Excellent Brake Noise Comfort by Simulation – Advanced Methods to Create Stability Maps

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Abstract: Numerical methods for brake squeal analysis are widely accepted in industry. The approach to use complex eigenvalue analysis is successful to forecast the appearance of squeal noise. Using simulations in an early design stage helps to reduce time to market, to save costs and to improve the physical behaviour and robustness of the system. This paper will show advanced methods that use a specific strategy to reduce computation time. The industrial example of a PORSCHE brake system is analysed by this simulation strategy. The process starts with the validation of the disc by comparison of mode shapes from test rig with FEM results and subsequent geometrical optimization. The next step is model updating of the transversely isotropic material of the pad. All remaining parameters, like brake pressure, rotational speed of the wheel, friction coefficient between disc and pad, and Young's modulus of disc are samples in one single PERMAS run. The resulting stability map expands the knowledge about the NVH behaviour and robustness of the brake system for a wide range of parameter combinations. The additional information from the stability map strongly supports the development of a robustly quiet brake system.

Keywords: Brake squealing, complex eigenvalues, sampling, stability map, high performance

1. Use of Simulation in Industry

Simulation methods for brake squeal analysis are widely used in industry. The complex eigenvalue analysis (CEA) as implemented in PERMAS with special extensions for improved accuracy and high performance is successfully applied by PORSCHE AG to predict squeal noise (see Fig. 1). Using simulations in an early design stage helps to reduce time to market, to save costs and to improve the acoustic behavior and robustness of the brake system.

The first step in CEA is a static contact analysis, which takes into account friction, brake pressure, vehicle speed and moment of inertia due to rotation as well as any pretension, which is applied in the brake system by bolts or other means. The resulting contact status is used to linearize the nonlinear contact problem in order to proceed with modal dynamic methods.

The second step is the calculation of real eigenvalues. Additional stiffness effects are taken into account due to rotation and contact.

The third step is the calculation of complex eigenvalues. Here, the gyroscopic effect and additional damping due to contact is included in addition.

The fourth step is sampling, which will be explained in the subsequent sections of this paper.

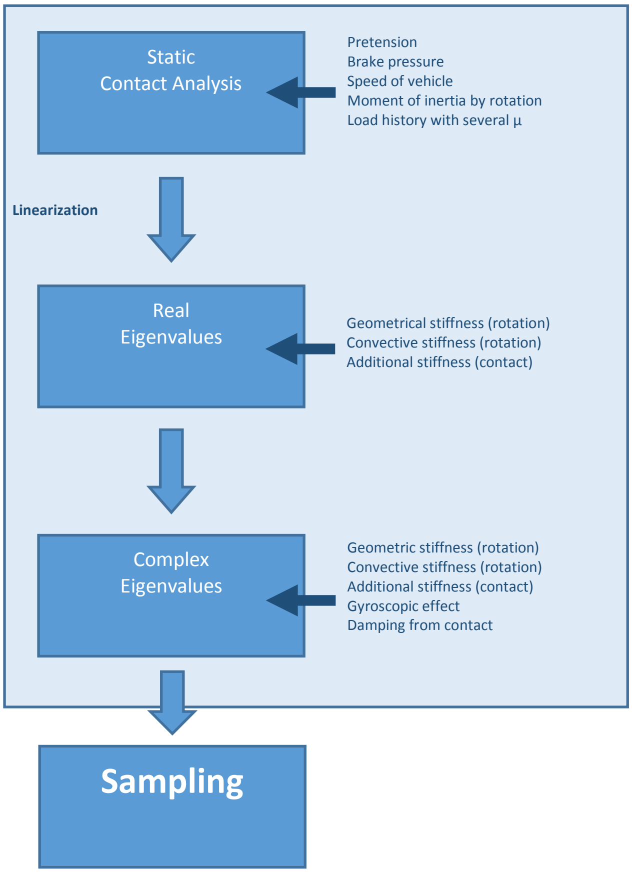

Figure 1: CEA with Sampling

2. State of the Art

Today's state of the art in brake squeal analysis is best characterized by performing one calculation run for one operational point with one set of parameters [1, 2]. Some of these brake system parameters could be determined in advance, e.g. by measurement, and will show only little change during brake operation. However, the brake system has an infinite number of operating points and many parameters with a wide spectrum of possible values. Due to wear of pads and disc some parameters change during lifetime of the brake system. Typical parameters are brake pressure, friction coefficient between disc and pad, and vehicle speed. Another group of parameters depends on conditions during production. A typical example for this group are material parameters.

3. Sampling by Simulation

It is an important target of brake development to create a brake system with a high robustness for several parameter sets. The key for that target is a stability map created by simulation (Fig. 1). If the established analysis procedures are simply repeated with several parameter sets, computing time would drastically grow with every additional parameter. Therefore, advanced time saving approaches are required to create such a stability map in acceptable run times. Such time saving approaches for the sampling method were developed and implemented in PERMAS.

4. Step 1: Stability Map to Understand System Behaviour

Sampling is performed for a brake system (Fig. 2) of a series production vehicle of Dr. Ing. h.c. F. Porsche AG.

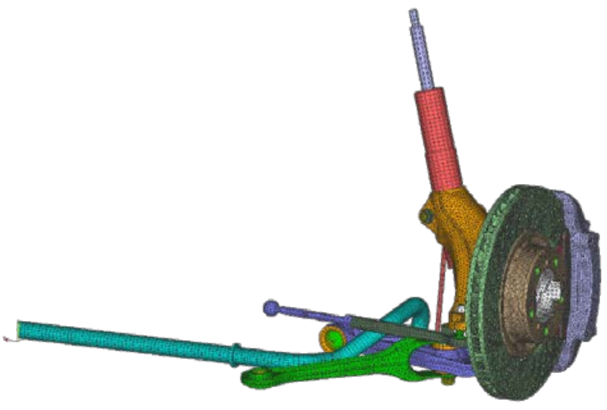

Figure 2: Brake model of a series production vehicle of Dr. Ing. h.c. F. Porsche AG

The development objective of the simulation is the design of the brake system with a high robustness for several parameter sets. The example model has 2,107,320 degrees of freedom and 18,852 contact degrees of freedom. There, 219 real eigenvalues and 240 complex eigenvalues are calculated. The following parameters are used with the new sampling method:

- 79 rotational speeds from ω = 1 to ω = 40 [rad/s] in 0.5 rad/s steps.
- 3 Young's moduli for the brake disc E = 96, 105 and 120 [GPa].
- 30 coefficients of friction between pad and disc from μ=0.3 to μ=0.9.

With the conventional method (79 * 3 * 30 =) 7,110 runs of the solver were needed.

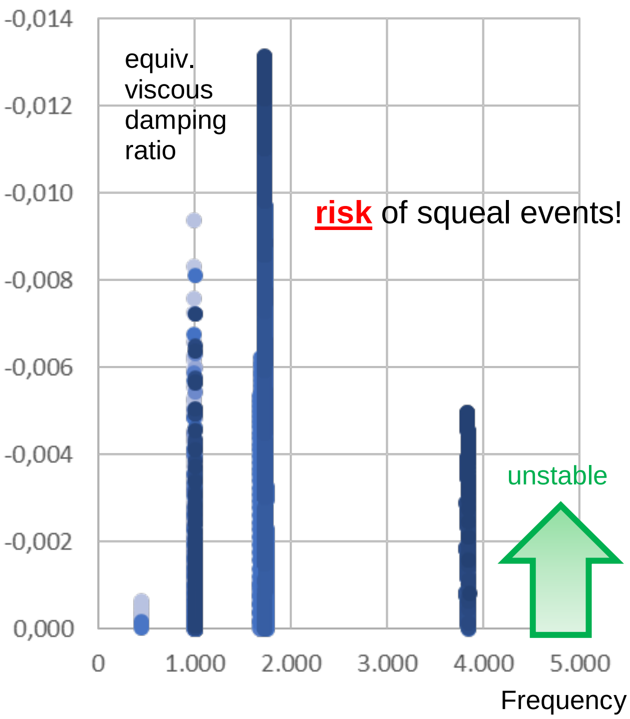

Figure 3: Stability map based on 7,110 results

Fig. 3 shows the stability map with the equivalent viscous damping ratio over frequency for the 7,110 sets of parameters. There, only the negative values are shown, because they indicate the risk of squeal events. The larger the negative value, the higher the risk of squeal events. The stability map gives an overview of the squeal behaviour. At three frequencies, potential squealing is identified (about 1000 Hz, 1700 Hz and 3800 Hz). At 1000 Hz, there is only a small change in frequency, and only some parameter sets indicate a high risk of squeal events.

A split into two frequencies exists at about 1700 Hz. A stronger dependency on the various parameter sets is present at about 3800 Hz.

The stability map is generated using all results from 7,110 samples, which are all calculated by one single computation run in about 5 hours on a single compute node.

Based on the stability map, the highest risk of squeal events is seen around 1700 Hz. A detailed analysis of this frequency range is easy, because all data required are already available. Hence, the dependency from each parameter can be extracted easily and without additional computational effort. In Fig. 4, the equivalent viscous damping ratio is on the ordinate as before and the abscissa represents the rotational speed in both charts.

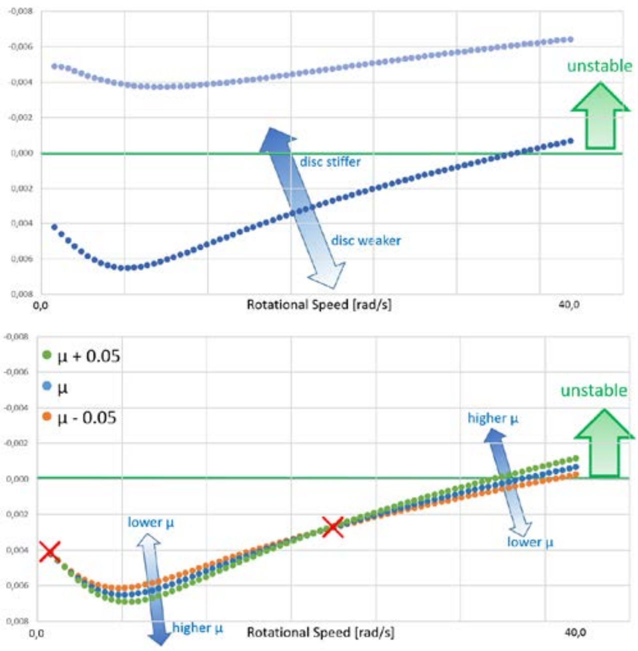

Figure 4: Change of the equivalent viscous damping ratio with respect to the rotational speed of the brake discs and with respect to the disc stiffness (top) and friction coefficient (bottom)

The upper figure shows the equivalent viscous damping ratio for a change in the brake disc stiffness over the rotation speed. An increased stiffness for the disc thus results in a higher risk of squeal noise. The increase at lower speeds is bigger than at higher speeds. The lower figure shows the variation of the friction coefficient in addition to the speed change. It can be seen a positive effect of the increase of the friction coefficient in the low speed range and a negative effect in the higher speed range. Highlighted are the speeds without sensitivity to the friction coefficient. The next step to understand the system behaviour better is provided by the participation of the real modes in a potentially unstable complex mode (Fig. 5).

Real mode 78 dominates the influence of real modes on complex mode at 1700 Hz with a participation factor of almost 35%. Real mode 75 and 79 show deformations mainly in the brake disc, while modes 83 and 84 show a deformation of the caliper. In contrast to the other modes with high participation, real mode 73 shows an influence of many components of the system.

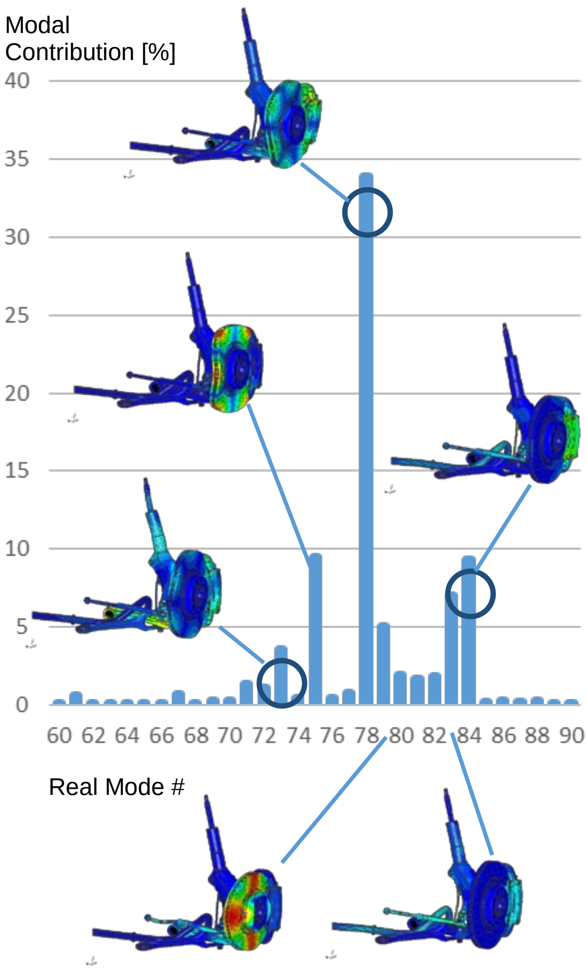

Figure 5: Participation of the real modes in the complex mode at 1700 Hz in percent

A more detailed analysis of the real modes is using the strain energy distribution. Figs. 6, 7 and 8 shows the strain energy of mode 73, 78 and 84 in two different viewings. The brake disc and hub are hidden, so the components behind become visible. High strain energies are in the range of the knuckle and the brake caliper.

5. Step 2: Sampling for Detailed Investigation

A further step in understanding the system behaviour, a second sampling is performed based on the results of the first analysis step. The second objective of this step is to quantify the potential for improvement by simulation.

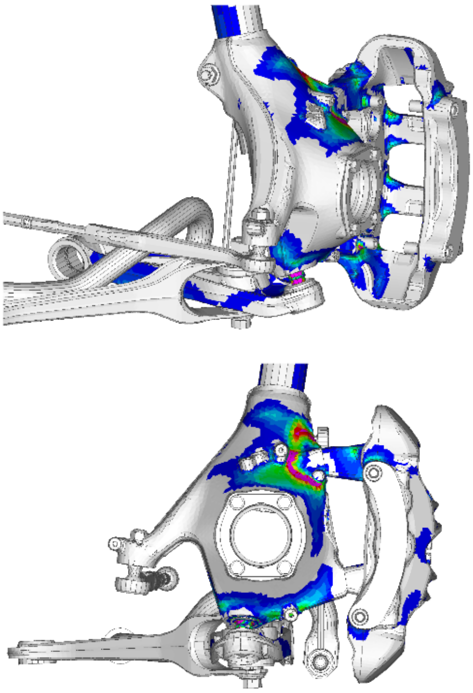

Figure 6: Real Mode 73, areas with high strain energies

The focus here is, as an example, the stiffness of the components, which have high strain energy in important real modes. In this case, these components are the knuckle and the brake caliper. Both are sampled with five different stiffness values. In addition, many rotation speeds are used, as this is possible in PERMAS [3] without significant impact on the computation time.

Fig. 9 shows that the system behaviour is more stable at a higher stiffness of the two components. The influence of the caliper is much stronger than that of the knuckle. This sampling method can be also applied to other components.

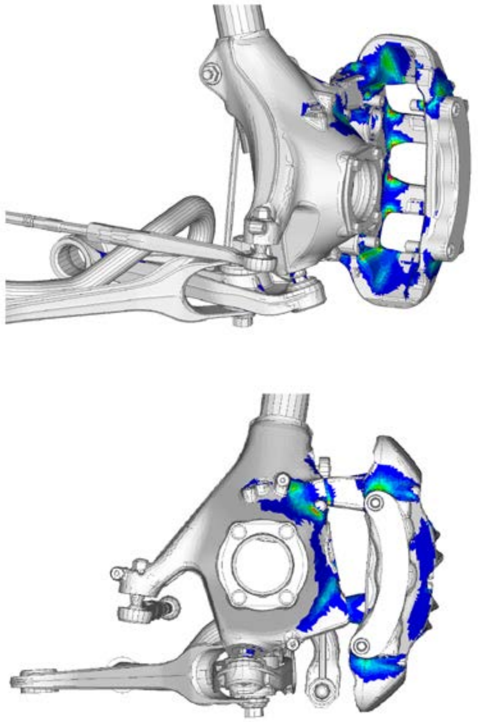

Figure 7: Real Mode 78, areas with high strain energies

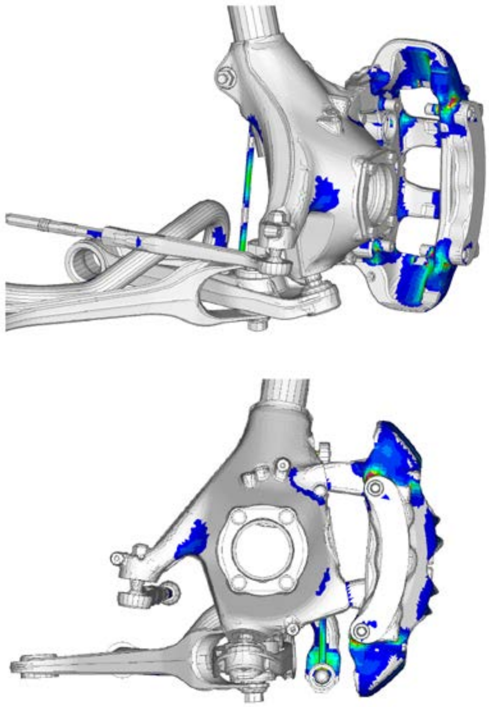

Figure 8: Real Mode 84, areas with high strain energies

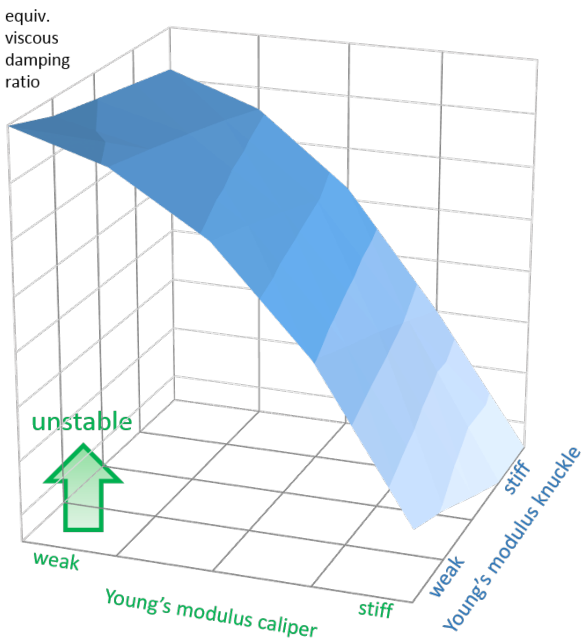

Figure 9: Effect of component stiffness on the equivalent viscous damping ratio

6. Summary and Conclusions

Parameters such as the brake pressure, the rotational speed of the brake, the friction coefficient between the brake disc and brake pad, and the elastic modulus of the brake disc can be varied in PERMAS by sampling in a single simulation run and therefore fast and efficient. The resulting stability map expands significantly the knowledge about the expected NVH performance and the robustness of the brake system for a variety of parameter combinations. In addition, the sampling, as shown by the example, is very useful for further steps of detailed analysis. Due to the very efficient new approaches, the runtimes for PERMAS sampling (about 5 hours for 7,110 samples) are significantly shorter than classical methods for parameter variation. The additional information on the stability map supports the development of a robust low-noise braking system.

7. References

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