M. Speckert, K. Dreßler, H. Mauch

MSB Simulation of a hexapod based suspension test rig

Vorwort

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Darüber hinaus bietet die Reihe ein Forum für die Berichterstattung über die zahlreichen Kooperationsprojekte des Instituts mit Partnern aus Industrie und Wirtschaft.

Berichterstattung heißt hier Dokumentation des Transfers aktueller Ergebnisse aus mathematischer Forschungs- und Entwicklungsarbeit in industrielle Anwendungen und Softwareprodukte – und umgekehrt, denn Probleme der Praxis generieren neue interessante mathematische Fragestellungen.

Prof. Dr. Dieter Prätzel-Wolters
Institutsleiter

Kaiserslautern, im Juni 2001

MBS Simulation of a hexapod based suspension test rig

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Summary:

Testing a new suspension based on real load data is performed on elaborate multi channel test rigs. Usually wheel forces and moments measured during driving maneuvers are reproduced on the rig. Because of the complicated interaction between rig and suspension each new rig configuration has to prove its efficiency with respect to the requirements and the configuration might be subject to optimization.

This paper deals with modeling a new rig concept based on two hexapods. The real physical rig has been designed and meanwhile built by MOOG-FCS for VOLKSWAGEN.

The aim of the simulation project reported here was twofold: First the simulation of the rig together with real VOLKSWAGEN suspension models at a time where the design was not yet finalized was used to verify and optimize the desired properties of the rig. Second the simulation environment was set up in a way that it can be used to prepare real tests on the rig.

The model contains the geometric configuration as well as the hydraulics and the controller. It is implemented as an ADAMS/Car template and can be combined with different suspension models to get a complete assembly representing the entire test rig. Using this model, all steps required for a real test run such as controller adaptation, drive file iteration and simulation can be performed. Geometric or hydraulic parameters can be modified easily to improve the setup and adapt the system to the suspension and the load data.

Keywords:

Test rig, MBS simulation, suspension, hydraulics, controlling, design optimization

1   Introduction

In automotive industry complex multi channel servo-hydraulic test rigs are used for physical testing of suspensions. Typically, wheel forces measured on a test track using wheel force transducers (target loads) are to be reproduced on the test rig. Since tests on the track as well as on the rig should be as short as possible, these loads are high with respect to amplitude and frequency thus posing high requirements on the testing equipment. Although the suspension, when mounted on the test rig, behaves a bit different than the suspension in the car on the test track, the quality of test on the rig is related to a good accordance of the wheel forces measured on the track and the ones observed on the rig.

Thus the most important requirement for a test rig is the capability of applying the target loads to a suspension in the laboratory. Since a wheel hub in general has 6 degrees of freedom (DOF) it is natural to design the testing system in such a way, that one hydraulic actuator is dedicated to one DOF i. e. an actuator for the vertical force, one for the longitudinal force etc. Most of the traditional rigs are based on this approach. Such systems however need a rather big amount of additional components such as lever arms for the application of moments to perform the test. If steering has to be supported, the entire set of actuators has to be moved to follow the wheel.

A new concept for suspension test rigs can be based on the hexapod technology as shown in figure 1.

Here six actuators are driving a common platform (parallel kinematics) thus supporting all 6 DOFs of the wheel hub [3]. Steering can be taken into account by a simple additional actuator for the steering wheel or the steering rod. The amount of mechanical components needed for the concept is restricted to the actuators and the platform. The symmetry of the construction makes it possible to use identical actuators, which is an additional advantage.

In 2004 Volkswagen decided to introduce this concept into its suspension-testing environment. The hexapods of the new test rig have been developed by MOOG-FCS in the Netherlands. A project has been initiated to set up a simulation environment for the new testing system in order to accompany the introduction of the system, optimise the design and give support during the preparation of future suspension tests on the system.
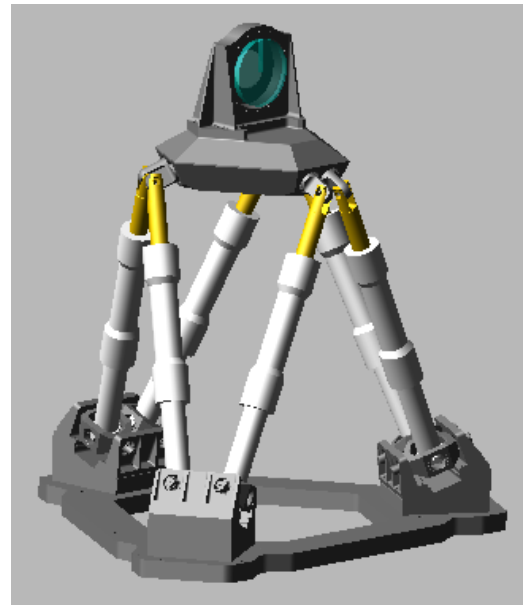

Fig. 1: The hexapod concept

The main goals of the project are described in section 2 of this paper. Section 3 contains a description of the suspension models, which have been used during the project, and section 4 explains the new test rig model in detail. Finally, section 5 contains the application of the model to the tasks described next.

2 Goals of the project

The virtual model of the system should be capable of simulating an entire physical test including the hydraulics and the control mechanisms. To reach this goal, the system is divided into subsystems, namely the suspension model (M1), the mechanics of the hexapod (M2) and a model for the hydraulics and the controlling (M3). These subsystems are assembled to the entire testing environment as sketched in figure 2. The subsystems M2 and M3 have to be instantiated for both sides of the suspension.
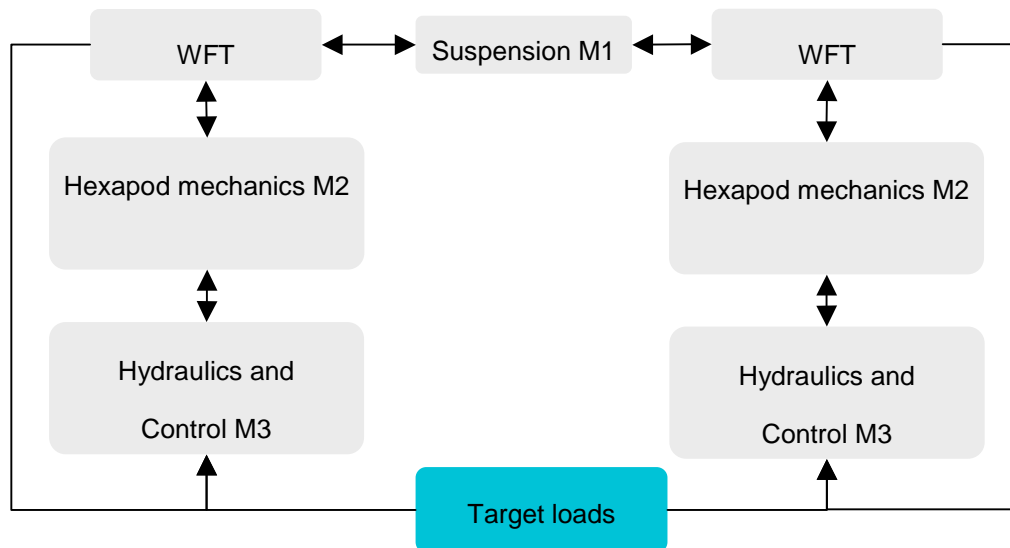

Fig. 2: The model of the entire system and its parts

M2 and the hydraulic part of M3 have been implemented as ADAMS/Car templates. The controller can be attached to M3 via user subroutines. Co-simulation with the controller in MATLAB/Simulink is also possible. The templates can be used to create assemblies with arbitrary suspension models existing as templates in ADAMS/Car. The suspension models used in this project have been taken from existing and elasto-kinematically validated databases at Volkswagen.

Using this environment a complete suspension test including iteration of drive signals can be simulated. In this paper the application of the model to the derivation of the requirements for the actuators and the improvement of the first design of the hexapod is described. To this end, only the "mechanical part" M2 of the hexapod has been used. Additionally some simulations with the complete assembly have been done and compared to data recorded on a physical prototype at FCS.

3 The suspension models

A front and a rear suspension have been chosen for this project. The front suspension is based on the McPherson concept and includes a stabilizer and a steering subsystem. The drive train has not been considered. The stabilizer is modelled using ADAMS beam elements. All other components are modelled as rigid bodies connected to each other via non-linear bushing elements. An ADAMS-test rig developed at Volkswagen can be used to create an assembly of the suspension and simulate the suspension based on 6 forces and moments for each wheel hub, see also [1] and [2].
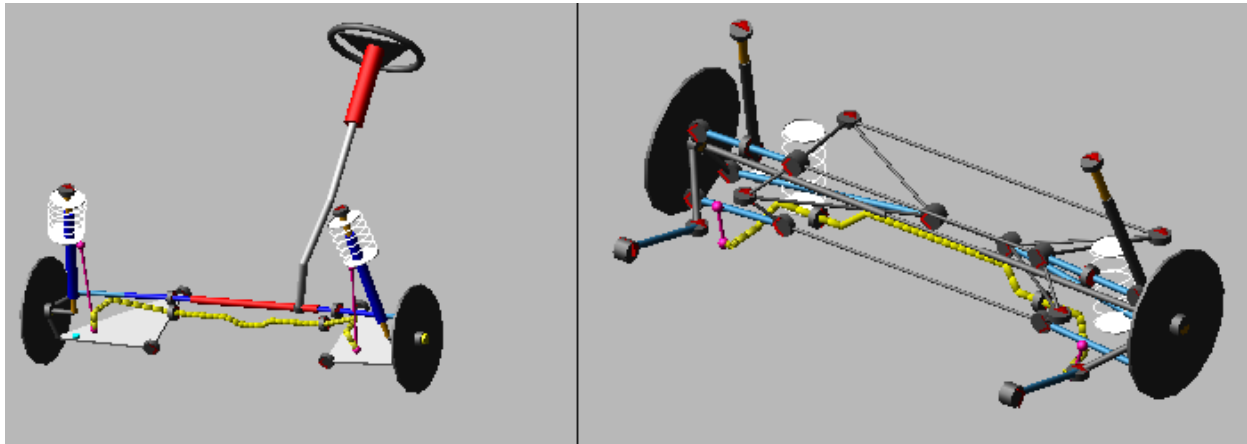
Fig. 3: Sketch of the suspensions used in the paper: to the left the front suspension including steering device and to the right the rear suspension

The model of the rear suspension is build similarly. The stabilizer consists of beam elements, all other components are rigid parts connected to each other via non-linear bushings or joints. Again, the drive train is not considered.

The loads for the simulations are taken from measurements on a test track containing bad road profiles, curves and braking events. Vertical displacements and the accelerations of the wheel hub in the simulations have been calculated and compared to the measurements in order to check the quality of the suspension model. Some modifications of the damping parameters have been done to achieve a better agreement between measurement and simulation.

For the application of the results to the hexapod test rig, the acceleration of the wheel hub are very important. The following table contains the extreme values of the accelerations in m/s^2 as measured on the test track.

Wheel	direction	Minimum	Maximum	RMS
rear left	Longitudinal	-366	192	14
rear right	Longitudinal	-394	142	14
rear left	Lateral	-100	112	9
rear right	Lateral	-100	100	9
rear left	Vertical	-225	323	38
rear right	Vertical	-234	315	38
front left	Vertical	-232	308	28
front right	Vertical	-219	285	30

Table 1: Measured maximum accelerations in m/s^2 of one lap of the proving for the front suspension and a shorter segment for the rear suspension

The following figures contain some examples of the simulation results. As indicated, the simulation results have been low pass filtered to suppress numeric effects.
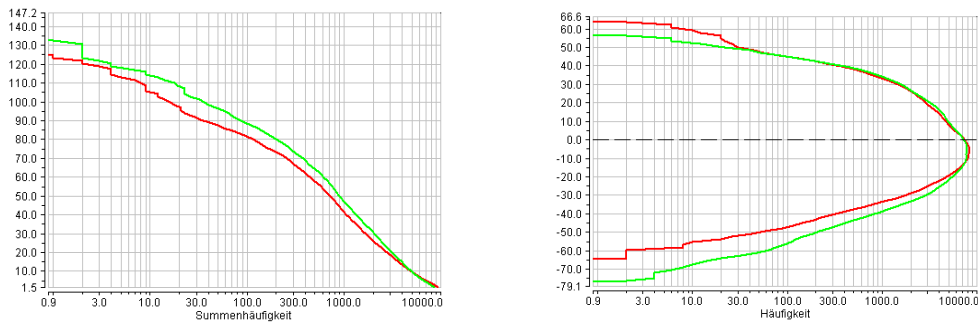
Fig. 4: To the left range pair to the right level crossing diagram for the measured (---) and the simulated spring displacement (---) in mm (right front wheel, one lap on the proving ground).
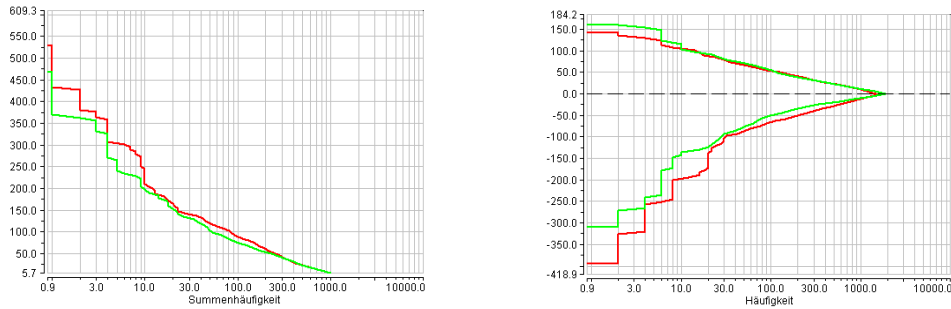

Fig. 5: To the left range pair to the right level crossing diagram for the measured (---) and simulated (---) longitudinal acceleration (right wheel of rear suspension, 82 seconds of the proving ground lap). The longitudinal displacements have been 80 Hz low pass filtered in order to suppress numeric effects.
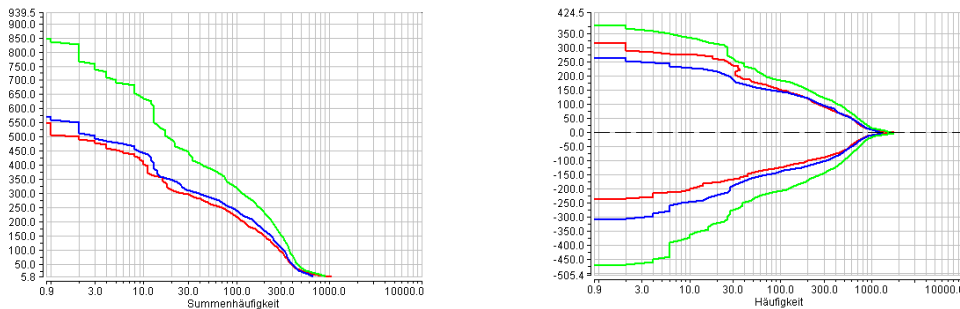

Fig. 6: To the left range pair to the right level crossing diagram for the measured (---) and two simulated vertical accelerations (right wheel of rear suspension, 82 seconds of the proving ground lap). The green (---) curves show the accelerations after 80 Hz low pass filtering the displacements, the blue (---) curves show the accelerations after 20 Hz filtering.

Similar results have been obtained for the left wheel.

Since no measurements for the rotational accelerations have been available, the filter found to give good results for the linear accelerations has been applied in this case. The following table shows the results for the right wheel of the rear suspension (section of 82 seconds of the test track):

Rotation	Minimum	Maximum	RMS
x-axis	-371	401	24.3
y-axis	-10	26	0.6
z-axis	-385	658	15.8

Table 2: Calculated maximum rotational accelerations in rad/s^2 for the rear suspension

Again some slight modifications concerning damping of some bushings have been performed to achieve these results.

The low pass filtering of the simulation results has been adapted to the measured data. Using 80 Hz for horizontal and rotational accelerations and 20 Hz for vertical acceleration has been found to give a reasonable overall agreement with the measurements. The simulation results shown above have been accepted as a basis for the following analysis.

4 The test rig model

4.1   The multibody model M2

As shown in figure 1, the hexapod consists of a base and a top platform, which are connected via 6 identical actuators. The joints between the actuators and the platforms have 2 rotational degrees of freedom (DOFs). One actuator contains the piston and the cylinder, which in turn are connected using a cylindrical joint. This construction contains 6 DOFs, namely the displacements Δl_i of the pistons or equivalently the distances l_i of the joints. This uniquely defines the position x_R and the orientation α_R of the wheel centre point (called tool centre point TCP). It is also possible and in fact even simpler to express the actuator quantities Δl_i or l_i by the TCP variables x_R, α_R, that is to right down the relations $l_i = l_i (x_R, \alpha_R)$. The inverse relation cannot be expressed in explicit form. If needed the corresponding equations have to be solved implicitly.

The concept of the test rig implies, that essentially all actuators have to be active for nearly all required motions. This leads to a distribution of the load to all actuators, especially in the case of the most often needed vertical motion.

To write down the equations of motion for the hexapod system, the following notions are introduced:

Bezeichnung	Erläuterung
x_R, α_R	Position and orientation of TCP in global coordinates
x_S	Centre of mass of top platform in global coordinates
x_{Zi}, $x_{Ki}, i = 1,\dots,6$	Centre of mass of cylinder and piston in global coordinates
x_{Pi}, $x_{Bi}, i = 1,\dots,6$	Position of joints at the base and the top platform in global coordinates
m_p, m_{Zi}, $m_{Ki}, i = 1,\dots,6$	Mass of platform, cylinder and piston
I_p, I_{Zi}, $I_{Ki}, i = 1,\dots,6$	Inertia tensor of platform, cylinder and piston
ω_p, $\omega_i, i = 1,\dots,6$	Angular velocity of platform and cylinder/piston
G_p, G_{Zi}, $G_{Ki}, i = 1,\dots,6$	Gravitational force on platform, cylinder and piston
F_p, M_p	Wheel forces and moments in the local wheel coordinate system
S_p	Transformation matrix from local wheel coordinate system to the global reference frame
F_{Pi}, $F_{Bi}, i = 1,\dots,6$	Section forces at the upper (P) and lower (B) joints
$n_i, i = 1,\dots,6$	Normalized direction vectors of the actuators (from lower to upper joint)
$f_i, i = 1,\dots,6$	Hydraulic or actuator forces generated by the actuators

Table 3: Notions for the hexapod model

All kinematical variables in the table can be uniquely expressed as functions of the TCP variables x_R, α_R .

The force balance for the top platform is

$$\text{(P1)} \qquad m_p \ddot{x}_s = S_p F_p + \sum_{i=1}^{6} (F_{P_i} + f_i n_i) + G_p$$

where

$$\text{(P2)} \qquad \ddot{x}_s = \ddot{x}_R + \omega_p \times (\omega_p \times (x_s - x_R)) + \dot{\omega}_p \times (x_s - x_R)$$

The torque balance with respect to TCP is

$$\text{(P3)} \qquad \frac{d}{dt}(I_p \omega_p) + m_p (x_s - x_R) \times \ddot{x}_R = S_p M_p + \sum_{i=1}^{6} (x_{P_i} - x_R) \times (F_{P_i} + f_i n_i) + (x_s - x_R) \times G_p$$

Here, it has been used that no torques can be applied at the upper joints. The left side of (P3) contains the inertia terms including the difference between the centre of mass and the TCP where the torques are related to. The unknown section forces F_{Pi} , $i = 1,…,6$ need to be calculated from the equilibrium equations of the actuators:

$$\text{(A1)} \qquad m_{K_i} \ddot{x}_{K_i} + m_{Z_i} \ddot{x}_{Z_i} = -F_{P_i} + G_{K_i} + G_{Z_i} + F_{B_i} \ , i = 1,..,6$$

$$\text{(A2)} \qquad \frac{d}{dt}(I_{K_i} \omega_i + I_{Z_i} \omega_i) = (x_{K_i} - x_{B_i}) \times G_{K_i} + (x_{Z_i} - x_{B_i}) \times G_{Z_i} - (x_{P_i} - x_{B_i}) \times F_{P_i} \ , i = 1,..,6$$

The torque balance equation (A2) already resembles the fact, that the angular velocities for cylinder and piston with respect to the lower joint are the same. A closer look to the equations (A1), (A2) shows, that there are only 5 independent equations. Using a decomposition of the forces at the lower joints into a radial part along the actuator and a perpendicular part

$$F_{B_i} = F_{n,B_i} + F_{s,B_i} \ ,$$

and considering the cylindrical joint between cylinder and piston, the radial part can be written in the form

$$F_{n,B_i} = m_{Z_i} \omega_i \times (\omega_i \times (x_{Z_i} - x_{B_i})) - (G_{Z_i} \cdot n_i) n_i$$

which contains the rotational movement (cross product term) and the gravitational force of the cylinder (no gravitational piston term). Inserting this into (A1) and (A2), solving for the section forces F_{Pi} and combining the result with (P1) and (P3) leads to a system of 6 equations relating the 6 DOFs of the top platform with the wheel forces and moments and the actuator forces.

These equations are linear in the actuator forces f_i. Thus they can be calculated easily if the motion of the top platform is given. The equations have been implemented in MATLAB to calculate the actuator forces needed for certain suspension tests. This is described in more detail in section 5.

Additionally the hexapod has been modelled in ADAMS/Car as a template. A linear bushing element has been used for the connection to the wheel hub of the suspension models. The geometric parameters such as radius of base and top platform, neutral length of the actuators, location of TCP etc. can easily be modified to analyse their influence on the performance of the test rig.

4.2 The model M3 for hydraulics and control

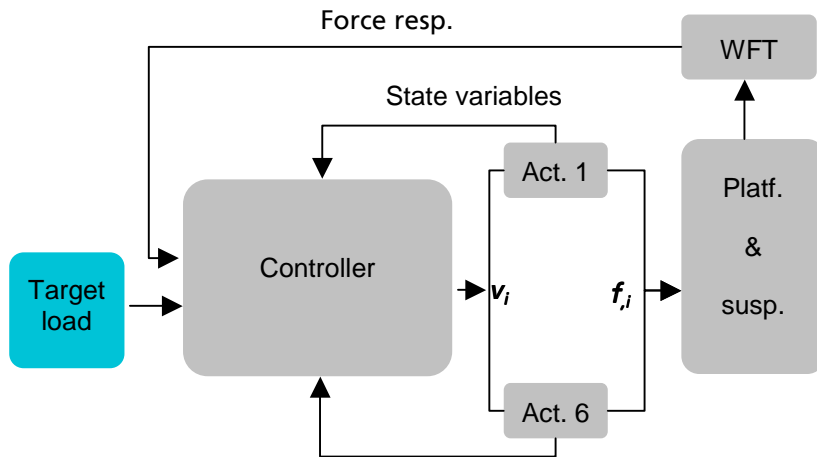The following figure shows the rough structure of hydraulics and control.

Fig. 7: Sketch of the control mechanism

The target loads i.e. the wheel forces measured on the test track are fed into the controller together with the observed wheel forces (response) during the simulation. The deviation is calculated for each DOF and a control signal (valve setpoint) for each actuator is calculated. The controller used in the simulation is essentially the same as the one used in the hardware. The valve setpoints v_i are input into the hydraulic model as sketched in figure 8.
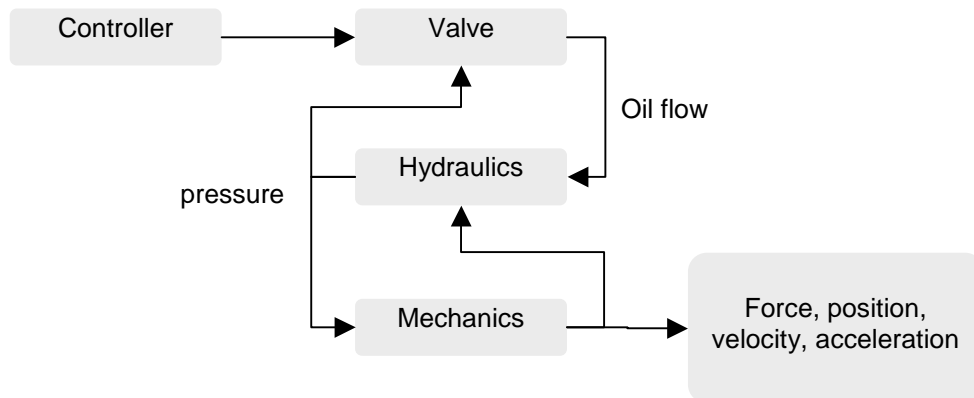

Fig. 8: Sketch of the actuator modelling

The valve model is used to calculate the oil flow and from this, the pressure and the actuator forces fi are determined. Leakage and friction are taken into account. All parameters such as compressibility, viscosity etc. can be adapted easily.

4.3   The complete model

The multibody model M2 and the model for the hydraulics and control are combined in an ADAMS/Car template. The controller is implemented in an external library and integrated via user subroutines. The communicator mechanism is used to create assemblies with arbitrary ADAMS/Car suspension models as sketched in figure 2. The following figure 9 shows an assembly for a rear suspension.
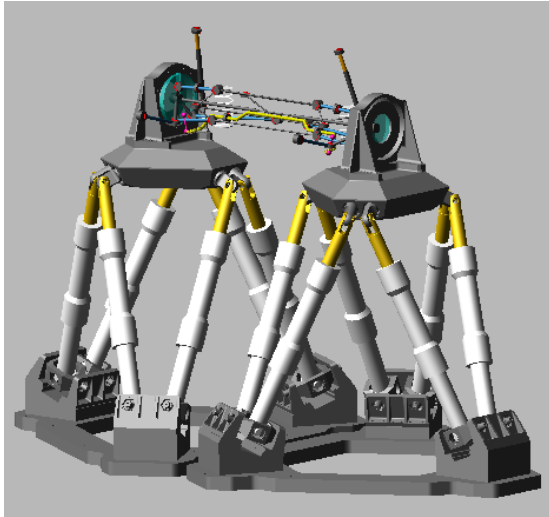
Fig. 9: The model of the entire testing system. The construction for fixing the suspension is not shown here.

With this simulation environment all steps, which have to be performed during a physical suspension test can be simulated.

5 Analysis: Configuration and design of the hexapod

5.1 Calculation of the actuator forces

As can be seen from table 1, the highest value of the measured accelerations is about 400 m/s^2. For each 10 kg mass of the top platform 4 kN are required to generate the acceleration. This value can even increase, if motion in several directions is considered. In addition, the static load and the mass of the suspension have to be taken into account. In general, one single actuator need not generate the required force, but this simple investigation shows, that the load on the actuators will be very high.

The suspension models and the rigid body model of the hexapod have been used to calculate the actuator forces needed to execute a test. To this end, the slightly modified suspension models as described in section 3 have been excited with the target loads and the wheel displacements (reference displacements) have been filtered and logged. For the front suspension a segment of about 70 seconds from the test track data including bad road, steering and braking events has been used. For the rear suspension a segment of about 80 seconds from the test data including high accelerations has been chosen.

Using the MATLAB-model (section 4.1) the actuator forces have been calculated from the target loads and the reference displacements. Figure 10 shows the process for the calculation and validation of the actuator forces.
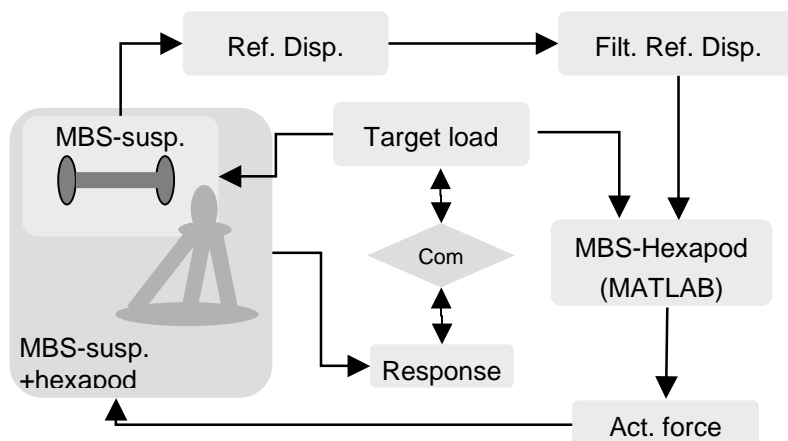

Fig. 10: Sketch of the process for calculating and verifying the actuator forces

Figure 11 shows the level crossing and range pair histograms of the calculated forces based on the rear suspension data.
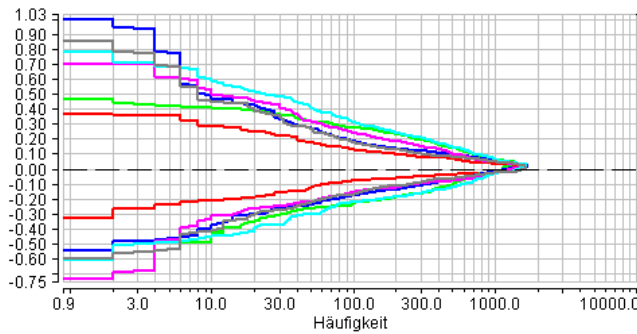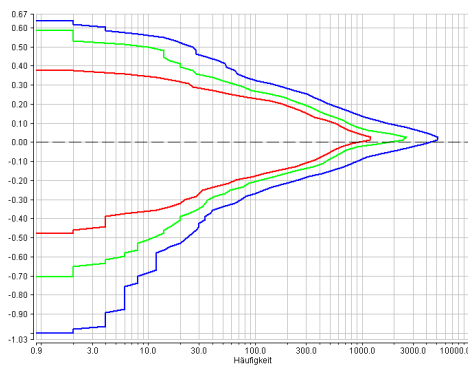

Fig. 11: Level crossing diagram of the 6 actuator forces for the rear suspension (normalized)

The forces of the actuators differ from each other. Typically, the peaks are very sharp and implied by high accelerations. Smoothing the reference displacements decreases the accelerations and thus the actuator forces without considerably changing the amplitudes of the displacements. Using the front suspension data the validity of this procedure will next be discussed in a bit more detail.

In addition to the 80 Hz cut off frequency, the displacements have been filtered with 50 Hz and 30 Hz and the corresponding actuator forces have been computed. The following figures 12 and 13 show the influence of the filter process on the forces.

--- 80 Hz, --- 50 Hz, --- 30 Hz

Fig. 12: Level crossing diagram of the force (normalized) at one actuator for different filter levels
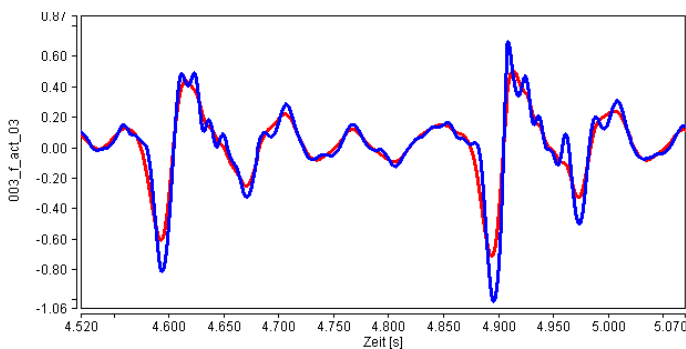
--- 80 Hz, --- 30 Hz

Fig. 13: Short segment of the force (normalized) at one actuator for different filter levels (amplitudes decrease with cut off frequency)

To validate the computational process and the results, the forces have been used as input into a model consisting of the suspension and the multibody part of the hexapod (see figure 10). The response of this model has been logged and compared to the target loads and reference displacements. A good agreement has been found. Of course, the deviations increase with decreasing cut off frequency but the differences are much smaller as expected. The following figure shows a segment of the data for the vertical displacement, the longitudinal wheel force and the vertical torque.
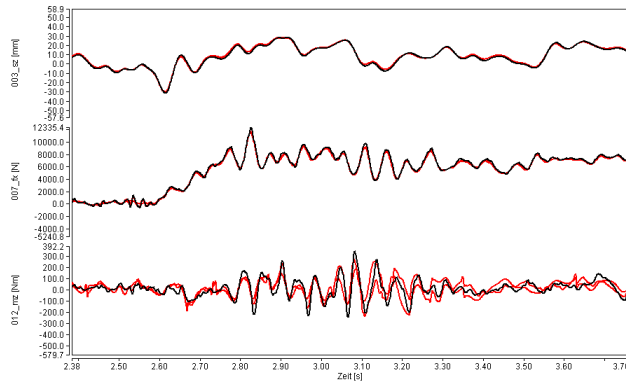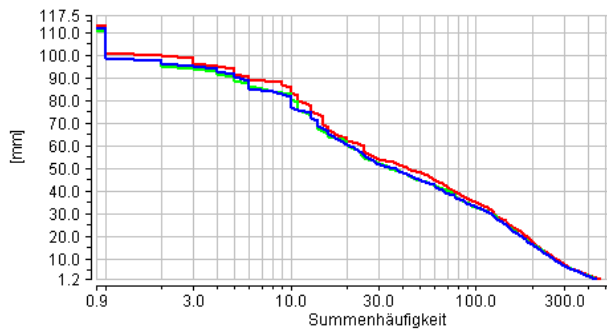
Fig. 14: Comparison of target loads und reference displacements (---) with the results obtained after applying the calculated actuator forces (2 filter levels) (---)

The following figures show some typical range pair results for the data coming from the 80 Hz and 30 Hz filter. The 50 Hz data lies in between and is not shown here.

--- Reference disp. vertical [mm]
--- after 80 Hz filtering [mm]
--- after 30 Hz filtering [mm]

Fig. 15: Vertical wheel displacement
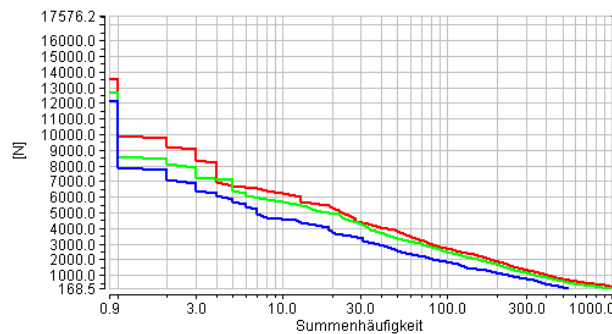
--- Target load longitudinal [N]
--- after 80 Hz filtering [N]
--- after 30 Hz filtering [N]

Fig. 16: Longitudinal force

The results not shown here are similar. The filter process strongly affects the dynamics and thus the actuator forces. However, the different actuator forces lead to almost similar response of the suspension. This fact can be used during the formulation of the requirements and the design of the hexapod. The drive file iteration process on a physical rig will implicitly apply a kind of smoothing, because only the deviation between target loads and response data is minimised in a certain sense, whereas displacements or accelerations are not controlled explicitly.

5.2   Influence of different configurations on the actuator forces

The geometry of the hexapod governs the relation between the actuator forces and the wheel forces and accelerations. Modifying the configuration can thus be used to adapt the system to the testing requirements.

To analyse this relation, the forces have been calculated from the target loads and reference displacements using different configurations. Data from the right wheel of the front suspension was

used. The analysis was performed separately for different segments of the test track. Bad road, steering and braking events were chosen.

The radius of the base and top platform, the neutral actuator length and the distance of the joints at the top platform were modified. Smaller angles between the actuator neutral position and the base platform will increase the performance of the hexapod in the horizontal plane.
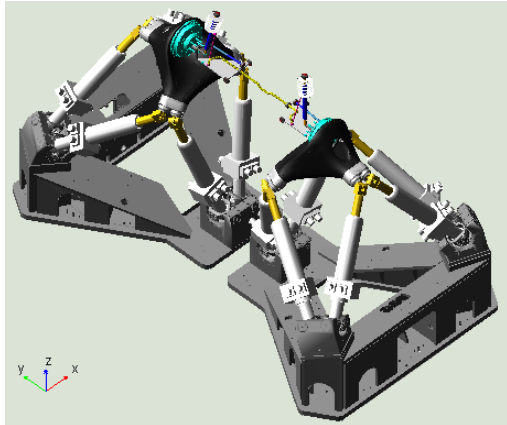

Fig. 17: New design of the hexapod

Thus, it is expected that the actuator forces will decrease with such a configuration. In fact, this effect can be observed for all segments under consideration. However, the reduction of the actuator forces was rather small. Therefore, other configurations motivated by the simulations have been analysed since then and a rather different configuration has been found (figure 17) to reduce the required actuator forces. In addition to geometrical modifications, the actuators have been strengthened and the mass of the platform has been reduced.

6 Conclusion and future work

The goal of the project, namely the development of a complete simulation environment for suspension testing on the hexapod concept, has been reached. All models are integrated into an ADAMS/Car environment and can thus be used in the development and testing process at Volkswagen.

The control mechanism has been redesigned at FCS in the meantime and the new concept will be integrated into the simulation environment. The hydraulic model used in this project has not been finally validated. Both topics are subject to ongoing work.

The multibody model of the hexapod has successfully been applied to the improvement of the hexapod design. The separation of the complete assembly into the suspension and the hexapod model and the consideration of the reference displacements has proven to be an effective way to optimise the configuration.

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