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TRANSIENT SQUEAL ANALYSIS OF A NON STEADY STATE MANEUVER

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ABSTRACT

Brake squeal is one of the main NVH challenges in the brake development of passenger cars. The conflict of goals in the development process and the late testability lead to the need of a deeper basic understanding of the squeal phenomenon and definition of design rules. On the other hand, brake squeal is still a very interesting field of research also for the universities because of its combination of different fundamentals, such as friction and stability behaviour of systems with local nonlinearities.

Major nonlinearities of the brake system are the joints, especially the contact area formed by the oscillating brake pad and the caliper. The state-of-the-art calculation method, which still is the 'complex eigenvalue analysis', linearizes these joints, hence neglecting its nonlinear influence in the stability analysis. Vehicle and bench experiments show that special driving manoeuvres like parking, where the brake pad often leaves the steady state, are likely causing brake squeal. The system in these manoeuvres sometimes behaves opposed to the linearized stability analysis, indicating a limit cycle beyond the Hopf Point. Therefore, these states must be investigated more closely.

This paper investigates the nonlinear influence of the pad caliper joint in a fixed brake caliper, also called abutment. Bench tests with pressure foils at the abutment of the brake caliper and mode shape analysis were done and a simple FE model for a transient simulation is proposed. It is shown that the joint activity varies with driving manoeuvres, leading to different stability behaviours and limit cycle amplitudes.

A new load case is proposed to excite modes artificially, which lead to less simulation time in the transient domain. The system can reach the zone of attraction at a steady state and does not need to be simulated with a moving pad.

INTRODUCTION AND PREVIOUS WORK

The NVH requirements for passenger cars are high and are probably going to increase with the ongoing electrification of the power train. In addition, the customer can nowadays also be standing outside of the vehicle, e.g. parking his car remotely. Brake squeal is still one of the main NVH phenomenon's at the vehicle brake, challenging both, the bench and vehicle testing and the simulation departments. For the bench tests many test procedures and sequences have been developed to test the different states of the brake and indices have been created to evaluate the occurrences, amplitudes and frequency of the squeal [2]. Also vehicle testers develop a 'feeling' for the brake, which lead to critical working points, that often create squeals even though the brake is silent in the rest of the test drive. One of this manoeuvres is parking, which was linearly analysed in [8] and this analysis shall be continued with nonlinear methods in this work.

From the simulation side large effort is carried out to tackle the squeal problem in general. Despite its known limits, the linear method of calculating the 'complex eigenvalues' of the system is still the state of the art. Because of its fast calculation time many working points can be investigated leading to a good overview of the robustness of the brake [6]. Unfortunately, the brake system is highly nonlinear [4, 5, 7, 9] and linear stability analysis can lead to wrong results. Not only the amplitudes of the limit cycles cannot be calculated but also subcritical flutter can occur, meaning that with the right initial conditions a stable limit cycle beyond the linear stability borders can be found. Oberst [7] showed in his report that if one solely considers the nonlinearity of the pad and disc contact subcritical flutter can already occur. Kruse et al. [5] showed in a minimal model, that the joints e.g. the contact between brake pad and caliper, play a significant role and also can lead to subcritical flutter of the system. In the following investigation, the change in direction of the brake disc velocity while braking is examined. This will be called parking. The effects of cornering is not part of this work.

PRESSURE FOIL MEASUREMENTS

The main mechanism while parking is the movement of the brake pad in the caliper. To qualitatively understand how the brake pad gets in contact with the caliper after a change of direction of the car, pressure foil measurements on a real brake system are done between the brake pad and the caliper. The sampling rate of the pressure foil is 100 Hz, hence too low to see the oscillation of the squealing. But as seen in figure 1, the change in the contact surfaces of the abutment area can be observed. Figure 1 shows two different low pressure brake applications exactly after a change of direction of the disc. The same pad/disc configuration at different pressure levels and temperatures are used. The brake pads touch the abutment at the beginning of the parking brake manoeuvre and then the contact area increases over time until a steady state is reached. The increase in contact area is not the same for every working point, due to e.g. the application point of the friction force and is also dependant on production tolerances, especially at the edge of the back plate. The inner and outer pad behave differently, which can be explained with the deformation of the one side mounted caliper.
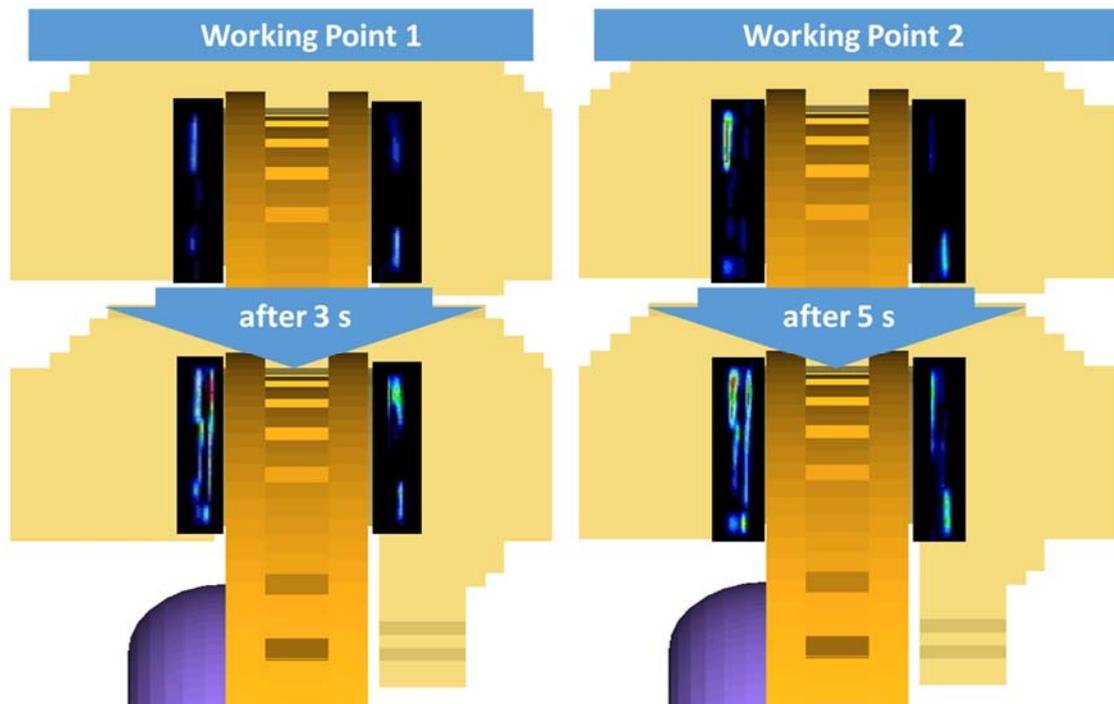
Figure 1: Pressure foil measurements at two working points with the same pad at the beginning of the brake application and a certain time interval later. The foil results are embedded into mesh data for better clarity

More measurements are done with different working points and different pads. Because of the mentioned variability and the influences of the production tolerances, the pressure foil measurements can only lead to a better understanding of the parking process, rather than to validate the simplified FE simulation. The measurements motivate the simulation of a variable abutment surface and its influence onto the squeal probability. This was already done in [8]. The abutment contacts between inner and outer pad could be different, which can also be included in a linear stability analysis.

Furthermore, the measurements of figure 1 motivate a nonlinear analysis of the process. As shown in [5] the nonlinear joint at the abutment can lead to a subcritical flutter. Assuming that the change in the contact states shown in figure 1 can be simplified as a change in the axial stiffness of the minimal model in [5], during a parking manoeuvre the system has a higher probability to hit the critical area between a Fold Point and a Hopf Point. In this area, stable limit cycles can occur, but the system does not need to show an instability in the linearization around the equilibrium. In addition the 'hit' of the pad from leading to trailing edge, represent a large change in the initial conditions, which also increases to probability to reach the zone of attraction of a stable limit cycle.

LINEAR AND NONLINEAR STABILITY ANALYSIS

To analyse the nonlinear stability behaviour of the FE brake model introduced in [8], nonlinear transient simulations are carried out. The linear stability is shown in figure 2. The stability border for the 2 kHz instability is either around $\mu_{crit} \approx 0.8$ for the axially free abutment or with a fixed abutment around $\mu_{crit} \approx 1.0$. The same parameters will be used for the nonlinear transient simulation, except for the friction value μ between disc and pad, which will be $\mu = 0.4$, so no stable limit cycle can be anticipated, based on the linear stability analysis.
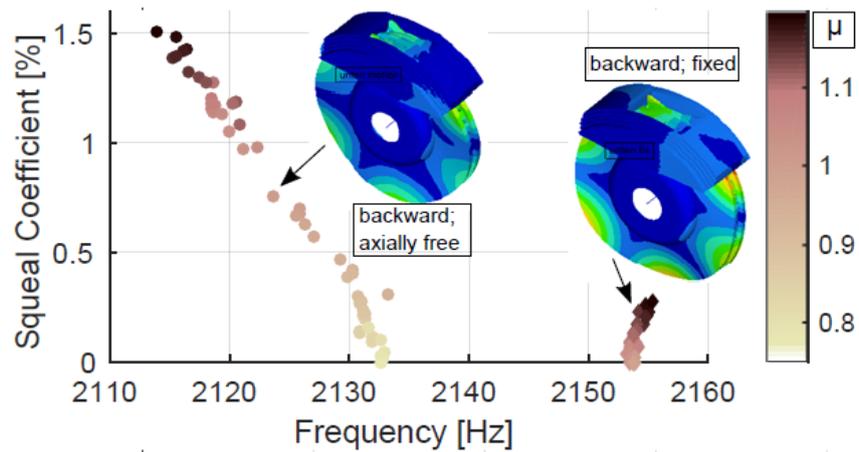
Figure 2: Linear stability analysis of working points with increasing μ. With an axially free abutment the 2 kHz instability appears at around $\mu_{crit} \approx 0.8$ and with fixed abutment around $\mu_{crit} \approx 1:0$

Model discussion: a) Nonlinearities in the system

There are two main nonlinearities in the model: the contact between disc and pad and the contact between pad and caliper, the abutment. As shown in [7] the pad disc contact solely can be enough to introduce subcritical flutter. However, for this investigation, the movement of the pad between leading and trailing edge has to be simulated, and therefore contacts at the edges of the pad are necessary. Another nonlinearity could be the piston to caliper contact or piston to back plate contacts. In the presented study these contacts are neglected, but they should be investigated in future work.
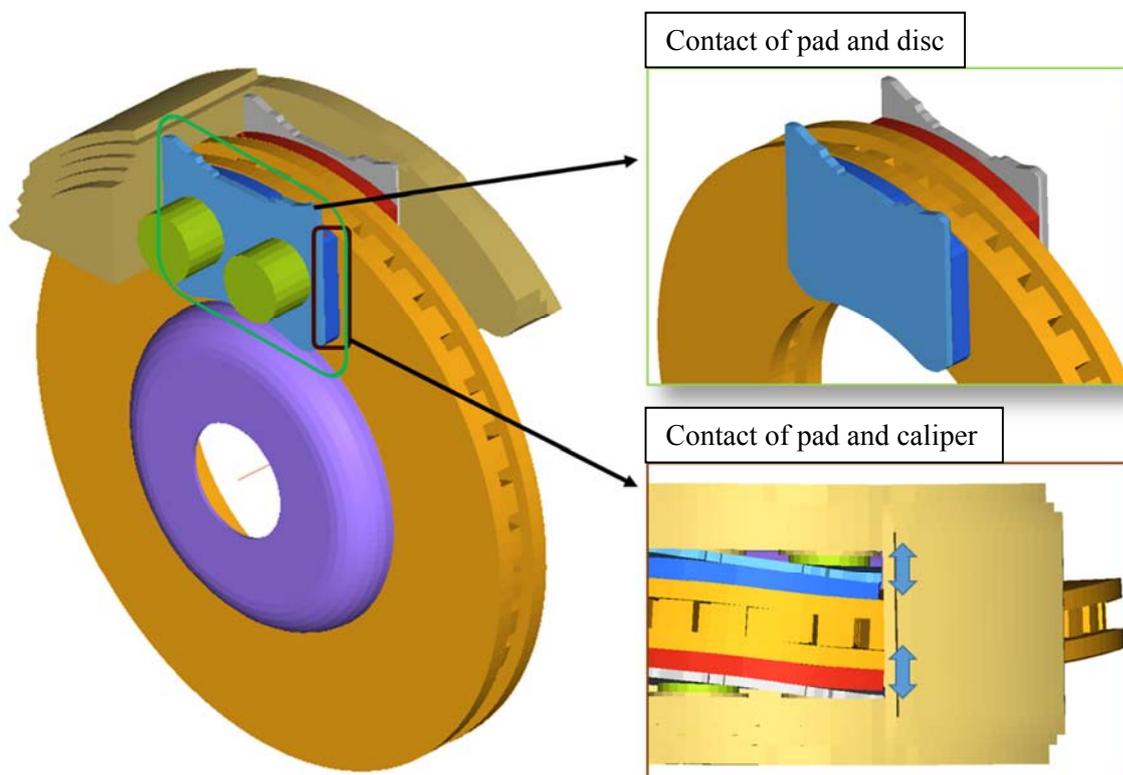

Figure 3: Model overview and zoom into the two nonlinearities of the transient model

Model discussion: b) Damping and numerical damping

As discussed by many authors, the brake squeal phenomena is very sensitive to damping [1, 4]. But despite the clear need, the mechanisms and implementation of damping is still not fully understood. Hence, in this study no material damping is used. Only the damping resolving from the friction force and the numerical damping of the dynamics of the solver. The parameters of the solver will then be tuned, so that the results are matching the bench tests. This again shows the difficulty in predicting the squeal probability of an unknown brake system without available hardware at this stage of the development.

For this study, the implicit Hilber-Hughes-Taylor α-Method (HHT) is used in Abaqus. The solver dynamics is dependent on the numerical parameter α, the time step and the angular frequency under investigation [3]. Figure 4 shows the results of three different α values. For α = 0 (a) many frequencies occur, which is unrealistic. With small numerical damping (b) a 9 kHz instability arises, which cannot be seen in the higher damped simulation (c).
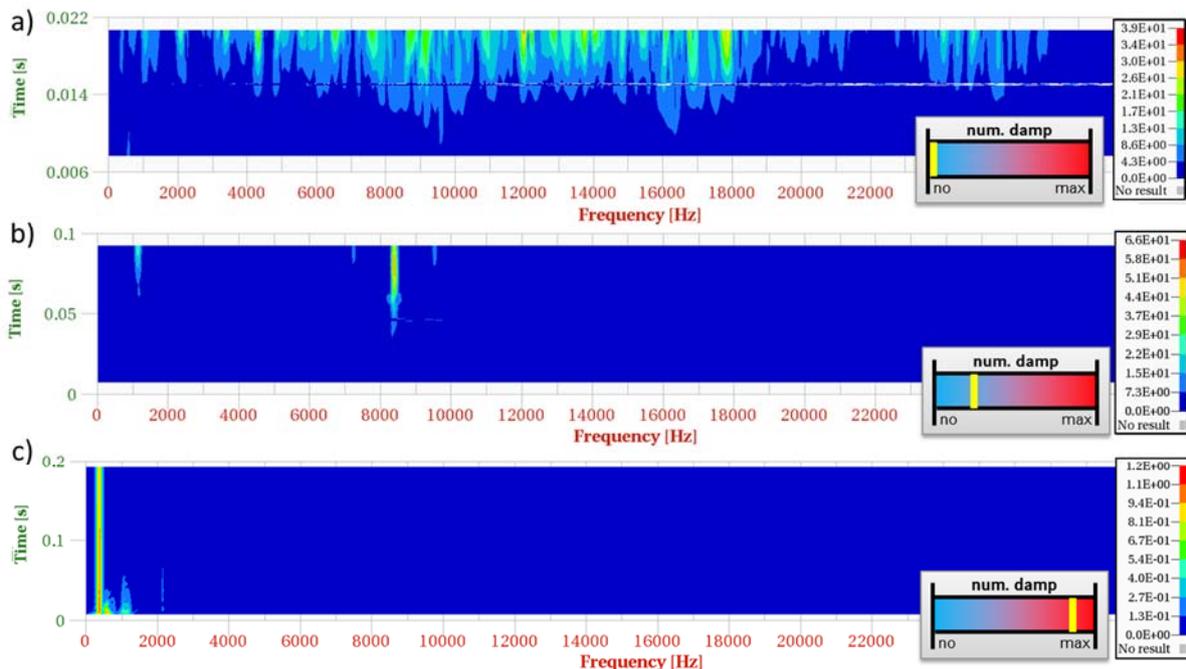

Figure 4: Waterfall diagram of the velocity of a disc node for the transient results with different numerical damping. a) α = 0; b) α = -0.1 and c) α = -0.41421 (Abaqus default value for contact problems). Colour scale of the plots is different.

Because of the described unknown system damping, one can a priori not judge, if the lesser damped result is representing a real squeal. At the bench tests, with the underlying real brake system, a 2 kHz and a 9 kHz squeal could be observed. Hence, the lower damped simulation could represent a real squeal scenario, even though this study focuses on the low frequency squeal. If not mentioned otherwise the 'high' numerical damping will be used in the following.

Nonlinear transient results

The goal of the nonlinear transient simulation is to compare it with the linearized results and to simulate different pad movements and their influence to the resulting limit cycle. As shown

in figure 2 the critical μ of the used working point is $\mu_{crit} \approx 0,8$. Hence, the system should not show any stable limit cycles at this frequency range, assuming the linearization state is correct. In Figure 5 (a) and (b) waterfall diagrams of the velocity of a disc node is shown. The transient simulation starts with a perfectly aligned pad for the forward and backward direction, so there is no pad movement necessary to establish contact with the abutment. Initially one can observe a 2 kHz vibration, but it is damped out over time, which correlates to the linear analysis.
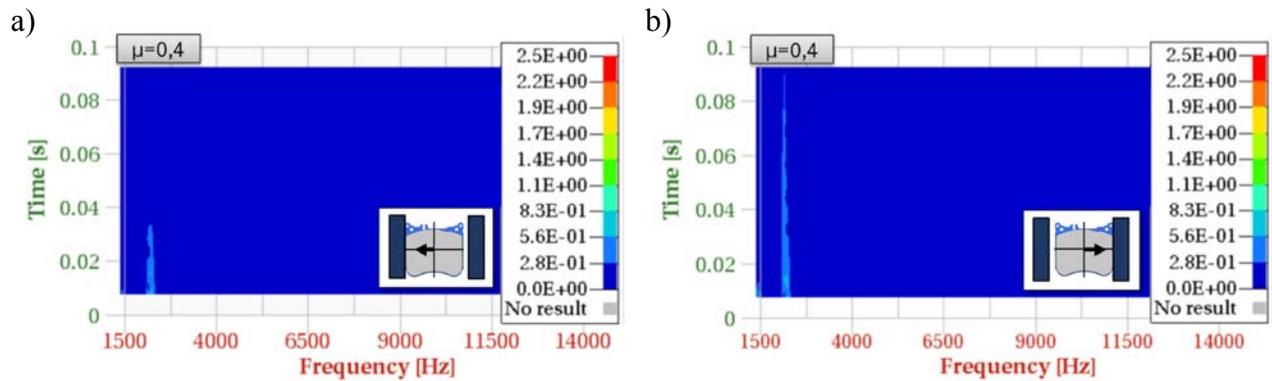
a)

b)

Figure 5: Waterfall diagram of the nonlinear transient analysis starting with a perfectly aligned pad. Displayed is the velocity of a disc node.

Figure 6 shows the waterfall diagram of the same node of the mesh, but with initially a gap between pad and caliper. The gap leads to a hit of the pad against the caliper causing the critical initial conditions for a stable 2 kHz limit cycle. With no tested linearization state this instability could be found. Hence, the system here has a subcritical Hopf bifurcation. Based on the experiences from bench and vehicle test, this could highlight the criticality of the parking manoeuvre. Not only the brake experiences different contact states, but the manoeuvre itself also leads to critical initial conditions for a subcritical flutter.
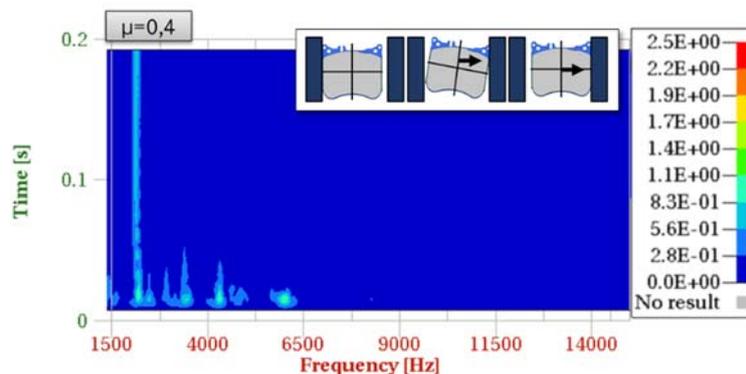
Figure 6: Waterfall diagram of the nonlinear transient analysis starting with a gap between pad and caliper. Displayed is the velocity of a disc node.

Comparisons of the linearized mode shapes with the transient results

To get a deeper understanding of the differences of the linear and nonlinear results the mode shape of the linearized system and the operating deflection shape (ODS) of the stable limit cycle is compared. In the macroscopic comparison in Figure 7 the accelerations look nearly the same of the linearized and nonlinear transient results. The main differences obviously lie in the local nonlinearities.
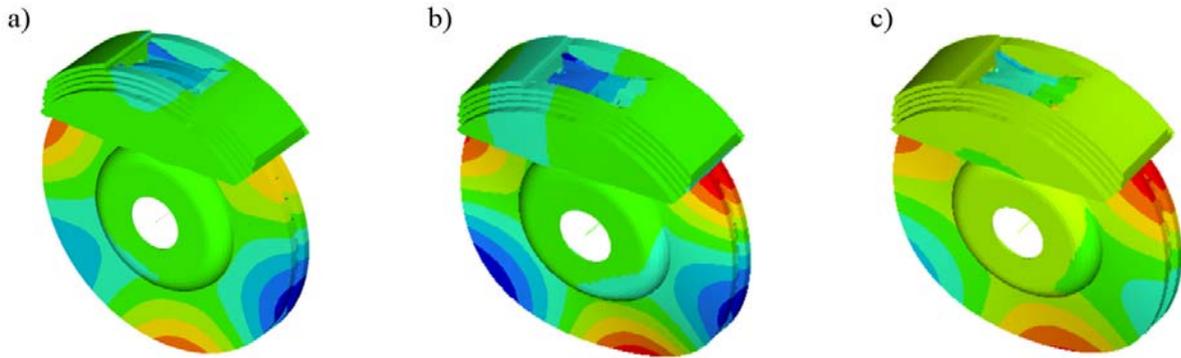
Figure 7: Nonlinear transient momentum state at t=0.199 s showing only the axial acceleration (a). Mode shapes of two different linearizations at a close frequency (b) and (c) for comparison.

The resulting limit cycle of the transient simulation in this specific working point is very dominant, hence the results are directly comparable to the linear mode shapes. Figure 8 shows the acceleration of the transient result at 0.199 s. The deformation scale is increased to see the relative movement at the abutment. Even though the main motion of the trailing side of the pad in this case is a rotation around the contact line at the abutment in figure 8, there is also a small relative movement visible. Hence, in one cycle of the transient oscillation this joint behaves nonlinearly and switches from sliding to sticking. Furthermore, the contacts between disc and pad open and close in certain areas, which introduces another nonlinearity in one cycle. These results again highlight the importance of a nonlinear analysis to investigate brake squeal and to closely look at the joint activity.

The often used Modal Assurance Criterion (MAC), to compare two mode shapes or ODS, can lead to false security. E.g. comparing the two linear mode shapes in figure 7 the MAC is ≈0.96, which is a very high value. This has its origin in the similar macroscopic deformation at that frequency. But as seen in the transient results, small local nonlinearities can change the system behaviour significantly.
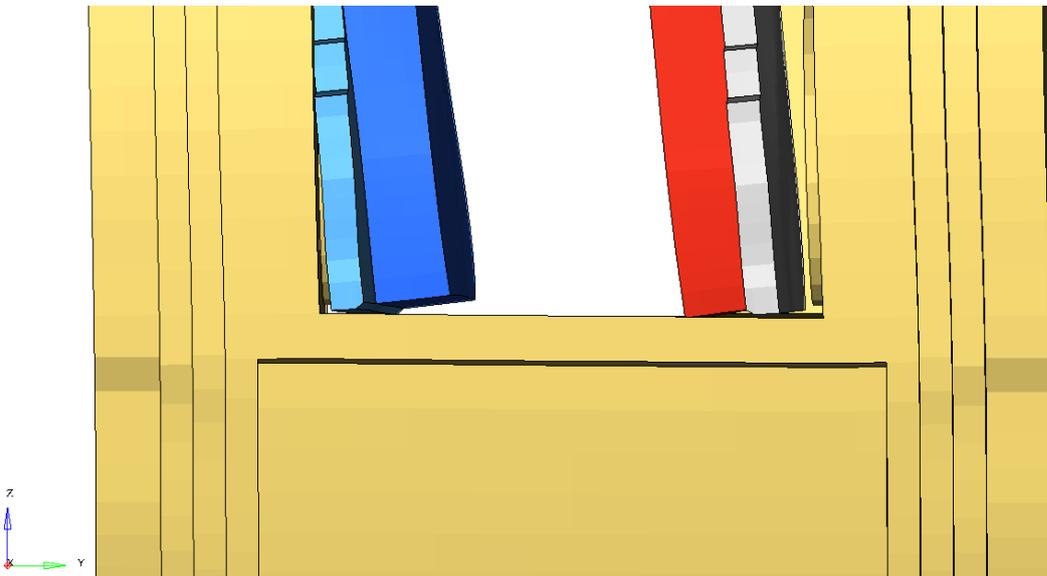

Figure 8: Scaled acceleration of the nonlinear transient result at t=0.199 s.

Figure 9 shows the comparison of the displacements of two caliper nodes at the abutment and their corresponding back plate nodes. At the outer pad, sticking can be observed. The inner pad shows a more dominant 2 kHz oscillation. Thus, sticking phases are harder to detect. These exemplary results show the high joint activity at the abutment. Also differences

between inner and outer pad can be seen, which corresponds to bench results at low pressure.
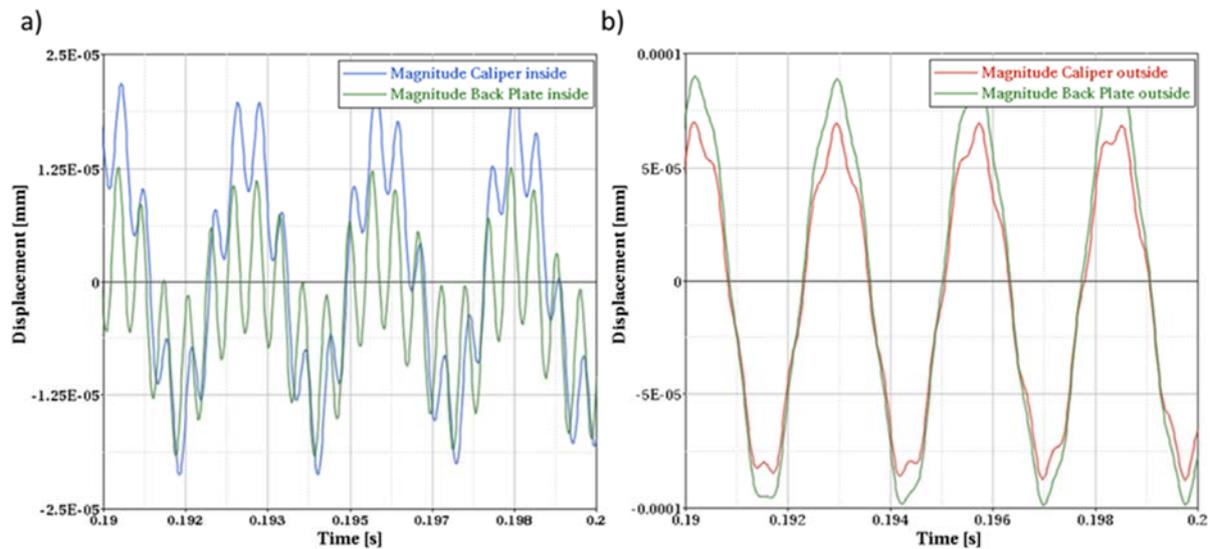

Figure 9: Comparison of the displacements of two caliper nodes at the abutment and their corresponding back plate nodes. (a) shows exemplary nodes on the inner side of the brake; (b) shows nodes of the outer side; the back plate is mostly sliding at the abutment, but there are also areas where the back plate sticks at the caliper.

Excitations to reach the observed limit cycle

The presented system has at least one stable limit cycle branch beyond the Hopf Point as seen in the results of figure 6. Motivated from experience in the bench and car tests the movement of the pad and its hit against the abutment provoked the system to jump from the 'quiet' solution into a stable 2 kHz limit cycle. In the following, more initial conditions are tested to evaluate the critical initial conditions and to find excitation methods, which lead to less simulation time.

Figure 10 shows an overview of different initial positions of the pad and the resulting time simulations. It seems that there are more than one stable limit cycle branches at the same frequency. Different amplitudes can be observed, especially when the pad is initially skew.
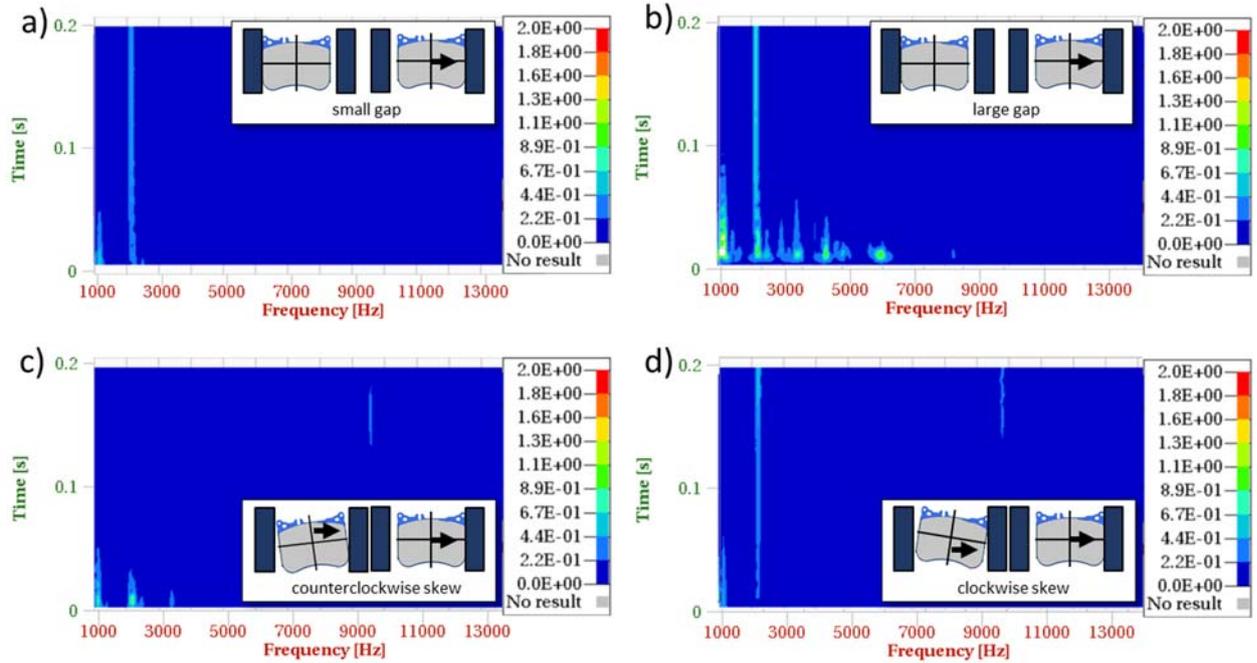
Figure 10: Nonlinear transient results with different initial conditions. The resulting amplitude of the limit cycle in (a), (b) and (d) is different. In (c) no limit cycle appears.

When the pad is perfectly aligned, an artificial excitation to provoke the 2 kHz limit cycle could not be found and must be investigated in future work.

CONCLUSIONS AND OUTLOOK

This work investigates the nonlinear stability behaviour of a fixed caliper brake system with the load case 'the change in direction of the brake disc velocity while braking'. The nonlinear transient simulation is motivated by pressure foil measurements at the contact area of the brake pad and the caliper. The measurements show, that the contact area changes widely due to a parking manoeuvre of the vehicle. This validates on one hand the assumptions made in a previous study, in which the linearization points of a complex eigenvalue analysis take the variation of the abutment area into account.
On the other hand, the measurements raise the question, how good the brake system can be simulated with a linear analysis, especially under such manoeuvres.
Subsequently, nonlinear transient simulations are done to investigate the resulting limit cycles of the brake. It is found that at the working point under investigation, the brake system is 'quiet' if the brake pad initially is placed perfectly aligned to the caliper. If the brake pad has initially a gap and hits the caliper, a limit cycle can be observed. With no tested linearization state this instability could be found at the same working point. Like other authors reported before, this indicates, that the nonlinearities of the brake pad can lead to subcritical flutter. Finally, different excitation and initial states are investigated to reduce the needed simulation time of the nonlinear analysis. It seems that not the axial movement of the brake pad is provoking the limit cycle, but the change in the normal force at the abutment. This could also explain, why parking is such a critical test manoeuvre, because of its large normal force variation.

In future work more effects of the real parking manoeuvre will be included. Additionally, the third nonlinearity of the brake pad, the contact to the piston will be investigated. It would be also very interesting to search for design parameters, which can influence the system in such

way, that the subcritical flutter is transformed into a super critical flutter. The stability borders found in a linear analysis would then be valid again.

REFERNECES

(1)   Fritz, G.; Sinou, J.-J.; Duffal, J.M.; Jezequel, L.: "Effects of damping on brake squeal coalescence patterns - application on a finite element model", Mechanics Research Communications, Elsevier, 34 (2), pp.181-190, 2007

(2)   Häsler, K.; Bender, A.: "GLONORE - An Approach for a Total Evaluation of Brake Noise", Dresden, EuroBrake2012

(3)   Hilber, H. M. ; Hughes, T. J. R. ; Taylor, R. L.: "Improved numerical dissipation for time integration algorithms in structural dynamics", In: Earthquake Engineering and Structural Dynamics, 1977

(4)   Hochlenert, D.: "Nonlinear stability analysis of a disk brake model", Nonlinear Dyn, 58(1):63–73, 2009

(5)   Kruse, S.; Tiedemann, M.; Zeumer B.; Reuss, P.; Hetzler, H.; Hoffmann, N.: "The Influence of Joints on Friction Induced Vibration in Brake Squeal", Journal of of Sound and Vibration 340, 239-252, 2015

(6)   Nunes, R.F.: "Robustness Analysis of Brake Systems: Build the Bridge Between Simulation and Quality Control Tests", Dresden, EuroBrake2015

(7)   Oberst, S.; Lai, J. C. S.: "Nonlinear Transient and Chaotic Interactions in Disc Brake Squeal", Journal of Sound and Vibration 342, 272-289, 2015

(8)   Stump, O.; Nunes, R.F.; Seemann, W.: "Brake Squeal: Influence of the Contact Linearization at the Pad Abutment", Milano, EuroBrake2016

(9)   Tiedemann, M.; Dethlefs, A.; Kruse, S.; Uhlmann, E.; Hoffmann, N.: "Design of Contact Interfaces to Reduce Noise in Friction Brakes", Dresden, EuroBrake2015