Modernisation of a test rig for determination of vehicle shock absorber characteristics by considering vehicle suspension elements and unsprung masses

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Abstract. This paper presents a modernization approach of a standard test bench for determination of damping characteristics of automotive shock absorbers. It is known that the real-life work conditions of wheel-suspension dampers are not easy to reproduce in laboratory conditions, for example considering a high frequency damper response or a noise emission. The proposed test bench consists of many elements from a real vehicle suspension. Namely, an original tyre-wheel with additional unsprung mass, a suspension spring, an elastic top mount, damper bushings and a simplified wheel guiding mechanism. Each component was tested separately in order to identify its mechanical characteristics. The measured data serve as input parameters for a numerical simulation of the test bench behaviour by using a vibratory model with 3 degrees of freedom. Study on the simulation results and the measurements are needed for further development of the proposed test bench.

1. Introduction

Properties of automotive dampers are fit to the customer's requirements, by adapting the valve configuration, choosing the right damper elements dimensions, the hydraulic oil properties, the gas pressure and some other parameters [8]. There is no simple method of evaluating overall damper performance on the bench with good correlation to vehicle test results [9]. Bench rig like in Figure 1 is current automotive industry standard for damper performance evaluation. The test is usually conducted with a sine-wave excitation at constant stroke and different frequencies. Peak forces measured during subsequent tests are used to plot force vs. velocity diagrams used to describe the damper.

The test bench consists of a frame and a force sensor which is mounted to the beam. In the lower part a hydraulic ram is fixed to the bottom. Its cylinder movement is numerically controlled which allows to move the test object with piston displacements and piston velocities defined by the user. The limiting values of operation are given by the utilized hydraulic pumps. Another option to force a displacement of the tested object is to use a driving shaft which can realize different piston velocities but with quasi-sinusoidal profile and constant stoke.
Equipped with a measurement system, in detail the mentioned force sensor, displacement sensor and/or velocity sensor, damper characteristics can be obtained, similar to these presented within Figure 2 and Figure 3.

Figure 2 shows the damping force as a function of the piston displacement. The maximum force during the compression phase is about 500 N whereas the rebound force is nearly three times greater at about 1450 N at the greatest stroke velocity. An alternative is shown in Figure 3 where the damping force is presented as a function of the piston velocity.

This testing procedure only describes the properties of the dampers, but installed in a vehicle the whole suspension system together with unsprung masses let dampers work in a different way [2]. Especially on rough roads where the piston displacement is small and the damper can generate a noise which deteriorates the ride comfort.

The function of the shock absorber is to damp both sprung and unsprung resonance [5,6]. In terms of vehicle characteristics, its function is to damp body roll motion when the vehicle is rapidly steered, to damp body pitch motion when the vehicle is rapidly accelerated or decelerated, and to damp unsprung mass vibrations for ensuring the tyre-ground contact while driving on undulated roads. On the other hand, smaller damping forces can improve ride comfort while driving on a smooth road and reducing ride harshness while passing over a road bump. For instance, driving on a rough road insists high damping forces within the compression phase and tension respectively. In contrast, handling manoeuvres, like a lane change, do not require such high damping forces.
The main idea is extending the existing damper test bench (Figure 1) with a tyre-wheel and a guiding arm, imitating a vehicle suspension, joined with a damper module composed of a shock absorber, a coil spring, a spring aid, elastic top mount and bushings. The designed test bench should fulfil additional criteria, like easy change of the suspension parameters, simplicity, limited space taken. In order to better understand the suspension behaviour at the test rig and to provide information about boundary conditions for the real life tests, a dynamic model of the vibrating system with 3 degrees of freedom is formulated by using MATLAB/SIMULINK.

2. The considered wheel suspension system

The rear, undriven suspension of a Mini Morris Cooper S was considered as the research object. It consists of a trailing arm consisting the wheel hub and the attachment for the spring-damper module. The trailing arm is constrained by two lateral links. This kind of suspension mechanism belongs to the kinematic group of spherical mechanisms, where the sphere centre is located at the trailing arm front [7].

![Figure 4. Measurement of unsprung masses on the rear wheel suspension of a Mini Morris Cooper S.](image)

An overview of the measured elements, which were carried out at experimental test benches, is given in Figure 5.
Starting at the top of Figure 5 there are two elastic elements (top mount bushings) which are located at the upper and lower side of a disk and have an initial preload caused by the assembly. The disk is mounted to the car body. The upper element springs into action during a rebound and the lower element during a bump event. The next considered part of the damper module is the so-called spring aid. If the damper deflection in compression stroke is greater than a given value the spring aid comes into action. It works as a parallel spring to the main suspension spring increasing the net suspension stiffness. Next, the twin-tube damper and the coil spring. The last element used within the damping module is the lower bushing. All these described elements have been disassembled and measured at special test rigs. The obtained readings give information about the reaction forces as result of a quasi-static deflection.

Until now, only the elements which are in connection with the damper module have been considered. To imitate a vehicle suspension some more aspects are important. Also the pneumatic tyre of the 205/45R17 84V (Bridgestone Potenza with Run Flat), inflated with 2.6 bar, is taken into account. The tests were carried out in the lab [4] at the Institute of Automobiles and Internal Combustion Engines at Cracow University of Technology.

The measured mass on single wheel (250 kg) can be divided into the unsprung and the unsprung masses. To differentiate between them the suspension was prepared for separate measurements like presented in Figure 4. One step was to detach the anti-roll bar. Next, the damper module was disconnected from the trailing arm. This allowed to measure the unsprung suspension mass which consists of: brake disk, screws, wheel bearing, moving part mass of the trailing arm, break calliper which includes the brake fluid, the brake pads and the hydraulic pipes. Separately the wheel mass was measured, which consists of: rim and tyre.

To sum up, the goal was to obtain information about elements installed within the tested car suspension. Different methods have been considered. The easiest and quickest way is to make use of data sheets, but often the accessibility is poor which does mean that tests have to be carried out at test rigs. In this paper, readings were made on machines which are quasi-static or by making use of dynamic test rigs. The stiffness is described as a function of displacement or velocity respectively. An additional possibility to obtain information is to make use of non-commercial instruments like a scale, weights and a measuring instrument. This is an easy and cheap way to gain information about elements but is connected with a long preparation time.
All obtained characteristics of each car suspension component will be used within a numerical calculation. The next chapter will describe approaches of modelling a quarter-car model to obtain more realistic information about an extended and the newly designed guiding arm.

3. Mechanical models for vertical vibrations analysis
A numerical model has been created which takes into account suspension masses and characteristics of elastic and damping elements. The following models were assumed.

4-DOF quarter car model with longitudinal arm
A so called quarter car model with four degrees of freedom for analysis of vertical vibrations in the wheel suspension system, is presented in Figure 6. Due to the fact that the wheel and the damper module are not assembled in one axis, which does mean that that there is a ratio between this elements at the trailing arm, this has to be taken into account within the calculation.

The suspension ratio, defined as relation between the wheel vertical motion and the spring-damper deflection, is equal to 0.7. This does mean that the trajectory is different which effects that the vertical displacement or the difference \( z_2 - z_1 \) is not correct within the calculation. Respecting the ratio, the proper displacement \( z_{1r} \) has to be taken:

\[
z_{1r} = 0.7 \cdot l \cdot \sin(\alpha)
\]  

(1)

4-DOF quarter car model
The 4 DOF model allows to execute a more precise analysis by considering all characteristics and elements, as well as to estimate the error made by testing on a test rig without a sprung mass, which is not possible within a rigid frame at the test bench due to the available space. This simulation can be carried out by adapting the vibratory system as shown in Figure 7.

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Figure 6. Geometry of the wheel suspension with longitudinal arm.
3-DOF quarter car model

The absence of the sprung mass and attaching the top mount to the force sensor changes the system. Thereby $z_4$ equals zero. The equations of the oscillatory system are as follows:

\[ \ddot{z}_1 = \frac{c_B}{m_1}(z_2 - z_{1r}) - \frac{c_K}{m_1}(z_1 - s_B) \]  \hspace{1cm} (2)

\[ \ddot{z}_2 = -\frac{c_B}{m_2}(z_2 - z_{1r}) + \frac{c_G}{m_2}(z_3 - z_2) + \frac{d_G}{m_2}(\dot{z}_3 - \dot{z}_2) \]  \hspace{1cm} (3)

\[ \ddot{z}_3 = \frac{c_T}{m_3}(-z_3) - \frac{c_G}{m_3}(z_3 - z_2) - \frac{d_G}{m_3}(\dot{z}_3 - \dot{z}_2) \]  \hspace{1cm} (4)

These equations describe the movement of the suspension. Solving the equations the forces of each structural element can be obtained, which is an input for designing the test rig elements.

**Table 1.** Mass parameters of 4-DOF quarter car model.

| $m_1$ | 37 kg | Unsprung mass |
|-------|-------|---------------|
| $m_2$ | 2 kg  | Lower valve, inner and outer tube, lower bushing |
| $m_3$ | 1 kg  | Piston rod, piston, valve |
| $m_4$ | 213 kg| Sprung mass (car body part) |
4. Design of the proposed test rig

By knowing the masses and dimensions of all components, as well as the movement obtained by the simulation, the new elements of the modernised test rig have been designed. Overviews are shown in the following figures.

An existing test rig (Figure 9) which consists of a frame, a force sensor at the top and a hydraulic ram with a base plate at the bottom has been extended by a guiding arm, shown in Figure 8. This simplified suspension considers all geometrical ratios, all masses, real elements of the suspension strut and the real wheel of the vehicle.

![Figure 8. Test rig in initial position and sensors layout for extended damper testing.](image1)

![Figure 9. Final test rig assembly.](image2)

Before a test run was started, the test stand had to be equipped with measuring equipment. The amount and the types of available sensors have been:

- 6 acceleration sensors,
- 2 displacement sensors (at damper module and hydraulic ram),
- 1 force sensor (at the damper rod),
- 1 force sensor (at the test rig).

The signal of the base plane accelerometer A4 (Figure 8) provided information about the excitation frequency. The frequency was also obtained by reading the displacement measured by the test rig controller but to make a cross-check this information was also gained by acceleration sensor A4.

To obtain the damper characteristic, or more precisely the damping force as a function of the rod velocity, which is dependent on the frequency and excitation displacement, sensors D1 and F2 were used. The velocity is not measured directly but calculated by numerically differentiating the displacement signal.

Signals gained with sensors F1 and D2 are used to compute the transfer function from the excitation displacement up to the net module force, which is transferred up to the imitated car body attachment.
The dependency between signals gained by sensors A2 and A3 provided information about the suspension ratio. Accelerations measured with sensors A1 and A2 are used for NVH analysis of the damper module.

5. Damper characteristics obtained from measurements

The measured characteristics of the mean piston displacement as a function of excitation frequencies for different amplitudes (2, 5 and 10 mm) of sinusoidal excitation, are presented in Figure 10. Due to the fact that the car body is substituted by the fixed test rig frame, the unsprung mass can move only. The obtained characteristics is typical for a nonlinear system, where the unsprung resonance frequency depends on the excitation amplitude. In conditions of higher frequencies and amplitudes the tyre starts to loss contact with the exited base. This phenomenon can be used to compare different characteristics of the dampers.

The values of the piston displacement multiplied by the suspension ratio give approximated the mean displacement of the oscillating unsprung mass. Due to the unsymmetrical damper characteristic (the force during rebound is nearly four times greater than the force during bump motion) the displacement drifts from the initial position at 0.

![Image](Tyre_inflation_pressure_2.6.png)

**Figure 10.** Mean damper displacement D1 vs. excitation frequency.

A comparison of the standard deviations of the two measured reaction forces (F1, F2) can also be suggested for evaluation purposes (Figure 11). Recognizable is the fact that a deviation of the net force (F1) at the test rig differs from a deviation of the damping force (F2), especially at the resonance frequency.

![Image](Standard_deviation_of_reaction_forces_F1_F2.png)

**Figure 11.** Standard deviation of reaction forces F1, F2 vs. excitation frequency.
6. Validation of the numerical model
Based on measured data the numerical model (3-DOF, chapter 3) can be validated. For a first comparison a frequency of 1 Hz, a peak amplitude of 2 mm and the nominal tyre pressure is defined. The data obtained by the measurement is printed blue whereas the simulated results are plot in red colour within Figure 12.

![Figure 12](image1.png)

**Figure 12.** Measurement (blue) vs. simulation (red); Force F1 vs. Displacement D2.

At low excitation frequencies the controller of the hydraulic ram generates overlaid oscillations with a higher frequency. The irregularity within the measured signal is recognizable. Comparing the two signals similar displacements and reaction forces at the test rig sensor can be observed.

![Figure 13](image2.png)

**Figure 13.** Measurement (blue) vs. simulation (red); Force F2 vs. Displacement.

The simulation provides a similar damper characteristic (damper force as a function of damper displacement) as presented in Figure 13.

Unfortunately the simulation model provides differing results at higher frequencies than 4 Hz. A reason is that the characteristics of the compliant elements have been estimated with a quasi-static method. The high dynamic of this oscillatory system would require dynamic characteristics like reaction forces as a function of displacement and frequency. Additionally a more advanced damper model should be used.

7. Concluding remarks and further steps
This paper presents an approach of modernizing a standard shock absorber test bench. The test mechanism consists of a guiding arm with an additional unsprung mass, the wheel and other compliant suspension components fixed to a simplified suspension. Measurements of car suspension elements were carried out which serve as input parameters for a numerical simulation and for the design process, respectively. With the help of this enhanced test rig NVH analyses can be carried out which
can be compared to standard damper tests and real life application or to compare dampers of the same type but with different characteristics.

Initial information about the system response was captured using formulated multi body model analysis. The simulation results were based on solution of three, nonlinear, inhomogeneous, ordinary differential equations, which were implemented in Simulink/Matlab. Effects like change of mean position of the unsprung body mass caused by the unsymmetrical damper characteristic can be observed already before the real test mechanism is built.

Designing a test mechanism, which should fit to an existing damper test rig, must fulfil some criteria. For instance, the friction between the moving parts should be as less as possible. Standard components should be used to reduce the amount of parts which have to be newly machined. The assembling of the mechanism should be as quick and simple, as possible. And finally, the geometric ratio of the suspension mechanism should be considered. During assembling and attaching the test mechanism to the test rig it turned out that all parts are easy to install and give good possibilities of adjustment. Additionally, the guiding arm withstood all the excitations, even those near to the resonance frequencies.

The carried out measurements required lot of preparation time. The most complicated measuring element to install was the displacement sensor, but combined with the additional force sensor, it provided very important information about the damper characteristics and the oscillating system. The measurement gave additional information about the ability of the test rig hydraulic system how it realizes different excitation frequencies and displacements. In the range of higher frequencies and lower amplitudes of excitation, special focus should be laid on a background noise coming from the power hydraulic unit. Lateral slip and vibrations of the tire with respect to base plate, and collisions of tire side wall with suspension spring, occurred at the highest amplitudes, what led to additional disturbances.

For future analyses, based on the gained data, some parameters can be defined to evaluate the damper behaviour by comparing different shock absorbers at this test mechanism. As refinement for future projects a compensation of the unnatural unsprung mass movement, which already leads to tyre contact loss at higher frequencies and amplitudes, can be considered. Additionally, emphasis should be laid on the elastic bushing behaviour. The bushing reactions at higher frequencies are different to the quasi static measurements and herewith also the modelled system behaviour. Additional measurements should be carried out to obtain a better correlation between measurement and simulation results. Measurements with microphone should also be planned to find correlation between the system mechanical response and its noise emission.

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