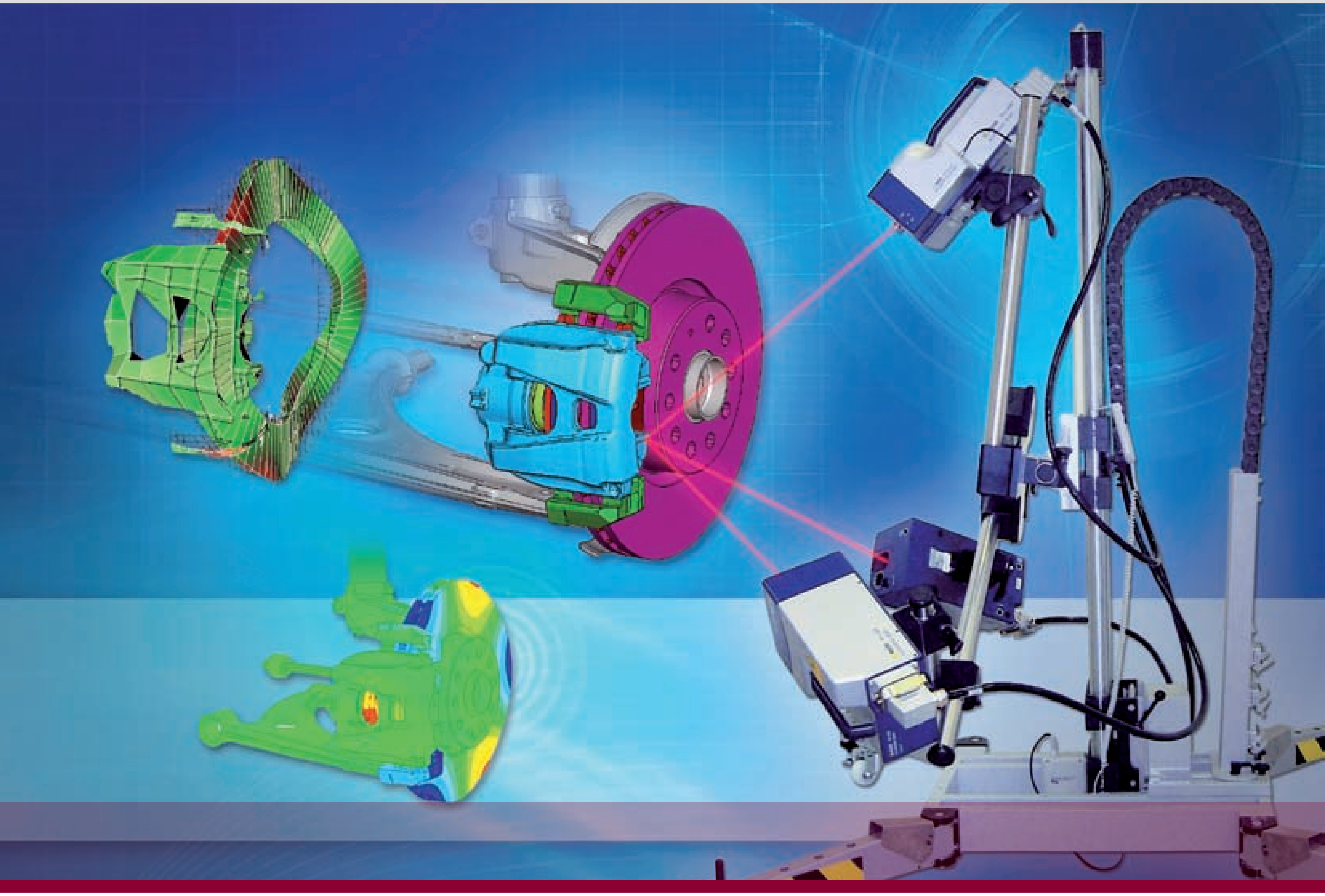

Innovative Vibration Measurement Technology for Brake Development

Apart from working reliably and providing safety, the comfort features of brakes in vehicles these days have become very significant. The requirement of a noiseless brake system always poses an important challenge when developing a new vehicle. The solution to the problem is made much easier today by using a combination of powerful analytical FE methods and measuring processes like 3-D laser vibrometry. This article describes a new measurement procedure resulting from a successful cooperation between Continental AG and Polytec GmbH.

1 Introduction

The optimization of noise and vibration has now become a significant part of the development process and generally starts during the design phase. This has been made possible by significant progress in computer-aided simulation of braking noise. The calculation of brake squeal based on the complex Eigen value analysis has become standard procedure for many vehicle manufacturers and system suppliers. This stability analysis constitutes a powerful tool for predicting brake noise. However, the method calculates more unstable modes than those that actually cause noise. Furthermore, at the start of development, not all the relevant system parameters are known. An example of this would be the material properties of the brake pads.

For these reasons, it is necessary to compare the results of simulation to dynamometer testing and verify whether the calculated vibration modes actually occur in practice. Only if there are very closely harmonized working methods between simulation and testing, is it possible to ensure that effective and selective optimization is carried out. Bearing this in mind, it is clear that even modern simulation processes can only be as good as the accompanying measurement technology allows. The aim of the integral approach described in this article is to unite numerical and experimental processes in such a way that the advantages of both techniques are fully exploited and the greatest possible efficiency results are achieved.

Measurement processes presented in the market so far can roughly be divided into two categories: non-contact optical processes, such as laser vibrometry or holography and, tactile multi-channel measurement devices using accelerometers. Both have specific advantages and disadvantages and each qualify for certain applications. Optical processes, for example, are limited to those areas, which are optically accessible, either directly or indirectly (for example by using mirrors). The parts of brakes hidden on the inside or shadowed can thus not be acquired. Tactile processes in turn are not reactionless and can be directly affected by the influences of the surroundings. Additionally, tactile sensors have to be applied with great effort and are limited by the number of measurement channels. Because of these limitations, it is not possible to achieve the quality of the simulation using just one of the given measurements methods.

To be able to contrast the calculated results with appropriate measurements, a new measurement process has been developed using operational deflection shape analysis obtained from 3-D scanning laser vibrometry. Optical measurement technology gains additional meaningfulness when intelligently combined with complementary, tactile measurement technology. The complete vibration data sets acquired this way allow effective updating of parameters in the described FE models and thus improve the vibration characteristics of the product by selective optimization of the structure. In addition, the process results in a number of other findings, which are also discussed in this article. Along with presenting the process based on selected examples, the potentials and problems of the new measurement process are discussed and the future perspectives are appraised.

2 Finite Element Simulation

Audible squeal is caused by self-induced vibrations of the brake system as a result of friction between the brake pad and the brake disc. Mathematically it can be described with the vibration equation

$$M\ddot{q} + D\dot{q} + Kq = f(t)$$

where M is the mass matrix, D is the damping matrix, K is the stiffness matrix and q is the displacement vector. An external, stimulating force f(t) is not necessary for the vibration to be excited. Because of the friction between the brake pad and the disc, there are entries in the stiffness matrix with non conservative restoring forces. The stiffness matrix thus becomes asymmetric which requires special software with a complex solver algorithm to solve the system of equations. With the process of complex Eigen value analysis, unstable modes can be determined numerically and the self excited vibration can be calculated. For this purpose, the first step is to prepare an FE model of the

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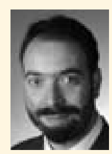

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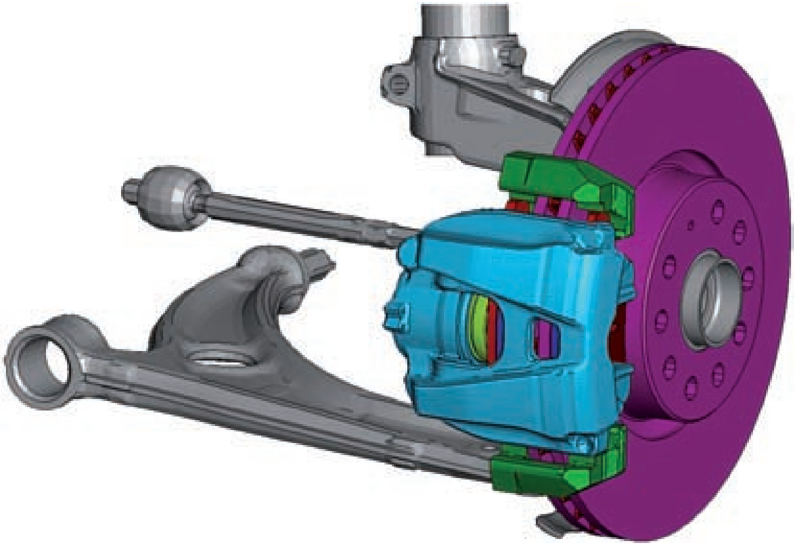

Figure 1: FE model of the brake system for complex Eigen value analysis

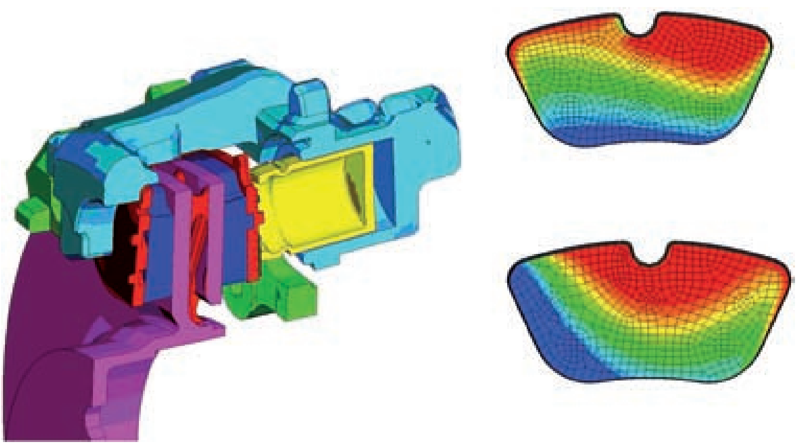

Figure 2: Presentation of the components (left) and calculation of deformation due to clamping and peripheral forces (right)

brake system with all adjacent chassis components. **Figure 1** shows such an FE model, consisting of several hundred thousand degrees of freedom.

In the second step, the static deformations in the brake system are calculated for the braking processes to be examined, depending in the braking pressure, the temperature and other parameters, **Figure 2**.

Next, the complex Eigen values of the system for every individual brake application have to be determined in the third step. Their excitation rates provide an indication, about which of the calculated frequencies could represent a possible cause of squealing brakes. **Figure 3** shows the results of the complex Eigen value analysis. In the given example, two potential problem frequencies were found, whereas in practice only one of them leads to an audible squeal.

3 Metrological Basics

Complete characterization of complex vibro-acoustic dynamics of a brake system is made easier through the high performance optical metrology that is commercially available today. For all measured points that can be reached optically either directly or using mirrors, the process of 3-D scanning laser vibrometry can be used to measure vibrations. The 3-D scanning vibrometer, as it is offered by Polytec, is a complete measurement system for fast, simple, non-contact and reactionless (non mass loading) acquisition of three-dimensional vibrations of complex mechanical structures. It supplies important high resolution dynamic structural information over space and time, such as response functions, resonant frequencies, Eigen shapes and operational deflection shapes. **Figure 4** shows the optical configuration of a laser vibrometer [1].

The process is based on the optical Doppler effect, which states that light waves experience a change in their vibration frequency after scattering at a moving surface. The frequency shift is directly proportional to the instantaneous value of the vibration velocity and despite its amazingly small relative value of less than 10−8, can be extremely precisely determined via interferometric processes. For this purpose, inside the vibrometer, the laser light scattered back is compared to a reference beam with a defined frequency (heterodyne technique, Figure 4). Only the velocity component on the direction of the beam has an effect on the Doppler frequency shift.

This is why the 3-D scanning vibrometer is used here to completely acquire the velocity vector at the measured point using three independent laser beams, **Figure 5**, measuring from different spatial directions to fully acquire the movement of the measured point. To measure operational deflection shapes, the three laser beams are synchronously scanned over a predefined grid on the surface of the test object and a complete vibration measurement is carried out at every measurement point. A simultaneously acquired phase reference provides a correct reference to the sequentially measured points and allows calculation and animation of the operational deflection shapes from the data sets. In comparison to other optical processes, such as holography or "electronic speckle pattern interferometry" (ESPI), the vibration has to be present for the duration of the scan. But in return, the full frequency response of the structure is obtained during one measurement so that complex modes (deflections with phase delay within the measured points) are correctly measuring and presented.

With its extension to measure all three spatial directions, 3-D scanning vibrometry has become a powerful tool for developing brake systems, enabling the measurement of both in-plane and out-of-plane

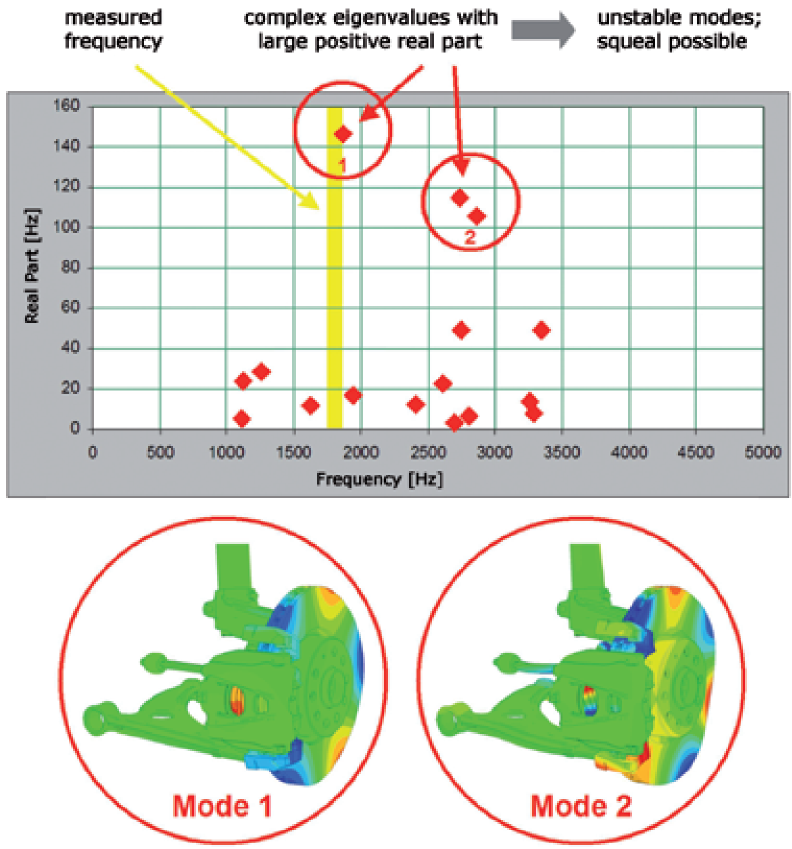

Figure 3: Two results of complex Eigen value analysis as modes 1 and 2 – only mode 1 at the measured frequency (yellow) leads to an audible squeal

operational deflection shapes of the relevant components in a single test setup. The non-contact technique significantly simplifies the measurement task, as complex sensor adapters and cabling of the object can be limited to the measured points that are not optically accessible.

4 Experimental Investigations

The same factors that apply to the simulation of squealing brakes equally apply to experiments on test rigs: The vibration characteristics need to be regarded as a system characteristic and require appropriate test structures. Not only the brake assembly, but also all parts of the suspension components attached to the steering knuckle (spring strut, wishbones, etc.) are relevant to the vibration and therefore also need to be attached. To be able to experimentally evaluate the noise behavior, an automatic noise search program is run first on the brake dynamometer, **Figure 6**, a so-called matrix test. This covers all operating modes relevant to comfort that the driver experiences in real traffic on the roads. Here, brake noise is recorded and evaluated with regards to parameters such as direction of travel, speed, braking pressure, temperature, squeal frequency, noise level and duration or number of braking cycles, respectively. For this purpose, there are generally defined programs, such as the software "AK Noise" from the working group „Bremsgeräusche", a expert group initiated by German automobile manufacturers and suppliers, or that in the standard SAE J2521, but also special programs from vehicle and brakes manufacturers [2, 3].

Once the parameters, under which the brake squeals, are known, the operational deflection shape analysis with the combined sensor system can be carried out. Before starting the actual measurement, the measurement system has to be prepared for the current measurement task. Setting up the vibrometer includes positioning the sensor head system in relation to the object, system-supported alignment of the coordinate systems of the object and the sensor system, defining the measurement grid (through FE import or interactively with video support) and setting data acquisition parameters for the measurement. For recurring dynamometer investigations, as is the case described here, a motorized overall stand supports reproducible positioning of the vibrometer on the dynamometer.

On the dynamometer, the critical parameters are targeted that lead to undesirable squealing of the brakes. This is carried out sequentially for all frequencies determined in the noise search program. To optically acquire as many measured points as possible, the vibration characteristics of the brake system are measured from different spatial directions. Instead of making several measurements from different positions and stitching them together, surfaces not di-

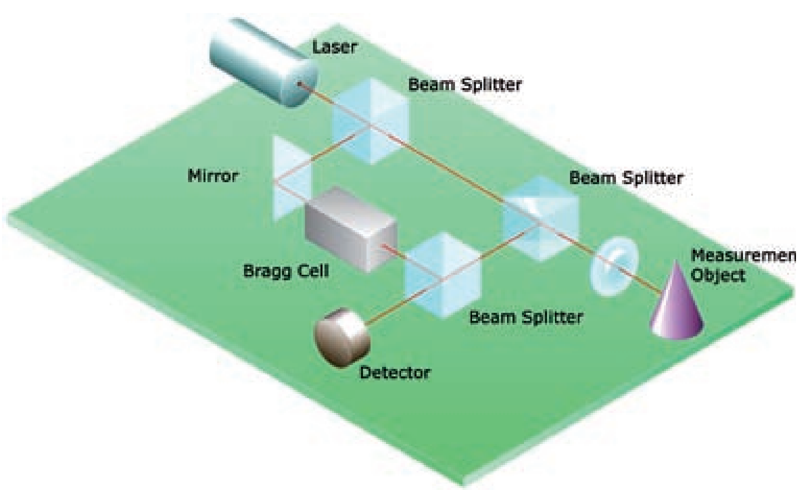

Figure 4: Optical configuration of a laser vibrometer [1]

Figure 5: Three 3D scanning vibrometer sensor heads

rectly accessible from the sensor position are measured via mirrors, meaning that measurements can be obtained in a single scan, without needing to reposition the vibrometer. In this case, coordinate transformation is necessary to make sure that all measurements both with and without a movable mirror are displayed in the same coordinate system. This can also be done automatically if the mirror positions are predefined during setup.

To compensate for the limitations of the optical process mentioned earlier, tri-axial accelerometers are used in addition to the laser scan [4], applicated in the cooling ducts of the brake disc, **Figure 7**. To do this, up to four tri-axial accelerometers are mounted on the inner and outer edge of both brake discs with the aid of a high temperature adhesive so that measurements can be made up to 180 °C. The measurement signals are amplified by small inline charge amplifiers that are also rotating and are then fed via a slip ring assembly or by using appropriate telemetry to signal conditioning and data acquisition.

To obtain phase matching and be consistent in the way the measurement data from 3-D scanning vibrometers [5] and accelerometers are treated and presented, the accelerometer signals are also acquired by the vibrometer's data acquisition system. The continuously occurring time signals from the accelerometers are synchronized per revolution with the position of the brake disc by a trigger pulse from an RPM sensor. The time data set allocated to a revolution is divided into up to 360 equally sized, angle equivalent sections to which the software allocates geometry segments, which had previously been defined in the stationary coordinate system.

Each section is subjected to FFT analysis and the acceleration amplitudes are converted into deflections. Typical rotation speeds of for example 10 rpm then correspond at 4° resolution to time data sets of 67 ms, which are sampled at 204.8 kHz, so that with 6400 FFT lines a frequency resolution of 12.5 Hz can be attained.Here the user always faces a conflict of aims. Either it's possible to attain a high angular resolution at the expense of frequency resolution or vice versa. This is due to the fact that in case of increasing the number of angle segments, the length of the time data set for each FFT will decrease. In this example, the frequency resolution falls to 50 Hz if the angular resolution is set to 1°.

The vectorial speeds measured by the vibrometer at optically accessible sample points are also FFT analyzed and transformed to deflections. A phase reference that is measured simultaneously is used to relate the optical and tactile measured data sequence data to each other. The result is a complete, three dimensional operational brake deflection shape analysis of all components of interest (optically accessible and inaccessible and rotational as well as stationary). A snapshot of the animation is shown in **Figure 8**.

The complete vibration data sets acquired in this way fulfill all requirements for effective comparison of parameters in the FE models described. Correlation can be verified by a manual comparison of the measured with the calculated frequencies and modes or through suitable numerical

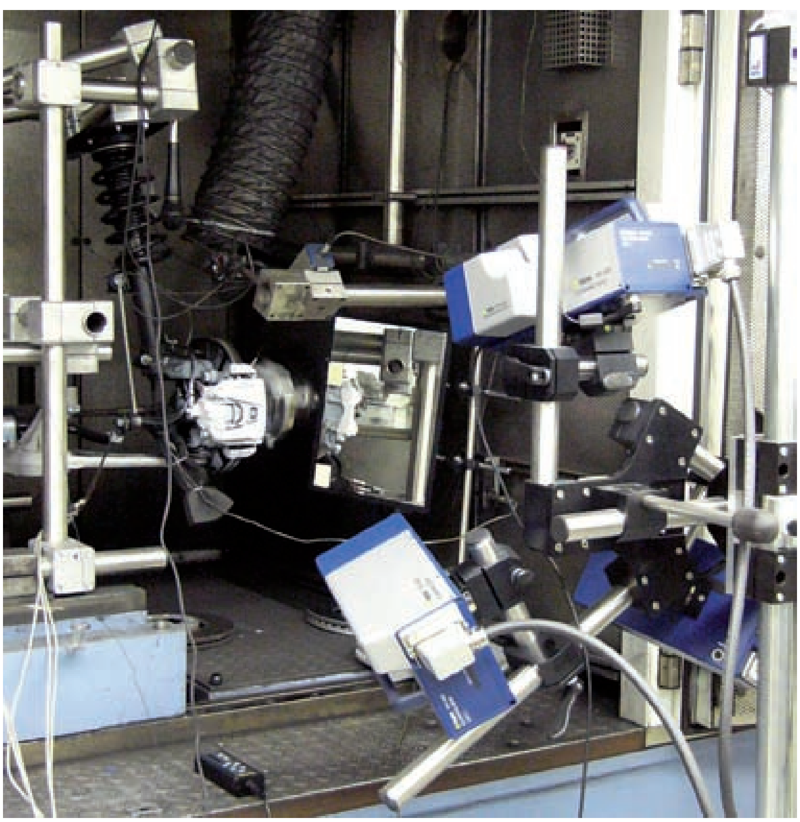

Figure 6: Brake system on the brake dynamometer [2]

processes. Primarily MAC analysis (Modal Assurance Criterion) should be mentioned here; it is used to determine the correlation coefficient between the measured and the calculated vibration. The correlation coefficient is a sign of accordance and allows a statement to be made on the quality of the simulation.

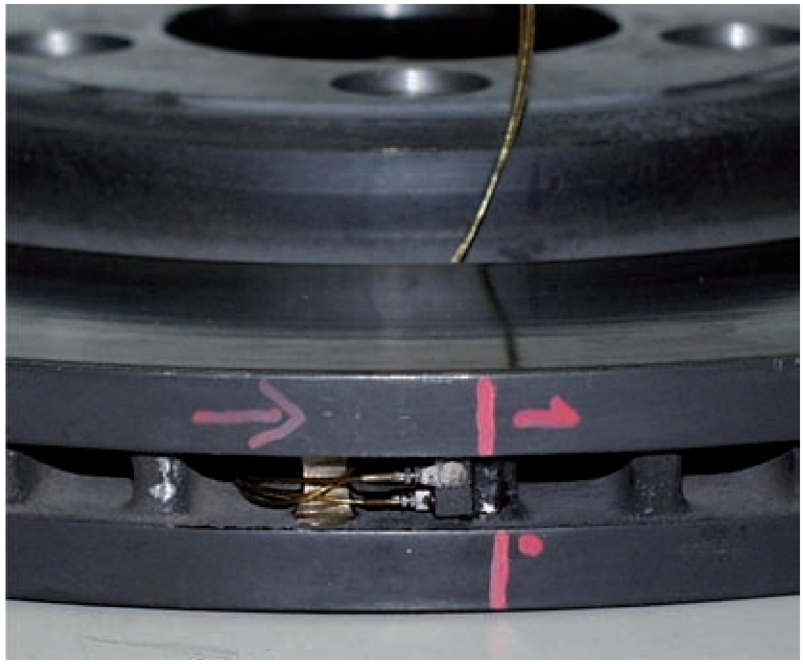

Figure 7: Brake disc with tri-axial accelerometer [4]

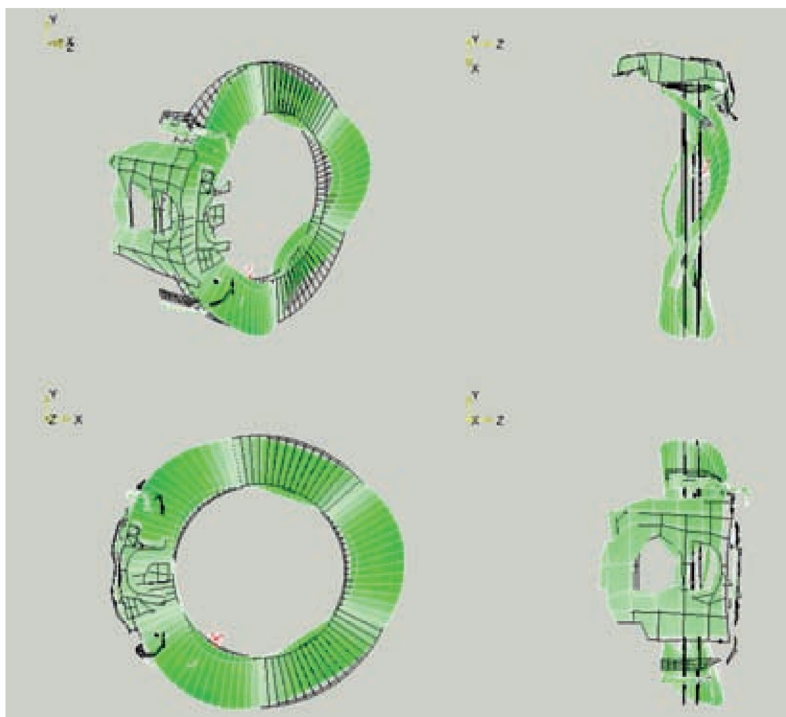

Figure 8: Measured operational deflection shapes

5 Countermeasures

Once the operational deflection shapes and mode-shapes have been determined and validated, suitable measures can be taken to prevent the emergence of self-excitation, allowing undesirable squealing of brakes to be suppressed. Structural modifications that can selectively shift the resonant frequencies are given priority here. Only after that are secondary measures taken which have an insulating or damping effect on the vibration. Using FE models is particularly advantageous for defining structural modifications.

From a technical testing point of view, selected variation of numerous combinations of different parameters is quite difficult to carry out. The real brake is subject to operational wear and tear and often only statistically detectable scattering of the noise relevant parameters is possible. For this reason, it is virtually impossible to create exactly reproducible operating conditions for different combinations of measures. Numerically however, it is no problem to change several parameters or only individual selected ones. The FE models that have been verified based on the operational deflection shape analysis thus provide the basic requirements for successful use of optimization algorithms. For this purpose, a design space is defined, which covers all potential parameters for reducing noise. Here, the participation factors that can be calculated from the 3-D measurements or the simulated vibrations are a great help, as they represent a measure of the proportional vibration energy of the respective component. A suitable system answer is also defined which needs to be optimized. A suitable system answer is the analysis of the system for instabilities which can arise from coupling two adjacent modes.

Figure 9 shows how the coupling of two modes is depending on the friction coefficient. The dashed red line shows the frequency of one mode, the solid blue line the frequency of an adjacent second mode. Increasing the friction coefficient changes the frequency of both modes. At first the frequency of the first mode decreases while the frequency of the second mode increases until both frequencies join at a frequency coefficient of 0.1. In this case the two modes couple; the Eigen values become complex. If the friction increases even further, then as a general rule, the affected Eigen shapes are excited. Due to the neg-

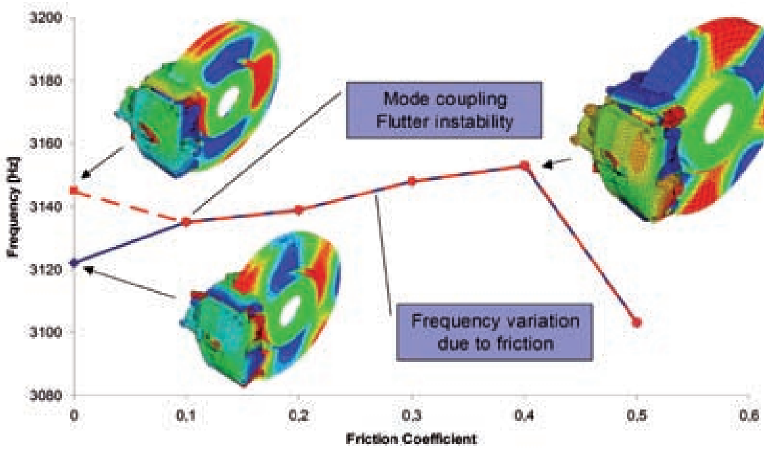

Figure 9: Emergence of squealing brakes due to mode coupling depending on frequency and friction coefficient

Figure 10: Structural imbalance due to additional mass on the brake caliper (solution, left, and basis, right)

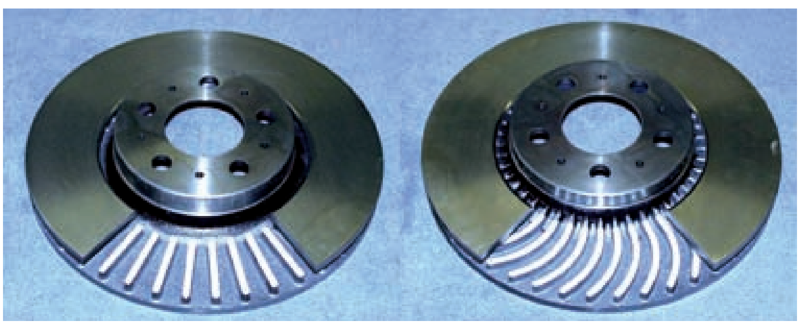

Figure 11: Structural imbalance due to modification of the disc stiffness (straight and bent cooling ducts)

ative damping that has occurred, the vibration rings out and leads to squealing brakes.

To eliminate brake squeal, the resonant frequencies are shifted so far that no mode coupling occurs any more. This is done using an optimization processes, which examines the given design space for the best set of parameters to reliably suppress the onset of self-exciting vibrations. The optimized design is then verified on the dynamometer and in the vehicle. Examples of measures taken are shown in **Figure 10** and **Figure 11**.

6 Conclusions and Outlook

To be able to accommodate the ever decreasing development times with ever increasing demands, vibration optimization has to be effective and targeted. It is better to include numerical noise simulation early on during the design stage. This requires interdisciplinary cooperation between the vehicle manufacturer and the system supplier and for example, presumes the exchange of design documentation (CAD data). This opens up the possibility of taking constructive measures at an early stage, which would otherwise lead to high costs or a time delay if required at a later stage in development.

Due to the given limitations, simulation processes cannot however replace brake dynamometer and vehicle testing but can only supplement them and at best, limit them to the absolute minimum. Experimental testing with meaningful yet simple and productively applicable measurement technology will continue to remain indispensable. The interlinked use of two complementary measurement procedures (3-D scanning laser vibrometry and multi-channel measurement devices using accelerometers) presented here by Continental AG and Polytec GmbH used in conjunction with FE modeling represent a tried and trusted method for efficient development of noise-optimized brake systems.

Continued development of computer hardware and increased processing performance promise even faster results in the area of FE calculation. This will allow the predefined design room to be extended by additional parameters and increase the attractiveness of numerical optimization processes even more.

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