining surface, the friction coefficient and the elasticity of pad lining in tangential direction are modelled as random parameters. The global sensitivity analysis reveals the contact surface roughness and friction coefficient strongly influence the generation of instabilities more than the material properties of pad. By comparing the results with test squeal data, the uncertainty analysis has been shown to provide better predictions than the deterministic analysis for both smooth and rough pads. However, the uncertainty analysis still over-predicts the number of instabilities and the over-prediction increases with the brake pressure. This over-prediction issue may be alleviated by a feed-in energy analysis which will be the subject of a future study.

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provision of brake squeal dynamometer test data and the test rig for assembling brake system by Chassis Brakes International is gratefully acknowledged.

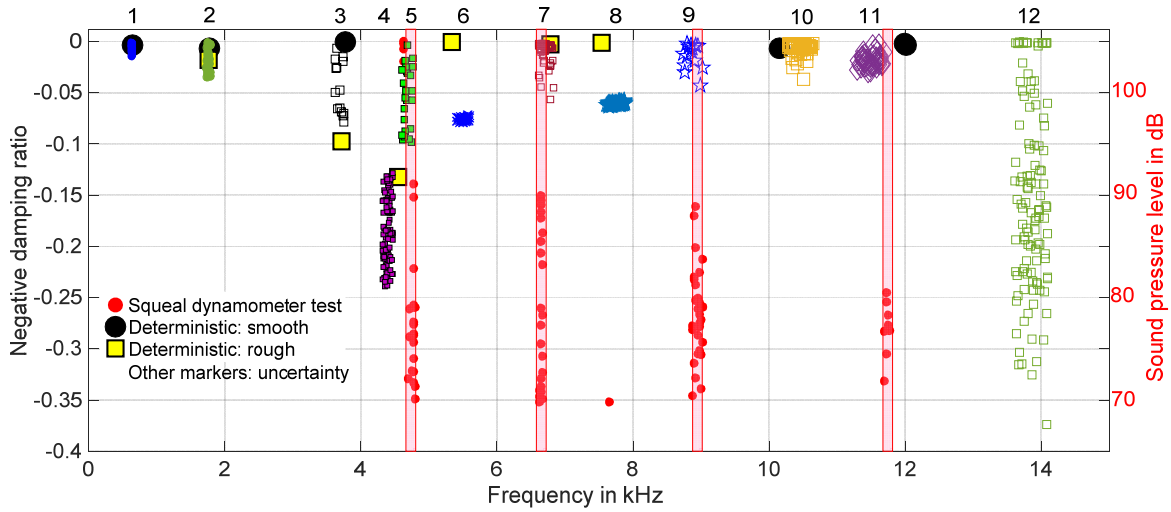

Figure 7. Comparison of instability prediction of deterministic and uncertainty analysis with squeal dynamometer test (Pressure = 1.0 MPa)

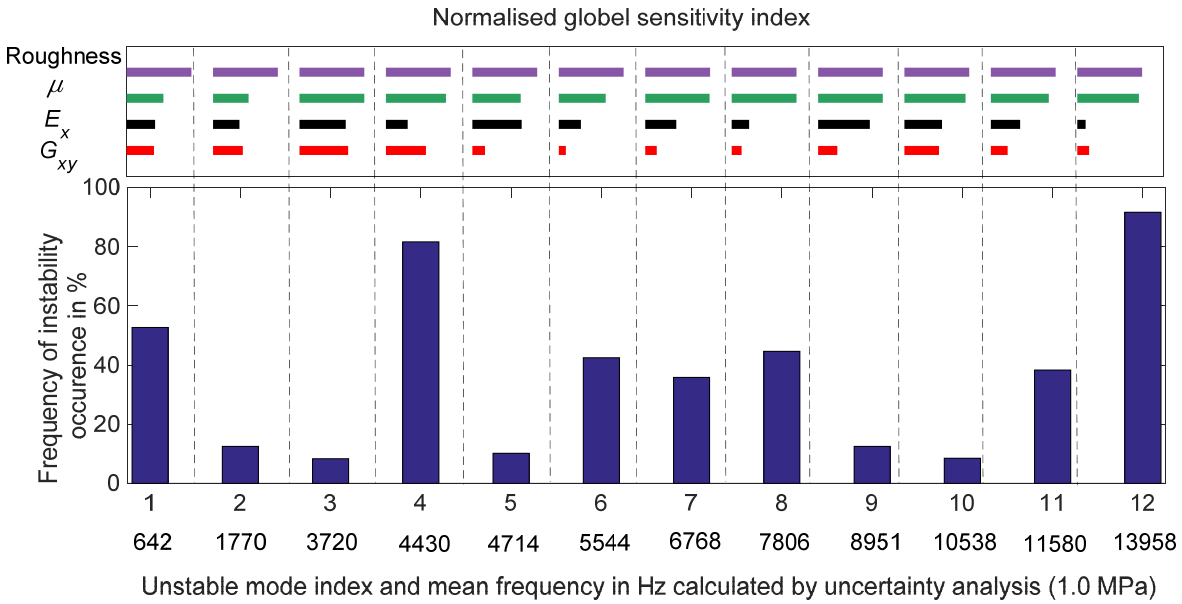

Figure 8. Frequency of occurrence and global sensitivity index for predicted unstable modes (Pressure = 1.0 MPa)

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