THE INFLUENCE OF SEMI-ACTIVE DAMPERS ON THE LOADS OF PASSENGER CARS

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Abstract Mechatronic suspension components in modern passenger vehicles affect the forces and moments acting on the car and its components. Since accurate load assumptions are mandatory when determining durability, it is important to investigate how controlled mechatronic suspension components influence the loads. Hence, vehicle measurements are performed with a car which is equipped with semi-active dampers. In addition, the loads on the vehicle have been studied in multi-body simulations. The results show that the vertical forces at the shock absorber towers as well as at the wheel centres change if different damper settings are applied. The investigations lead to the important result that the influence of semi-active dampers needs to be considered when determining fatigue life.

Keywords Fatigue life, load, mechatronic suspension component, multi-body simulation, semi-active damper

1 INTRODUCTION

The number of mechatronic components in modern car suspensions is increasing continuously. Many of these systems provide safety functions and assist the driver in critical situations. Furthermore, conventional car components are given new features by equipping them with electronic devices. Conventional vehicle dampers, for instance, can be expanded to semi-active systems in this manner. In comparison to customary dampers, the damper forces of semi-active systems are adjustable, i.e. they can be adapted to the driving situation. Thus, variable damping systems help to resolve the design conflict between handling performance and comfort. As a consequence, the best suspension setting can be found for any driving situation that occurs during operation.

In general, the durability of individual vehicle components is proved on test rigs before road tests are applied. Thus, it is possible to optimise single parts in a cost-efficient and time-saving way before testing the overall system. Traditional methods in laboratory fatigue testing do not account for the influence of mechatronic suspension components on the loads of passenger cars, though. However, many mechatronic suspension components influence the system performance in a way that also affects the vibration behaviour [1]. Changes in the loads of cars due to controllable suspension systems raise the following new questions:

- Does this influence need to be taken into account when determining durability?
- Is customer behaviour appropriately represented on fatigue test tracks?
- Do extrapolation coefficients need to be adapted?
- Can rig testing account for the influence of mechatronic suspension components?

This paper deals with the influence of semi-active dampers on the loads of passenger cars. Vehicle measurements and numerical simulations are conducted to identify the influence of different shock-absorber settings. Thus, it is possible to evaluate if rig testing methods need to account for the influence of semi-active dampers on the durability of passenger cars.

2 CONTROLLABLE DAMPERS

In the following, the terms used to define different types of vertical dynamic systems (passive, adaptive, semi-active and active) correspond to the classification given in [2].

A well-designed suspension should result in good handling characteristics as well as good driving comfort. The former is achieved by reducing the dynamic tyre forces. However, in order to provide adequate riding comfort, the vibrations of the car body need to be minimized. Furthermore, medicinal reasons demand a limitation of the car body oscillations to avoid damage of human organs [3]. The above mentioned goals result in a design conflict [4].

In suspension systems with conventional *passive* springs and dampers, the force-displacement characteristics and the force-velocity characteristics are defined by constant curves [2]. Thus, tuning of the vehicle suspension travel response

is limited to modifications of these two curves. High damping rates result in good handling characteristics, whereas riding comfort deteriorates. Therefore, the final suspension set-up always is a compromise.
Mechatronic components, such as controllable dampers, resolve or at least alleviate this design conflict [5]. Consequently, *adaptive* systems have been introduced in order to expand the possibilities of tuning the chassis set-up. These systems enable switching between different curves during operation of the vehicle. In comparison, the characteristics of *semi-active* systems are defined by diagrams. Furthermore, any operating point can be reached within milliseconds. However, semi-active components cannot supply the system with energy – they only dissipate energy. In contrast, *active* systems have an external power source. Considering an active damper system, this means that the direction of the damping force can be reversed independently from the current suspension travel direction.
In this article, a semi-active damper is considered which is based upon a conventional hydraulic car damper. However, the size of the flow cross-section can be adjusted during operation by a controllable valve. Thereby, the energy dissipation can be adapted during operation. As mentioned above, the velocity-displacement behaviour of semi-active dampers is described by characteristic diagrams instead of constant curves. This is shown in Figure 1.

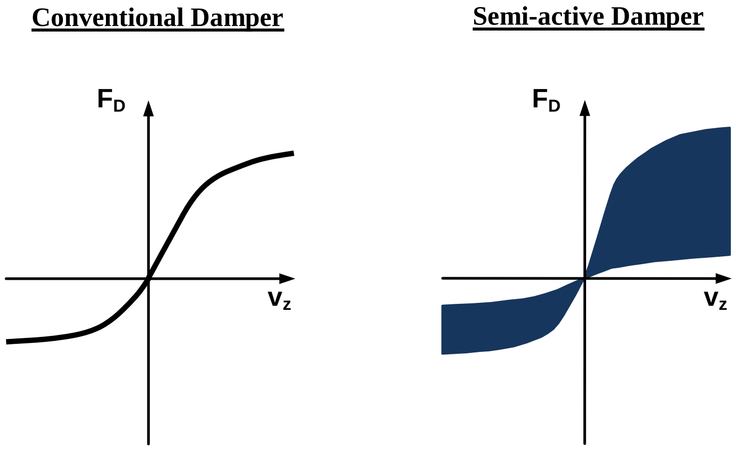

Figure 1 - Force-velocity curves of a conventional damper (left) and a semi-active damper (right)

The electronic control unit (ECU) of the damper controls the damper setting by taking into account the setting chosen by the driver, the road surface as well as the driving situation (see Figure 2). By selecting one of three different available settings of the semi-active dampers (comfort, normal and sport), the driver defines the baseline level of the force-velocity diagram. Selecting the sport setting, the available characteristic diagram is more limited compared to the comfortable setting, i.e. the minimum value of the absolute damping force is higher compared to the comfort setting.
Furthermore, the ECU analyses the driving situation and the road condition in order to reduce roll and pitch motion. By adapting the electric current of the controllable valve, the damping rate is adjusted within milliseconds. In situations where handling performance is the major requirement, the damping rate is high. When travelling on a straight and even road, the damping rate is low in favour of driving comfort.

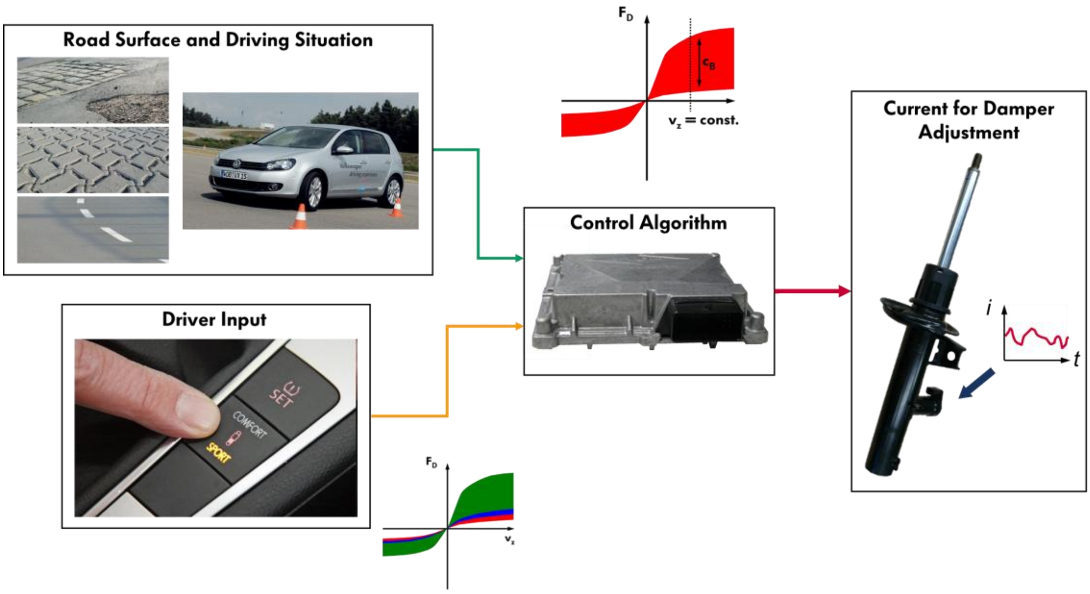

Figure 2 - Control strategy of the semi-active damper

3 NUMERICAL SIMULATION IN FATIGUE LIFE PREDICTION

Multi-body simulations are used to determine the loads acting on cars and their components [6]. Calculations of test track measurements are state of the art. Through simulation studies it is possible to evaluate the loads in an early state of the design process before hardware prototypes exist. Furthermore, forces and moments acting on joints and bushings can be calculated. These cannot be determined in an economic way during vehicle measurements. Moreover, testing procedures are developed and optimized with the help of multi-body simulations [7], [8]. Thus, it is possible to check the technical feasibility of rig tests beforehand.

Numerical calculations have already been successfully applied to determine dynamic loads which occur during anti-lock braking (see for example, [9]). Different methods for coupling or integrating multi-body systems and controller systems in simulation scenarios exist [10]. In so-called tight coupling, the differential equations which describe the multi-body system as well as the equations of the controller model are solved by the same programme. This means that the controller model needs to be integrated into the multi-body model or the other way around. On the contrary, weak coupling uses different solvers for the equations of the multi-body system and the controller. Hence, procedures for data exchange and simulation control need to be implemented.

4 METHOD

The influence of different damper settings on the loads is studied using a series-production passenger car equipped with semi-active dampers. Vehicle measurements as well as numerical calculations are used to determine changes of the vertical forces. Furthermore, conventional dampers are installed in the same car. Thus, it is possible to compare the loads on the conventional chassis to the forces on a suspension system with semi-active dampers. In this manner, it can also be investigated if existing procedures for durability prediction need to be adapted when considering mechatronic suspension components.

In order to estimate the loads which occurred during vehicle measurements, simulations with a quarter-car model are applied. Then, a passenger car is equipped with sensors and measurement devices to perform vehicle measurements. Thereafter, a model for multi-body simulations is set up and calibrated using the measurement results (see Figure 3).

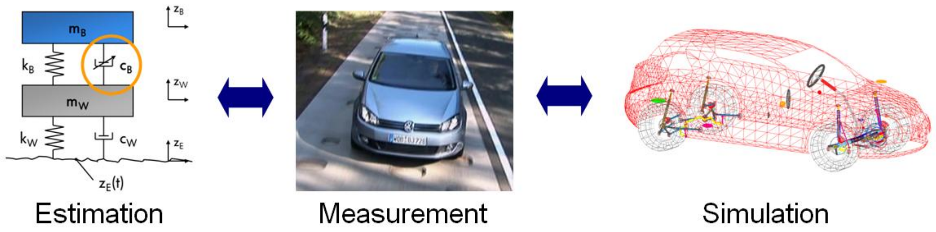

Figure 3 - Illustration of the method for determining the influence of semi-active dampers on the vertical loads

4.1 Estimation

In order to estimate the possible influence of the semi-active suspension system on the loads of the vehicle and its components, a quarter-car model is studied (see Figure 4). The vertical vibration behaviour of the car is modelled using the wheel mass m_W and the car body mass m_B. The former represents the unsprung mass whereas the latter represents the sprung mass of the vehicle. For the tyre stiffness k_W and the tyre damping c_W constant values are used. Furthermore, the suspension stiffness k_B and the suspension damping c_B are included in the model. The suspension stiffness is set to a constant value. However, the suspension damping of a car with semi-active dampers varies. In first calculations, constant values for the minimum and maximum possible damping are used.

The quarter-car model is described by the equations of motion [11]

$$m_B \cdot \ddot{z}_B = -c_B \cdot (\dot{z}_B - \dot{z}_W) - k_B \cdot (z_B - z_W) \tag{1}$$

and

$$m_W \cdot \ddot{z}_W = -c_B \cdot (\dot{z}_W - \dot{z}_B) - k_B \cdot (z_W - z_B) - c_W \cdot (\dot{z}_W - \dot{z}_E) - k_W \cdot (z_W - z_E) \; . \tag{2}$$

The kinematic road excitation z_E is represented by measurement signal input from a comparable car. Furthermore, sinusoidal excitations with different frequencies and amplitudes are used. The simplified model description is sufficient for basic estimations of acceleration and force magnitudes during vehicle measurements.

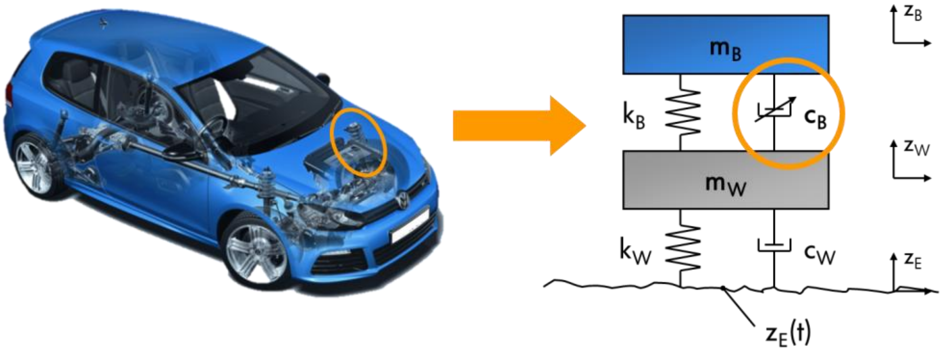

Figure 4 – Quarter-car model

4.2 Measurement

In order to evaluate the influence of semi-active dampers in vehicle measurement, the car is equipped with sensors measuring accelerations, forces, strains and suspension travel. The measurement data contain all the information which is usually required for full vehicle rig testing. The sensors are located on the front and on the rear vehicle axle system as well as on the car body, as shown in Figure 5. Furthermore, the forces and moments at the wheel centre are measured by means of measuring wheels. Due to the choice of measuring positions it is possible to analyse the distribution of forces.

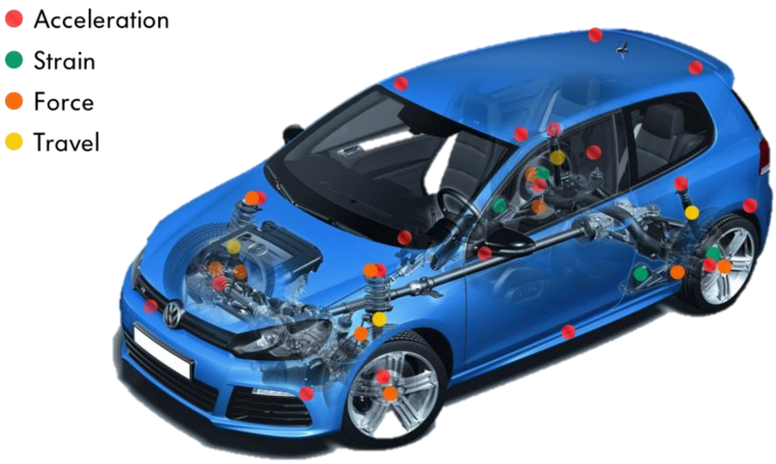

Figure 5 - Sensor positions for vehicle measurement

In addition to the measurement data describing the mechanical system, communication information of the vehicle is recorded. CAN-bus messages such as steering angle, brake demand and throttle demand are used for plausibility checks of the measurement signals. By means of a development ECU, input, output and internal variables are recorded.
The vehicle measurements are performed on test tracks as well as on standard roads. Thus, the influence of different road surfaces as well as the influence of various driving conditions can be studied

4.3 Simulation

The influence of semi-active dampers on loads of the car and its components is studied in a multi-body model. Tight coupling is used in order to represent the ECU, i.e. the equations which describe the controller algorithm are integrated in the multi-body car model. The software code of the model corresponds to the software on the controller unit of the test car. Moreover, measurement results are used to review the ECU signals and to calibrate the multi-body car model. The damper forces are represented through a three-dimensional spline originating from damper measurements. Thus, it is possible to calculate the damping forces dependent on the damper valve current as well as the relative motion between the damper piston and the cylinder. Furthermore, a digital model of the test track is available. In numerical simulations, it is possible to study the influence of different damper settings on the loads of the car independently from the driver behaviour.

4.4 Evaluation

Since the damper characteristics mainly influence the vertical movements of the vehicle, this direction is considered in the analysis. Another advantage is the fact that this direction is nearly unaffected by the driver who has a strong influence on the longitudinal and lateral forces. In order to minimize changes in road profile and the velocity of the car, previously defined velocities are used during test track measurements. The measurement signals and the simulation results are analysed according to the same procedure shown in Figure 6.

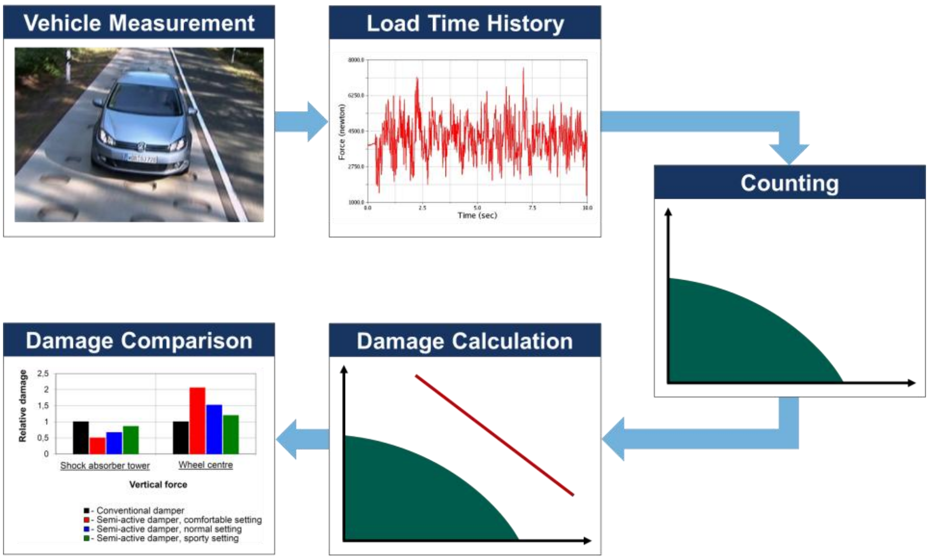

Figure 6 - Evaluation procedure

The load time history data originating from vehicle measurement and simulation are classified by Range Pair Counting. After that, a virtual S-N curve is set up using values according to a standard procedure. The idea is not to represent the real strength of the considered component. Instead, the curve is only used to compare the damage of different load time histories. Therefore, the used gradient of the S-N curve is an approximation value which could be applied to different car components.
Afterwards, a damage calculation according to the Miner Elementary Method is performed. The resulting damage values are compared to the damage values calculated from measurements with conventional dampers.

5 RESULTS

The vertical forces of the car which have been determined during vehicle measurements as well as simulations are dependent on the setting of the semi-active damper. The results are shown in Figure 7. In the following, the results of the loads on the wheel centres and on the shock absorber towers are discussed. The former represent the forces acting on the unsprung mass whereas the latter are the forces on the sprung mass of the vehicle.

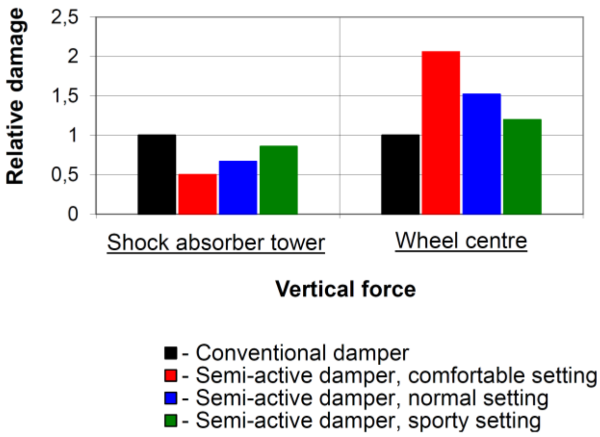

Figure 7 - Relative damage of vertical forces measured at the shock absorber tower and the wheel centre for different damper settings on a test track

Comparing the different settings of the semi-active damper, the biggest forces at the shock absorber towers are observed in the sporty setting. Lower damping ratios cause smaller vertical loads on the car body. However, the vertical wheel forces increase when lowering the damping. Thus, the biggest vertical loads on the wheel centres occur in the comfortable setting.
These results are in accordance with the results obtained from the quarter car model (Section 4.1) and simulations of the multi-body car model (Section 4.3).

6 CONCLUSIONS

The analysis of measurement results reveals that the influence of semi-active dampers needs to be considered when determining the durability of a vehicle. Multi-body simulations of a car model with integrated ECU (tight coupling) have been applied successfully to study the influence of different damper settings on fatigue life. Thus, it is possible to estimate the loads on a passenger car in an early state of the design process with the help of numerical calculations. In addition, the design of components as well as durability calculations can be improved.
The focus of the project presented in this article is on vehicle measurement and simulation. However, it is important to extend research also to rig testing procedures (see Figure 8). Therefore, testing methods which allow for the investigation of the influence of different damper settings in laboratory durability testing will be studied with the help of multi-body simulations.

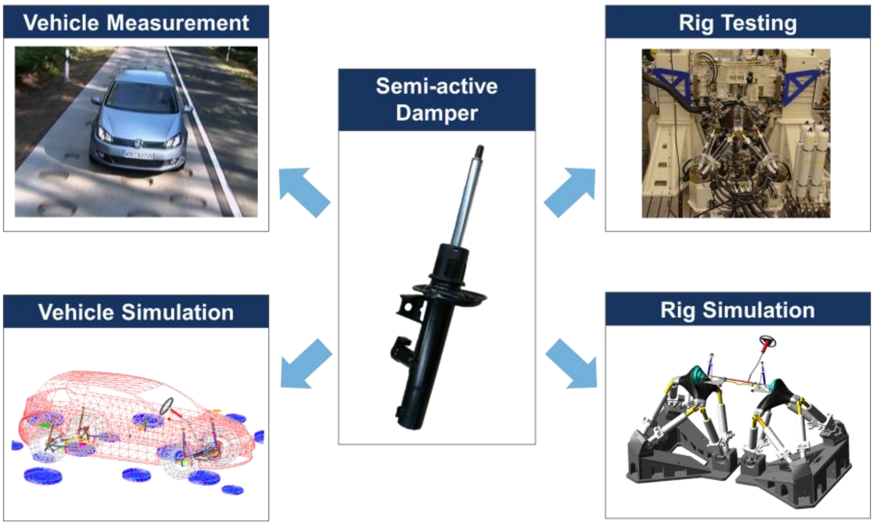

Figure 8 - Semi-active dampers in durability proving

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