

The Role of Nonlinearity and Uncertainty in Assessing Disc Brake Squeal Propensity

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ABSTRACT

Despite significant progress made in the past 20 years in discovering some of the mechanisms of brake squeal, it remains difficult to predict the underlying friction-induced instabilities reliably. Most numerical analyses are based on linear deterministic analyses of structural vibrations such as the complex eigenvalue analysis (CEA). However, nonlinear multi-scale processes govern friction contact with high sensitivities to operating and/or environmental conditions. In addition, uncertainties in the material properties and boundary conditions such as contact and friction laws are rarely considered. Hence, it is quite common to underpredict or overpredict the number of instabilities and extensive brake noise dynamometer tests are still required in industry to ensure acceptable brake noise performance. In this paper, simplified finite element brake models are used to illustrate the role of nonlinearity in brake squeal. By using nonlinear time series analyses, forced response calculations, dissipated friction work and acoustic radiations, unstable pad modes have been found to be responsible for the instantaneous mode squeal which, although observed experimentally, cannot be predicted with the traditional linear CEA. By considering coupled spring-mass-damper oscillators representing a pad on a sliding plate, the role of uncertainties of contact stiffness and friction laws in brake squeal is examined using probabilities of the positive real part of complex eigenvalues and positive friction work. The implications of nonlinearity and uncertainty for brake squeal predictions are discussed. Suggestions on how the new insights gained into nonlinearities and uncertainties can be exploited for practical brake squeal analyses in industry are proposed.

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INTRODUCTION

Brake squeal, defined as an audible noise above 1 kHz and caused by friction-induced vibrations, is a major concern to automotive manufacturers [1]. Brake squeal is caused by various, potentially interacting mechanisms such as mode coupling, stick-slip, sprag-slip, instantaneous mode squeal, decreasing friction coefficient with increasing relative velocity, rate dependency and hammering [1, 2, 3, 4, 5]. The standard industrial tool for its prediction is based on the linear complex eigenvalue analysis (CEA) [1,2]. However, the CEA can over-predict [6] or under-predict [4, 5, 6] the number of unstable vibration modes, hence squeal. Even the magnitude of the negative damping predicted by the CEA cannot be taken as an indication of squeal propensity [1]. This could be due to [2] the effect of nonlinearity not being accounted for as CEA is a linear method; deficiency of the friction contact model; simplistic and/or linear modelling in numerical simulations are widespread [2,9]; full brake models are often not properly experimentally validated and the acoustic radiation is rarely calculated. The consideration of only the onset of instability and using merely structural vibrations in the CEA is insufficient as not every and not only a limit cycle might lead to squeal. Uncertainties in and strong sensitivities to material properties, friction characteristics, contact interface, environmental and loading conditions [10] further affect the reliability of instability predictions.

Simulations incorporating nonlinearity are very much in their infancy stage [6,7]. Nonlinear stability and/or transient time domain analyses (TDA) have been shown to be more predictive than CEA [7,8,11]. Yet, their mathematical complexity and large computational costs are prohibitive factors for their application to a full brake system.

It has been shown by Butlin and Woodhouse [12] that the predicted unstable modes may change drastically with different friction models and that better agreement between numerical predictions and measurements is obtained if measured friction data are used. In contrast, Feeny and Moon [13] indicated experimentally and numerically that the attractor changes only slightly for different friction laws. How the different instability mechanisms modeled by different friction laws influence the attractor and the stability margin is unknown. Further, although modeling the contact between the pad and the rotor is important, little is known about the exact contact conditions [12] and options for modeling and influencing the contact in most commercial software are very limited. Pin-on-disc systems studied by Butlin and Woodhouse [14] or Giannini et al. [15] show more repeatable results than a real brake system [16,17]. However, pin-on-disc systems are experimental 'minimal models' [16], which do not reflect accurately enough the behavior of a full brake system with respect to the contact.

A brake system is complex and inherently nonlinear owing to friction, material properties such as the lining material or the shim, and boundary conditions such as e.g. joints and abutments [17]. By analyzing the microphone signal of a full brake system, the development of brake squeal has been shown to follow the Ruelle-Takens route to chaos [18]. By applying a nonlinear TDA to a simple pad-on-disc brake system, the intermittency route to chaos type II, which is related to subcritical Andronov-Hopf bifurcation, has been observed [8]. Quasi-periodic motion such as displayed by toroidal attractors (multi-instabilities [19]) seems to be an underestimated feature in brake systems [18,20,21,22].

Hence, more research has been aimed at applying statistical; sensitivity and uncertainty analyses to linear brake squeal predictions [20,22,23,24]. Néchak et al. [25] and Sarrouy et al. [26] were among the first to apply polynomial chaos expansions to CEA for brake squeal propensity predictions. By using an experimentally updated brake system, Tison et al. showed that squeal predictions could be improved using uncertainty calculations with a stochastic field approach applied to the contact condition between the pads and the rotor [27].

In this paper, the importance of pad modes and nonlinearity is illustrated by finite element simulations of a simplified brake in the form of a pad-on-disc model. The uncertainty in the contact area and the friction law used for modelling the contact interface is then examined using a stochastic approach for an analytical model of 3x3 coupled friction oscillators. The implications of the findings are discussed with regard to their practical potential for predictions of squeal propensity of a full brake system in industry.

FINITE ELEMENT STUDY USING A SIMPLE MODEL OF A BRAKE SYSTEM

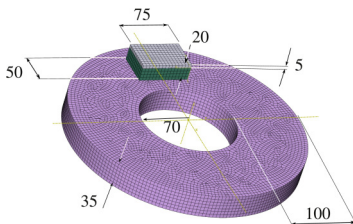


Figure 1. Pad-on-disc numerical model, all measures in mm [2].

The Finite element model

A finite element model of a pad-on-disc system is setup in ABAQUS 6.8-4 [2]. The pad's material properties and mesh parameters are listed in Table 1.

Table 1. Parameters used in the pad-on-disc model

Element type	Linear hexahedral, C8DI incompatible nodes elements
Number of elements	30443
Young's modulus (pad/disc)	210 GPA / 110 GPA
Poisson's ratio (pad/disc)	0.30
Density (pad/disc)	7200 kg/m ³ / 7800 kg/m ³

A mesh study according to Oberst et al. [2] is conducted to provide mesh independent stability predictions. Finite sliding contact conditions and a constant friction coefficient are applied. The simulations allow for the possibility of contact separation, which together with contact stiffness are described by nonlinearity within ABAQUS. For autonomous systems, after the application of both a nonlinear stress analysis and normal mode extraction [28] the system matrix is composed of asymmetric stiffness and/or negative damping terms owing to friction or follower forces etc. as shown in Eq. (1)

$$(\lambda^2 \mathbf{M}^* + \lambda \mathbf{C}^* + \mathbf{K}^*) \mathbf{y} = 0. \quad (1)$$

In Eq. (1) $\mathbf{M}^*$, $\mathbf{C}^*$, and $\mathbf{K}^*$ constitute the mass, damping and stiffness matrices respectively in their diagonal form e.g. $\mathbf{M}^* = (y_1, y_2, \dots, y_N)^T \mathbf{M}(y_1, y_2, \dots, y_N)$. If the real part of the complex eigenvalue λ is positive, the system's damping ratio becomes negative, indicating that energy is *provided* by friction via self-excited vibrations.

RESULTS OF THE FINITE ELEMENT STUDY

By conducting frequency domain analyses using the complex eigenvalue analysis and forced response calculations, a contact liftoff between the pad and the rotor has been identified, Figure 2. A penetration results in increased tangential and normal contact stiffness terms until the impenetrability condition is not violated anymore [2,28,29].

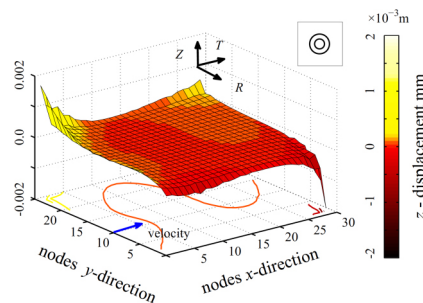


Figure 2. Contact separation in the numerical pad-on-disc model [2].

While the same instabilities as the full contact are observed for contact separation but at higher friction coefficient values, the radiated acoustic power is amplified by contact separation similar to a horn effect [2], Figure 3. The results show that a gap between the disc and the pad can amplify the sound pressure level and that local acoustic effects at the contact patch become significant. Compared to the small sliding definition which only allows small displacements and rotations and does not allow a lift-off, the unstable modes predicted for finite sliding show up at higher frequencies and higher friction coefficient values but remain otherwise of the same kind. Owing to the changing frictional contact, large frequency shifts of pad modes can be observed. Pad modes have been identified to couple with out-of-plane disc modes to form an unstable pair within a certain frequency range, which is then detected, to be unstable by the CEA. However, at some frequencies the pad-modes are predicted by the CEA to be robustly stable [28].

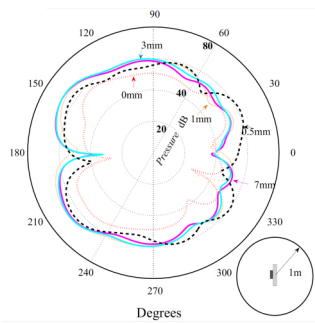


Figure 3. Out-of-plane sound pressure level (SPL) of an unstable $n=4$ disc mode (four nodal diameters) as a function of gap between the pad and the disc; with increasing gap the SPL increases significantly by up to 25 dB [2].

A forced response study indicates that the negative dissipated (i.e. provided) energy at resonances of in-plane pad modes increase cubically with the friction coefficient and pressure [2,29], Figure 4. However, these pad modes are predicted by the CEA to be stable in the structurally un-damped (marginally stable) situation. A transient nonlinear time domain analysis (TDA) with only self-excitation [8] shows that pad-modes are responsible in triggering secondary instabilities, after a diminishingly small limit cycle has been formed. Here, out-of-plane motion of the pad has been observed, which is similar in character to hammering the disc. This impacting vibration together with friction is responsible for energy being provided to the system and the development of unstable in-plane pad-modes. A detailed nonlinear analysis of the numerically generated time series of the motion of the pad shows that the system destabilizes over the intermittency route to chaos II; subcritical Andronov-Hopf-bifurcations are the underlying root cause here [8]. Comparing results of an analytical friction oscillator of an unstable pad sliding on a belt with a cubic stiffness model and its linearized contact stiffness confirms that subcritical Andronov-Hopf-bifurcations are not detected by the linear CEA [7].

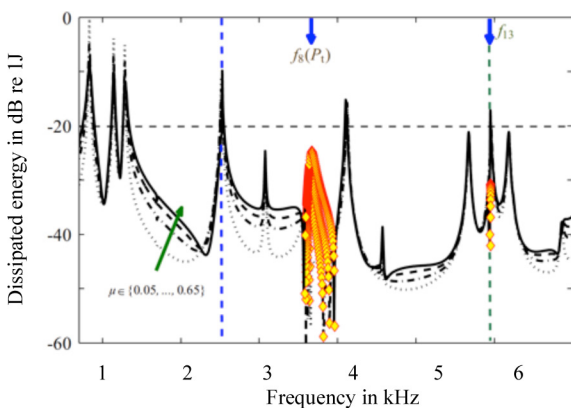


Figure 4. Negative dissipated energy; markers indicate energy provision [29].

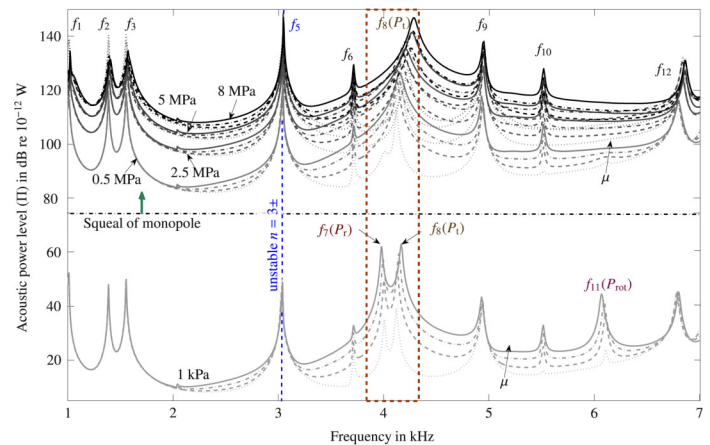


Figure 5. Acoustic power of the pad-on-disc system as a function of pressure and friction coefficient: the tangential pad mode (resonance f_8) changes nonlinearly in amplitude and frequency [30].

Results of the pad-on-disc study indicate that the finite sliding contact and the option for separation are influential for instability predictions [2,8,29]. By applying the acoustic boundary element method to the results of the forced vibration response calculations, the unstable pad-modes, even though acting mostly in-plane, have been shown to be acoustically relevant, Figure 5 [30].

Hence, in order to study the behavior of pad-modes in more detail especially the influence of frictional contact i.e. its friction law formulation, the contact and its partial separation on dynamic instabilities, an analytical model of a pad sliding on a plate is examined.

Analytical Model of a Plate Sliding on a Large Mass

The analytical model in Figure 6 [31] consists of 3x3-coupled linearized friction oscillators as developed by Hoffmann et al. [32]. For the uncertainty analysis with randomized friction parameters, Figure 6 (a) shows the analytical model of 3x3 oscillators with (a) a symmetric springs-/dashpots- arrangement and (b) an asymmetric arrangement in order to also examine the influence of symmetry on the results. The oscillators with mass m_i ($i = 1, \dots, 9$) are pre-loaded by the pressure P_i in point contact (modeled by the springs k_i^z) with a rigid plate sliding with a constant translational speed v in the x -direction and held by springs (k_i^j , $j \in \{x, y, z\}$) and viscous dashpots c_p^j .

The friction force acting on each mass in the x -direction is modeled either by the Amonton-Coulomb, the velocity-dependent [33, 34, 35], or the LuGre friction model [35,36]. As the plate does not slide in the y -direction, the relative velocity between oscillators and the plate is then the absolute velocity of the oscillator, which is assumed to be very low due to small deflection theory. Thus there is no need to model the friction in the y -direction by the velocity-dependent or LuGre friction model.