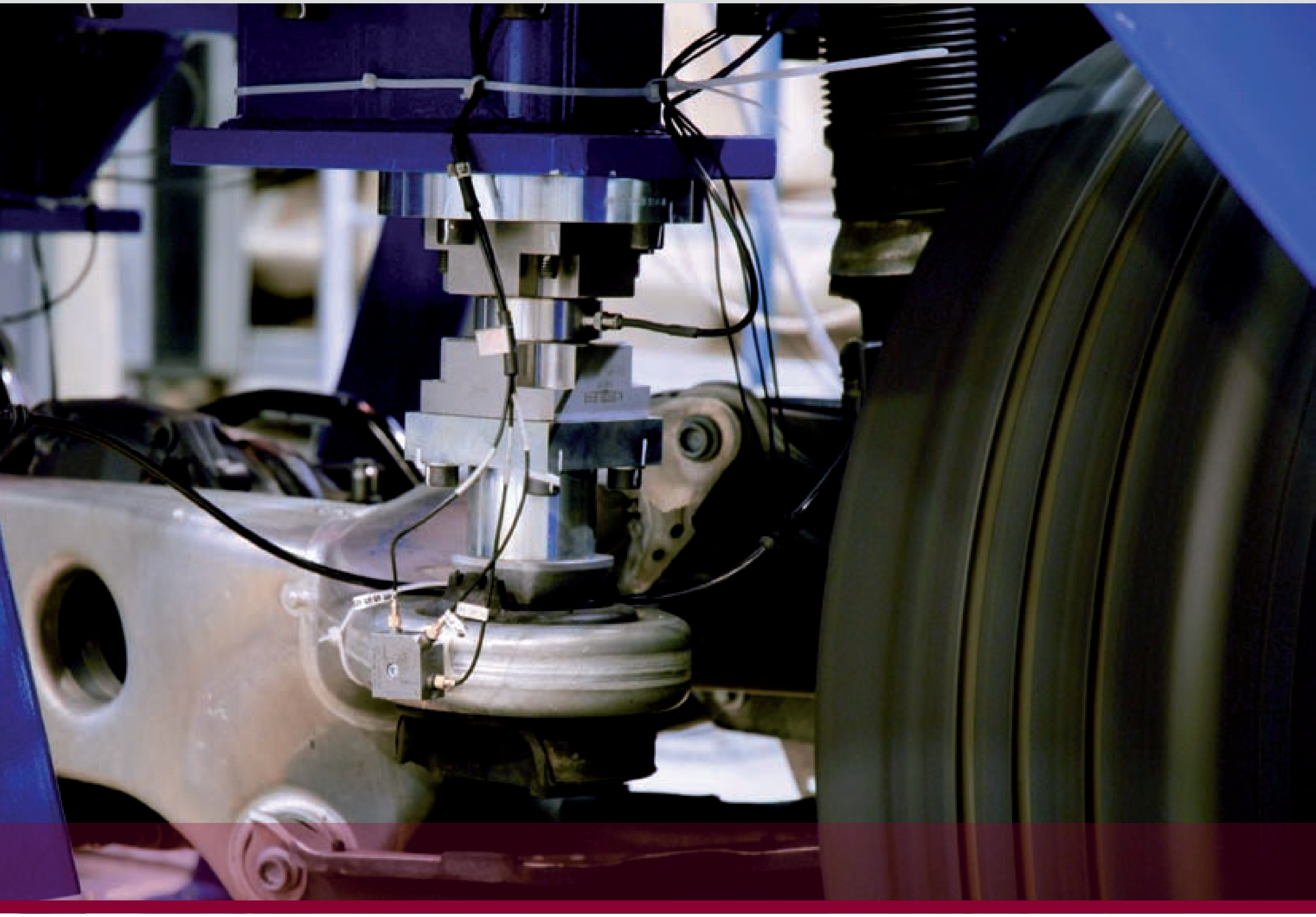


Vibration-optimized Components for Chassis Systems

Vibration optimization in vehicles increasingly requires analysis of individual suspension components as well as their interaction in a vibration system. The greatest challenge is the creation of suitable models that are as simple as possible, yet sufficiently conclusive for a detailed definition of component requirements. For that reason entire axle systems – including corresponding suspension struts – are characterized at Vibracoustic GmbH & Co. KG using dynamic measurement techniques. Parallel model development as a multi-body system makes it possible to define component requirements and analyze their optimization potential. Subsequent verification of results on the axles provides additional fundamental data for model development.

1 Introduction

In addition to intensive effort at the level of individual component design, improvements to driving comfort and vehicle acoustics will require accurate examination of the suspension as an overall system and the interaction that takes place between the components employed. The isolated examination and advanced development of individual suspension or damping components often leads to disappointing results in the overall system despite the great overhead expended. This is because the interaction of components has not been sufficiently investigated and the contribution that the improvement of a single component can provide has been incorrectly estimated.

At Vibracoustic, vehicle subsystems subject to vibration are investigated in their entire form in order to recognize existing potential and to optimally exploit this potential in the development process on the component level. Based on a detailed knowledge of the system behavior obtained from test stand investigations, computational simulation models can be derived which, as a multibody model, physical component model or material model, will contain all necessary characteristics to show the potential of component development with respect to comfort improvements. In particular, the vibration system formed by the chassis and suspension represents a complex interaction of different component and material characteristics whose investigation and description is presented below.

2 Experimental Analysis of Existing Axles

A number of axle test beds have been developed and built at Vibracoustic for experimental analysis of complete passenger car axles. Four experiment setups are used to determine the quasistatic and dynamic behavior of the axle. During all experiments, the axle is connected to the test bed's rigid frame structure by way of tri-axial load cells. These load cells are positioned at the points where the axle interfaces with the vehicle body. They detect the internal forces between axle and body. However, results produced using this experimental setup are applicable only to rigid boundary conditions. The elastic characteristics of the chassis are not included.

The objective of experimental investigation on existing axles is to determine their characteristics using static and complex dynamic stiffness rates. A distinction has to be made between input stiffness and transfer stiffness as described in the following.

2.1 Input Stiffness

Input stiffness is the ratio of applied force and applied distance. This is determined by setting up the axles without wheels. The wheel knuckles are mounted to flanges that are attached to load cells connected to hydraulic cylinders. Subsequently, the input stiffness is measured quasistatically for different amplitudes and dynamically at a fixed amplitude. This is achieved by stimulating the axle sequentially in the vehicle's longitudinal, lateral and vertical direction.

Different experimental setups are available for analysis in the vertical direction. If the complete axle is measured, the result is the overall stiffness rate of the suspension. In case Non-McPherson front axles are investigated, the axle can be measured without suspension struts. In that case the measured input stiffness corresponds to the parasitic spring rate, **Figure 1**. Additional experiments without stabilizer permit an experimental contribution analysis for overall and parasitic stiffness rates.

Overall and parasitic stiffness rates usually show a strong correlation to the amplitude of the excited displacement. Generally, a differentiation has to be made between the behavior for small and large signals. The suspension springs and suspension bushings are dominant for the stiffness rate at large amplitudes. Friction effects, such as those exhibited by ball joints or seals in hydraulic shocks, are dominating the stiffness rate for smaller amplitudes and contribute significantly to the stiffening of the axle. Some axles investigated revealed a stiff-

Authors

Dr.-Ing. Hendrik Sell is head of Acoustics, Vehicle Testing and NVH CV at Vibracoustic GmbH & Co. KG in Hamburg (Germany).

Dipl.-Ing. Tobias Ehrt is CAE engineer at Vibracoustic GmbH & Co. KG in Weinheim (Germany).

Dr.-Ing. Michael Meß is head of Advanced Development Hamburg at Vibracoustic GmbH & Co. KG in Hamburg (Germany).

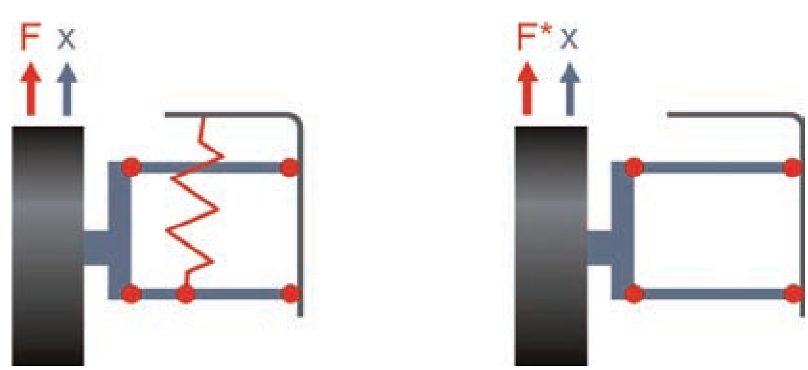


Figure 1: Schematic diagram to get the overall (left) and parasitic (right) spring rates at the force F and the distance x

ening factor of 25 between the parasitic spring rate at large amplitudes and those for small amplitudes.

The overall and parasitic stiffness rate shows a strong dependence to frequency. However, the behavior of individual components is not a major cause of this. The stiffness rate at higher frequencies is mostly dominated by the effects of inertia. At higher frequencies a constant displacement amplitude requires the inert mass of the suspension to be accelerated further and further. The resulting forces of inertia are also measured by the input-side load cell allowing for the calculation of the inert mass of the suspension.

2.2 Transfer Stiffness

In contrast to input stiffness, transfer stiffness is determined by measuring the force on the output-side. The test setup is comparable to the test setup for dynamic components in which one side of the part under test is excited and the force measurements are made on the other side, which is quiescent, **Figure 2**. Masses at the input and output sides of the system have no influence on the measurement of transfer stiffness. The test bed's actuators supply the displacement regulated stimulation applied to the input side. Inert masses at the input then lead to greater motion forces for the actuator. The output side of the test object is rigid, thus no forces of inertia are created.

In contrast to a component test bed, an axle analysis presents the challenge of being a „single input / multiple output" system. If wheel knuckles are stimulated in one direction, reaction forces are produced at all interfaces to the vehicle's chassis – or in this case to the test bed's frame, **Figure 3**. A rear axle with separate springs and shocks, for example, that is connected to the body with four rear axle sub-frame mounts has eight chassis connecting points at which the x, y, and z direction forces imposed at each point must be measured. This example therefore results in a system with one input and 24 outputs.

The frequency-dependent behavior of transfer stiffness factors is of particular interest. These reflect the suspension's Eigen-frequencies. A rear axle as described above, would hardly have frequency-dependent transfer stiffness factors at rates below 10 Hz.

Beyond 15 Hz it is possible for Eigen-frequencies to occur due to the elastic connection of the rear axle sub-frame mounts. In the frequency range between 60 and 100 Hz continuum resonances often arise from steel springs or bending vibrations of shocks that produce increased transfer stiffness in these transfer paths.

Greater transfer stiffness rate means that an increased forced is produced at the output side for the same input displacement. These reciprocal dynamic forces are largely responsible for vibration and acoustic comfort in a vehicle.

Figure 4 shows the transfer stiffness as a function of frequency. Every line represents a so-called transfer path. The color is a measure of stiffness contribution. Figure 4 also demonstrates that the pattern between transfer paths changes as a function of frequency. Depending on their frequency, the greatest forces are introduced at different locations into the body. This type of representation provides the OEM with information about potential interaction between axle and body. It is critical to assure that the resonant frequencies of the body do not correspond to the resonant frequencies of the axle.

2.3 Interface Force Spectra

After analyzing the suspension's input and transfer stiffness the axle will now be investigated including the wheels. The entire test setup is designed for a fixed axle connection via load cells such that it can be operated on Vibracoustic's acoustic chassis dynamometer. The speed-regulated rollers are now used as the source of stimulation; this permits stochastic excitation. The force spectra measured at the ends of all transfer paths (for example at the axle/body interfaces) will be measured, **Figure 5**. Constant speeds and run-ups will be measured and evaluated. This test setup allows to measure the forces directly at the interface points with the aforementioned restrictions.

In contrast, vehicle investigations for experimental noise path analysis are usually implemented with indirect force detection methods, such as the stiffness method or the inverse matrix method. Of particular interest are frequency distributions within force spectra and the paths showing especially great forces.

2.4 Transient Axle Behavior

Following the hydraulic stimulation of the axle with sine-wave shaped signals and the stochastic excitation of the rolling surface an analyses was performed with so-called bump strips attached to the roller. This is done by mounting profiles with various geometric shapes. Rolling over the bump strip causes an impulse-like stimulation having components in the vehicle's vertical and longitudinal directions.

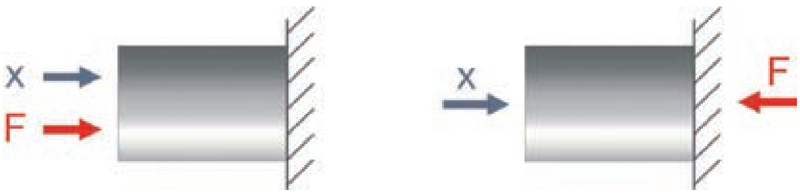


Figure 2: Input (left) and transfer (right) stiffness result from the nature of the experimental setup at the force F and the distance x

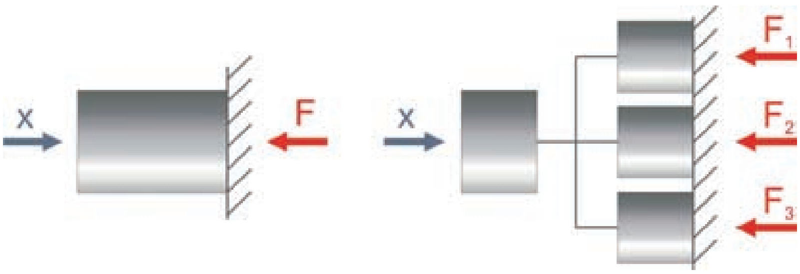


Figure 3: Differentiation between input / output behavior of a system in single input / single output (left) und single input / multiple output (right)

In addition to the force measurement techniques described earlier, these experiments also employ a high-speed optical measurement system for 3D coordinate detection. **Figure 6** shows the adhesive reflectors attached to the wheel. Each reflector corresponds to a 3D distance sensor that delivers 1000 measurements per second. This even allows for the assessment of bump processes with adequate resolution.

Evaluation of optical measurement system data provides two results. First, the wheel's upward motion curve is determined for transient stimulation. The wheel's upward motion curve reveals the interaction between the vehicle's suspension in the vertical direction and the deflection in the vehicle's longitudinal direction. A large longitudinal compliance of the axle ensures, for example, that smaller accelerations occur in the vertical direction. This is entirely plausible since the wheel has more time available to roll up onto the threshold.

The second result of optical measurement is an assessment of the axle's attenuation behavior. This is evaluated separately for the vehicle's vertical and longitudinal directions. Attenuation behavior of vehicle vibration in the vertical direction is decisively influenced by the shock absorbers. The attenuation behavior in the vehicle's longitudinal direction can be critical for comfort when front or rear sub-frame mounts with low damping in the longitudinal direction are used. After running over an obstacle this can lead to noticeable post-tremble of the axle.

3 Analysis of Involved Components

Most of the suspension components used to generate the desired kinematics, elasto-kinematics or the isolation of structure-borne noise shows a high nonlinear characteristic. This means that the dynamic behavior is dependent on amplitude, pre-stress and temperature. Components subject to friction, such as ball joints or oil-filled shocks, are also part of this component group together with rubber/metal components like suspension bushings or shock absorber bushings and top mounts. When measuring non-linear components it is important to

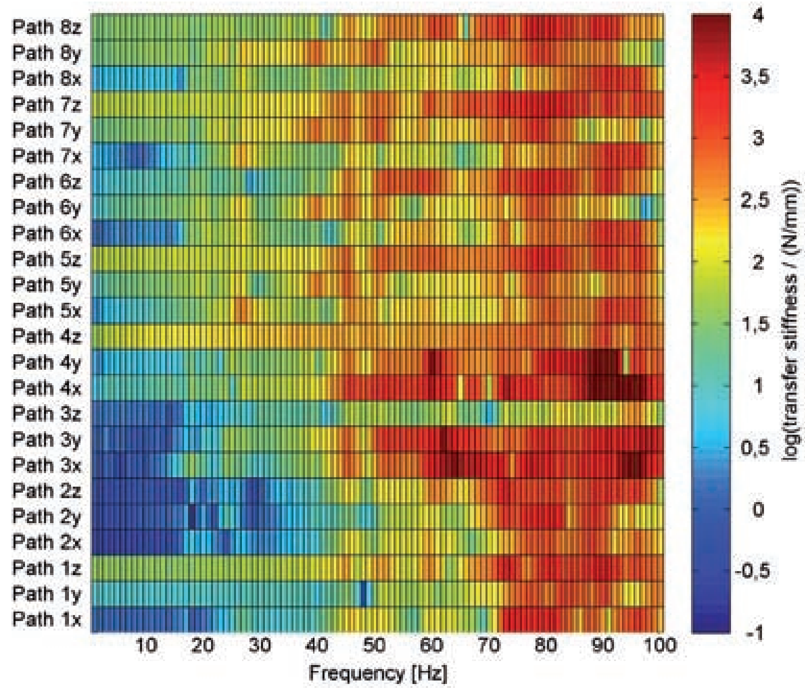
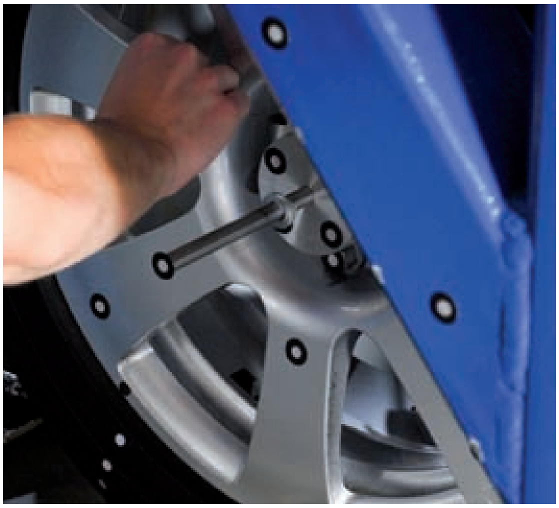


Figure 4: Presentation of measured output data as a matrix with transfer stiffness for the assessment of a vehicle axle

Figure 5: Assessment of an axle (with wheel) on a chassis dynamometer rolling test stand

Figure 6: Preparing the axle for a six-dimension dynamic test on a chassis dynamometer rolling test stand



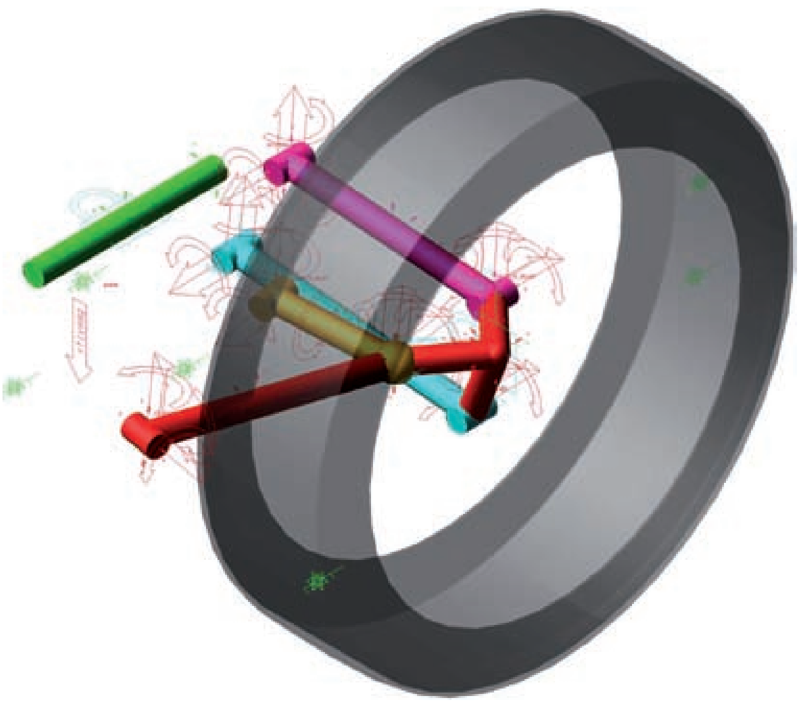


Figure 7: Axle setup as a multi-body system

select test parameters and test conditions such that they correspond to real application conditions.

Vibracoustic has defined an entire series of standard tests for components. The quasi-static behavior of components is described by force/displacement curves. These also reveal geometric non-linearities, which are changes to a component behavior as a function of the displacement. Geometric non-linearity is usually realized by linear travel and stoppers.

Frequency measurements identify the dependency of the dynamic stiffness rate to the test frequency. This is done by measuring at constant displacement amplitude and a variable test frequency. A strong frequency dependence is exhibited for hydraulic suspension components and tuned mass dampers used to avoid resonance phenomena.

The amplitude sweep is measured at a constant frequency and variable displacement amplitude. It shows the dependency of dynamic component stiffness rate for the excitation amplitude.

The dynamic stiffness as a function of the pre-load is measured at a constant frequency and constant displacement amplitude. In contrast to the initially presented quasi-static measurements, pre-load measurements determine the influence of geometric non-linearity with respect to dynamic stiffness. Dynamic stiffness is therefore not only determined for the standard working point of the component. It is also determined for situations in which the component is at its limits or on its way there.

4 Modeling the Axle

Usually a multi-body model of the respective axle is created to accompany the experimental investigation. The options available for switching effects on and off for simulation model components is supportive for the interpretation of measurement results and permits the contributing analyses and estimations of the potential for possible solutions.

4.1 Multi-body Models

A multi-body model represents the kinematics and elasto-kinematics of the axle. The model contains all suspension elements, as there are control arms or links and optionally a sub-frame, as an ideal stiff body, interconnected by coupling and force elements, **Figure 7**. This formulation of coupling permits the level of detail for the representation to be controlled. A real ball joint, for example, can be represented as frictionless or subject to friction; in turn, the friction can be made dependent on various influencing factors, such as joint pre-tension and joint load.

4.2 Contribution Analyses and Potential Estimate

Contribution analyses provide an initial indication of the component to be processed in order to reach a particular comfort objective. Individual components are replaced by elements with ideal characteristics. A ball joint or a suspension bushing, for example, is replaced by an ideal frictionless ball joint and then the effects on the global parameters are evaluated.

In contrast, the potential of an optimized component can be estimated with respect to its initial state and its ideal state. Furthermore, undesirable side effects, such as changes in the suspension's traverse stiffness can be investigated. These techniques make it possible to identify the technically optimal solutions in a very quick, economical manner.

The primary application field for contribution analysis is for calculations of overall and parasitic stiffness rates. Initially the trials of the experimental contribution analysis are recreated in order to validate the model. Subsequently, the contribution of individual components can be investigated; a process which would require substantial experimental effort or which even would not be possible in full scope at all.

The combination of contribution analysis and potential estimation results in a cost-optimized procedure. After contribution analysis indicates the primary component to be optimized, that particular component can be optimized according to engineering and economic perspectives. If, for example, the friction of the piston rod of the oil-filled shock is identified the primary source of increased parasitic stiffness rate at low excitations, it is not absolutely necessary to replace it with an expensive low-friction shock. Often it is possible to obtain the desired comfort effect by fitting the existing shock with a modified top mount.

Longitudinal compliance in the suspension increases driving comfort, particularly when driving over small bumps. Compliance is typically provided by an elastically connected sub-frame, frequently in conjunction with so-called „comfort bushings“.

The sub-frame is isolated from the remainder body of the vehicle by soft, progressive sub-frame mounts which are slightly dampened. Highly frequent disturbances such as the tire roll are not introduced into the body thus improving the vehicle‘s interior acoustics.

The best decoupling is achieved by slightly dampened sub-frame mounts. However, this leads to problems when the Eigen-frequency of the axle‘s sub frame system is stimulated by a harmonic of the wheel speed; because of the slight damping, substantial vibration amplitudes can occur in the axle system that will then be passed on with increased force into the body. Heavily dampened sub-frame mounts do indeed reduce the excesses of a resonance situation but this advantage comes at the price of severely reduced decoupling.

One solution for this is a hydraulic sub-frame mount which permits damping on a frequency selective basis only for the rigid body characteristic mode of the axle system – without significantly deteriorating the decoupling effect of the sub-frame mount. In the example shown, compare **Figure 8**, the rise of resonance was reduced by a factor six, just as if a highly dampened mount had been implemented. The decoupling is only reduced slightly with this solution whereby, with a highly dampened bearing, decoupling would be degraded by a factor of 7.5. This behavior can be confirmed in a test drive, both by measurement as well as subjectively. Simulation on multi-body models not only allows the identification of critical frequency for design of an optimally adapted hydraulic mount but also permits an estimation of decoupling reduction.

5 Component Development

The results of system layout serve as input values for the layout of individual components. Aside from the desired characteristics for stiffness and stiffness curves, the service life requirements of components must also be taken into consideration. The objective of component development at Vibracoustic is the smallest possible component which will still meet service life requirements with certainty.

5.1 Component Characteristics

Two things are required to transfer desired characteristics into a series-viable component structure: modern simulation software and a depth of expertise. The latter is needed to pre-select the components structure and rough dimensioning. Using Finite Element Analysis (FEA) the stiffness curves, the ratio of stiffness in the different directions of the mount and the shape of the rubber are designed in detail.

5.2 Service Life Requirements

The service life predicted by FEA has been successfully advanced for many years at Vibracoustic. The objective here is to achieve service life requirements in the vehicle already with the first prototype.

Initially load/time curves are converted into stress or strain/time curves using FEA to produce a durability prediction. Counting techniques, such as Rainflow counting, are used to classify the results which are in a next step linked to material characteristics. The stability of the material is represented in the form of Woehler curves or Haigh diagrams. The calculated damage factor provides the engineer with an indication of the component‘s service life certainty under real load conditions.

6 Verification

The functionality of newly developed components is tested in several stages. First, prototype components are assessed as described in Section 3. The results are then compared with the specifications of the component‘s layout. If the prototypes fulfill all requirements they are built into the test axle. The test axle is then used in a repetition of the trials described in Section 2. If the results correspond to the calculated potential estimate, the new components are built into a prototype vehicle. Subsequently, the success of modifications is evaluated in the vehicle, both subjectively as well as objectively by means of measurements.

7 Summary

To achieve improvements in driving comfort and interior acoustics it is not only necessary to improve individual components but also to address the vibration affecting the entire suspension system. This article describes the procedures that would be applied by Vibracoustic GmbH & Co. KG in the event that improvements to an existing axle were to be developed. The combination of experiments and simulation make it possible to proceed effectively (time and cost optimized) with the development of a new axle generation. ■

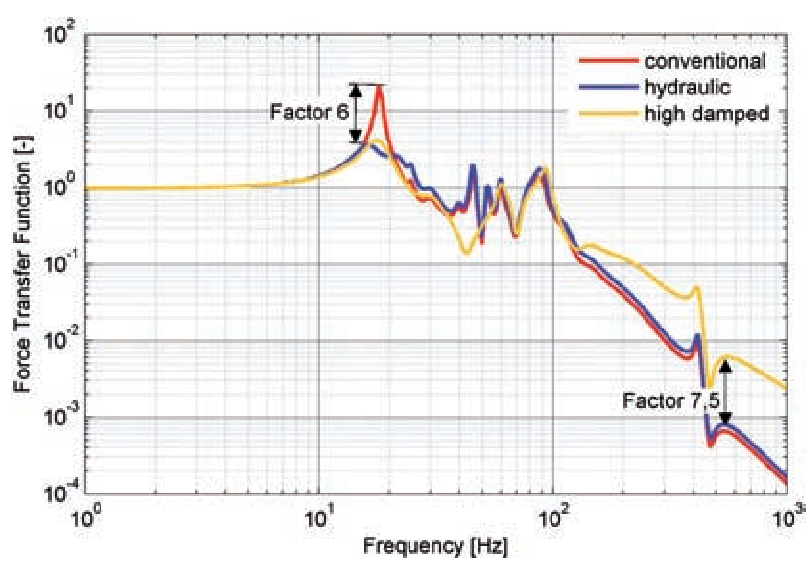


Figure 8: Transfer function of a hydraulic sub-frame mount in comparison with a conventional mount and a highly damped design