HIGH–DYNAMIC TEST BED FOR MECHATRONIC VEHICLE SUSPENSIONS

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Abstract

Advanced vehicles are mechatronic systems consisting of mechanical parts like rigid bodies, flexible tires, bearings, joints, springs and dampers as well as electronic components including actuators, control devices and sensors. In particular the electronic parts have to be tested rigorously by realistic loads. For this purpose a high–dynamic hydraulic test bed for actively controlled vehicle suspensions has been developed. The time–varying loads are obtained from a real–time simulation of multibody systems equations representing the mechanical parts of the vehicle. The test bed is operating up to frequencies of 35 Hz without dynamic errors in amplitude and phase.

1. Introduction

In recent years new concepts for vehicle design have been characterized by a strong impact of electronically controlled components like engine control, anti–scid breaking and actively controlled suspensions. However, the interaction between the mechanical parts of a vehicle and the electronic parts results in a strongly coupled mechatronic system. The mechanical parts are most reliable due to the long history of vehicle engineering while this is not the case for the electronic components. Therefore, there is a special need for testing the electronic components under realistic loads. This can be achieved by road tests but such a procedure is very expensive. Another possibility is the modeling of the mechanical parts of the vehicle as a multibody system. Then, the dynamics of the vehicle is completely known depending on the mechanical parameters of the system and the geometry of the test roads given. An analysis of the overall system can be performed by interfacing the electronic components to a real–time simulation of the mechanical parts. This concept of hardware–in–the–loop simulations of vehicle systems was also discussed in Ref. [1].

A vehicle as a mechatronic system is shown in Figure 1. The electronic parts of the vehicle include the actuators, the sensors, the control devices and the control software. These components show complex eigendynamics and, in contrast to the mechanical parts, the modeling requires extensive approximations impairing the accuracy of the mathematical model. From this point of view the hardware–in–the–loop simulations are a suitable approach for the dynamic analysis of vehicles. The hardware–in–the–loop model of an actively controlled vehicle suspension is presented in Figure 2. The hydraulic test bed serves now as an interface between the software model of the mechanical components and the hardware model of the electronic components.

Several aspects of hardware–in–the–loop simulations have been treated in the literature, e.g. Voy, Lenke and Garavy [2] applied an analog computer for the simulation of

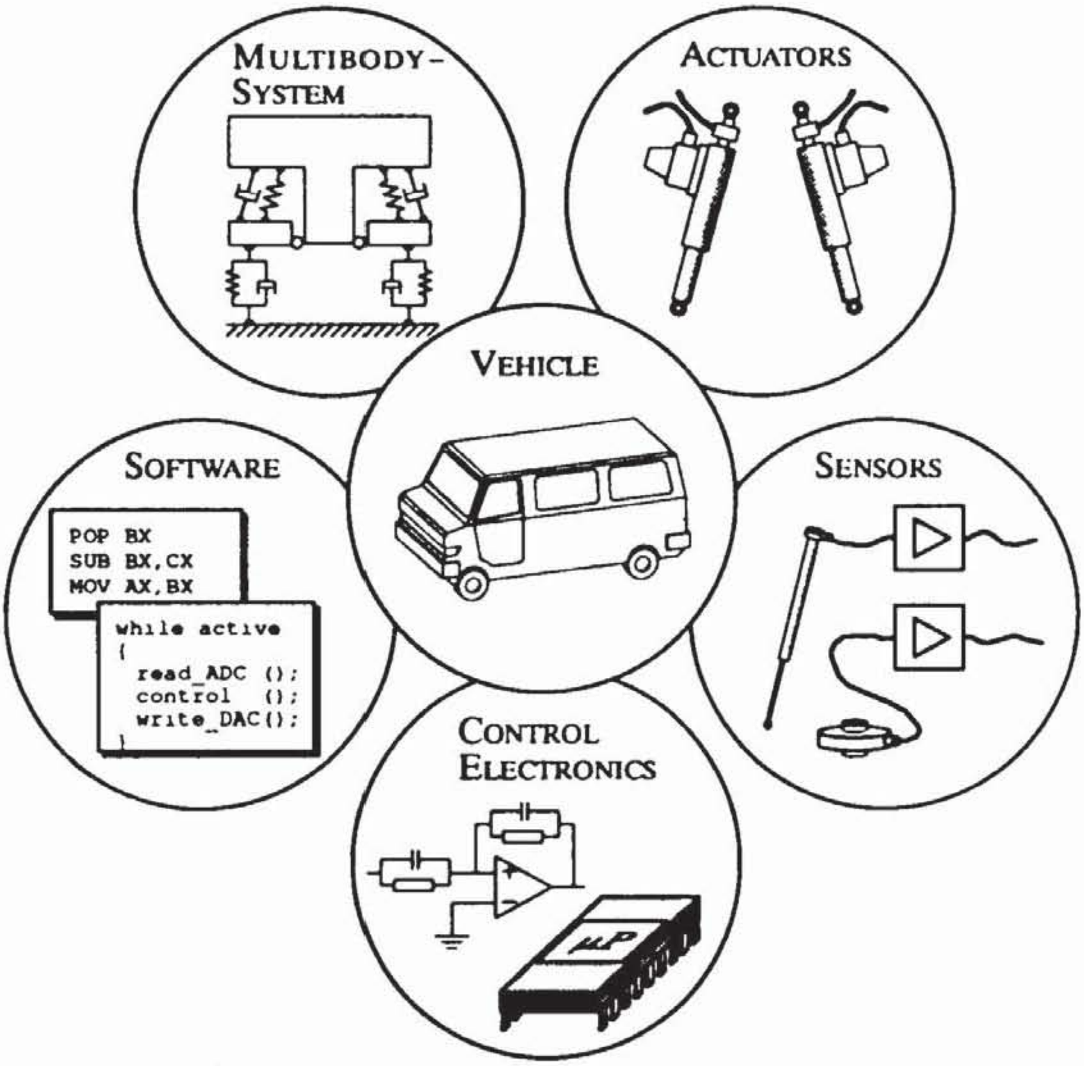

Figure 1: A vehicle as a mechatronic system

the mechanical parts, JÄKER [3] investigated the controller design for active suspensions with a hardware–in–the–loop set–up, HANSELMANN [4] shows the use of signal processors for such applications, and OCHNER and HENNECKE [5] used real–time simulations for the development of suspension systems in practical application. Today, real–time simulations of multibody systems are feasible due to fast algorithms, parallel computing and transputer technology, EICHBERGER, FÜHRER and SCHWERTASSEK [6], BAE, HWANG and HAUG [7], INMOS LTD. [8].

In this paper, symbolical equations of motion of multibody systems will be applied, the design of a high–dynamic hydraulic test bed is discussed, and some results on real–time simulations are presented.

2. Hardware–in–the–loop Modeling

For the development of a hardware–in–the–loop model of a mechatronic system the mechanical parts can be modeled with the multibody system method. As shown in Figure 3, a multibody system consists of rigid bodies, connected by constraint elements like joints and bearings and by coupling elements like springs, dampers or force–con-

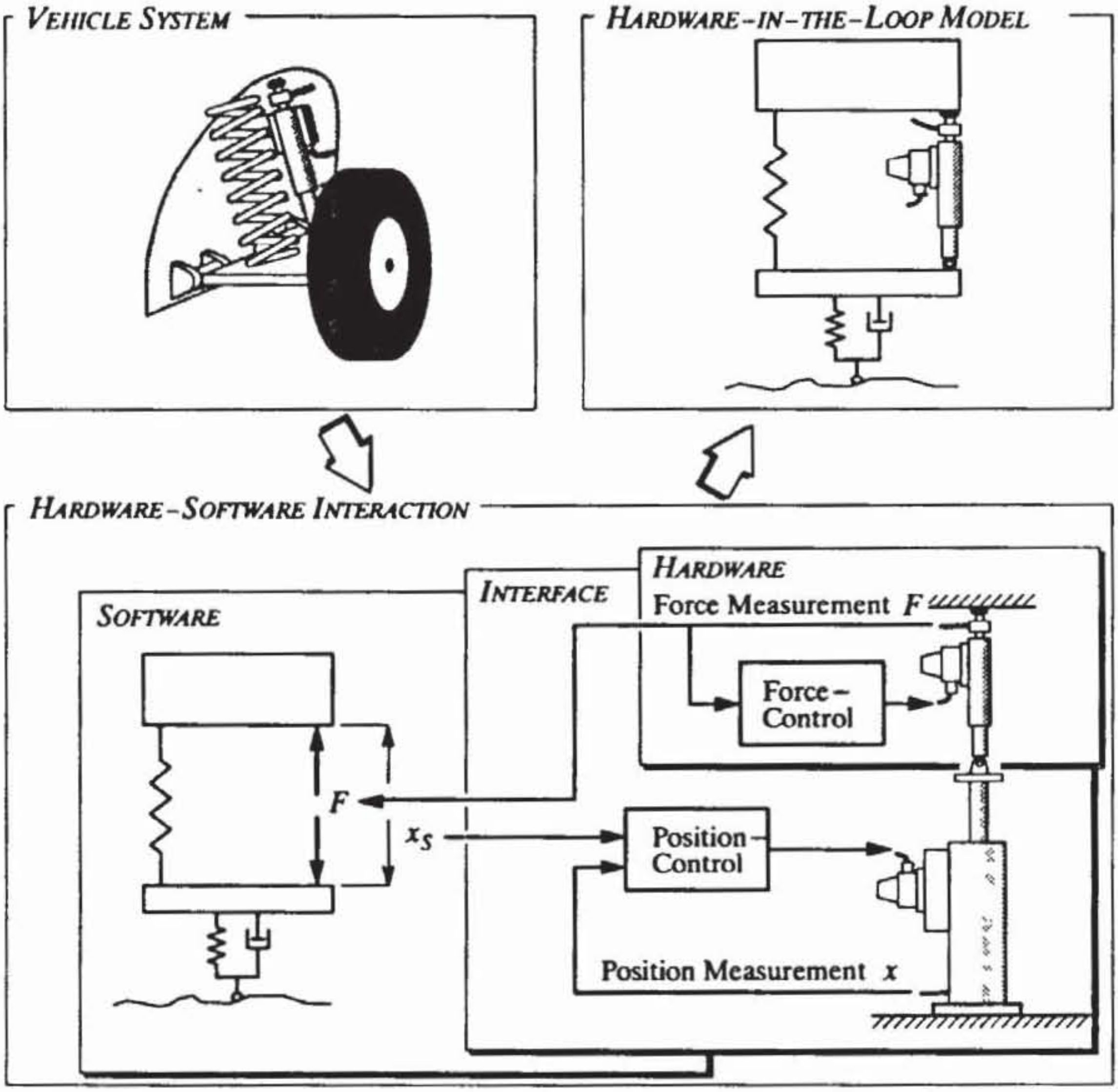

Figure 2: Hardware–in–the–loop model of an active vehicle suspension

trolled actuators. The theory of multibody system modeling is well established in the literature, e.g. SCHIEHLEN [10]. A compact and efficient formulation of the equations of motion can be obtained by using a minimal set of f generalized coordinates for the discription of the kinematics of the system, where f denotes the number of degrees of freedom. With the $f \times 1$ vector of generalized coordinates $\boldsymbol{y}$ and its first time derivative $\boldsymbol{z}$ the modeling process yields equations of motion of holonomic systems with the form

$$\dot{\boldsymbol{y}} = \boldsymbol{z},$$
$$\boldsymbol{M}(\boldsymbol{y},t)\,\dot{\boldsymbol{z}} + \boldsymbol{k}(\boldsymbol{y},\boldsymbol{z},t) = \boldsymbol{q}(\boldsymbol{y},\boldsymbol{z},t). \tag{1}$$

Here, $\boldsymbol{M}$ denotes the symmetric and positive definite $f \times f$ inertia matrix, the $f \times 1$ vector $\boldsymbol{k}$ contains the generalized Coriolis and centrifugal forces and the $f \times 1$ vector $\boldsymbol{q}$ contains the generalized applied forces. The modeling of multibody systems is supported by computer formalisms such as the program NEWEUL, KREUZER and LEISTER [11]. For

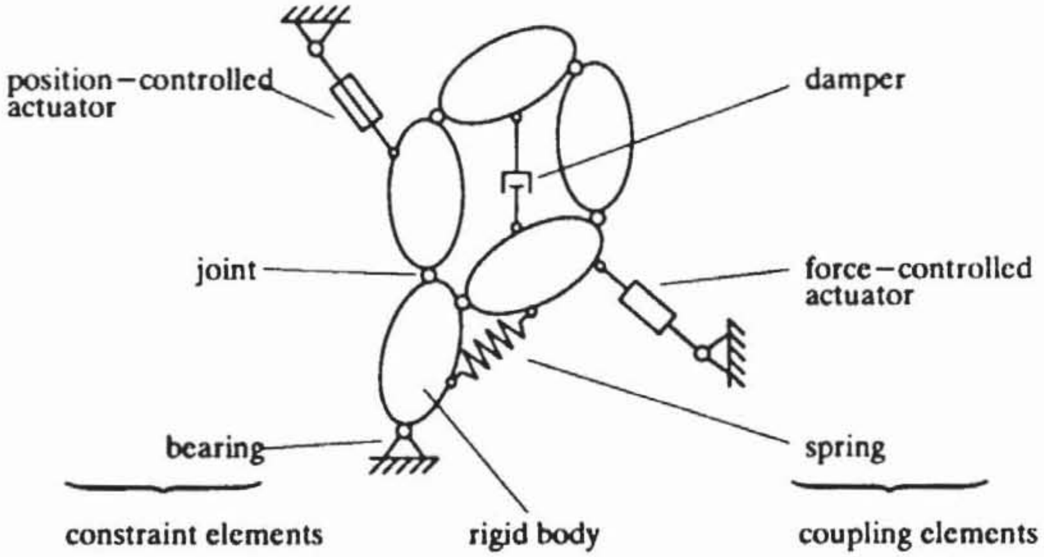

Figure 3: Structure of a multibody system

real-time implementation of the symbolic equations of motion on transputer networks special interfaces have been developed, SCHÄFER [9].

Eq. (1) serves as a model for the dynamic behavior of the mechanical system components. However, the behavior of the overall system is influenced by the dynamics of controlled elements exerting forces and torques on the mechanical parts. This is illustrated in Figure 4 for a multibody system with a force-controlled actuator. The actuator is mounted into the system at two nodal points B_i and B_j characterized by the reference frames K_{B_i} and K_{B_j} fixed to the corresponding bodies. As long as mass and inertia of the actuator can be neglected, the motion input to the actuator can be described by the relative displacement vector $\boldsymbol{c}_{ij}$ and the relative rotation tensor $\boldsymbol{S}_{ij}$ between the nodal reference frames and the corresponding velocities. These quantities can be computed from the motion of the adjacent bodies which, on the other hand, is subject to the forces $\boldsymbol{f}_{ij}$, $\boldsymbol{f}_{ji}$ and the torques $\boldsymbol{l}_{ij}$, $\boldsymbol{l}_{ji}$ produced by the actuator. These forces and torques result from the dynamic behavior of the actuator under the influence of the motion input and a given control input. Thus, the dynamic equations of the multibody system and the actuator are coupled and cannot be treated separately. If the states of the actuator are contained in an $r \times 1$ vector $\boldsymbol{w}$ the simulation model for the overall system takes the form

$$\begin{aligned} \dot{\boldsymbol{y}} &= \boldsymbol{z}, & \boldsymbol{y}(t = t_0) &= \boldsymbol{y}_0, \\ \boldsymbol{M}(\boldsymbol{y},t)\,\dot{\boldsymbol{z}} &= \boldsymbol{q}(\boldsymbol{y},\boldsymbol{z},\boldsymbol{w},t) - \boldsymbol{k}(\boldsymbol{y},\boldsymbol{z},t), & \boldsymbol{z}(t = t_0) &= \boldsymbol{z}_0, \\ \dot{\boldsymbol{w}} &= \dot{\boldsymbol{w}}(\boldsymbol{y},\boldsymbol{z},\boldsymbol{w},t), & \boldsymbol{w}(t = t_0) &= \boldsymbol{w}_0, \end{aligned} \tag{2}$$

where the dynamics of the controlled actuator is represented by an additional set of differential equations and $\boldsymbol{y}_0$, $\boldsymbol{z}_0$ und $\boldsymbol{w}_0$ denote the initial states at the initial time t_0. However, very often an appropriate model of this form is not readily available due to the complex dynamics of actuators and electronic control devices.

An alternative approach to the analysis of mechatronic systems consists of replacing the simulation model (2) of the overall system by a hardware-in-the-loop model. Instead of modeling the dynamics of actuators and control devices, these components are oper-

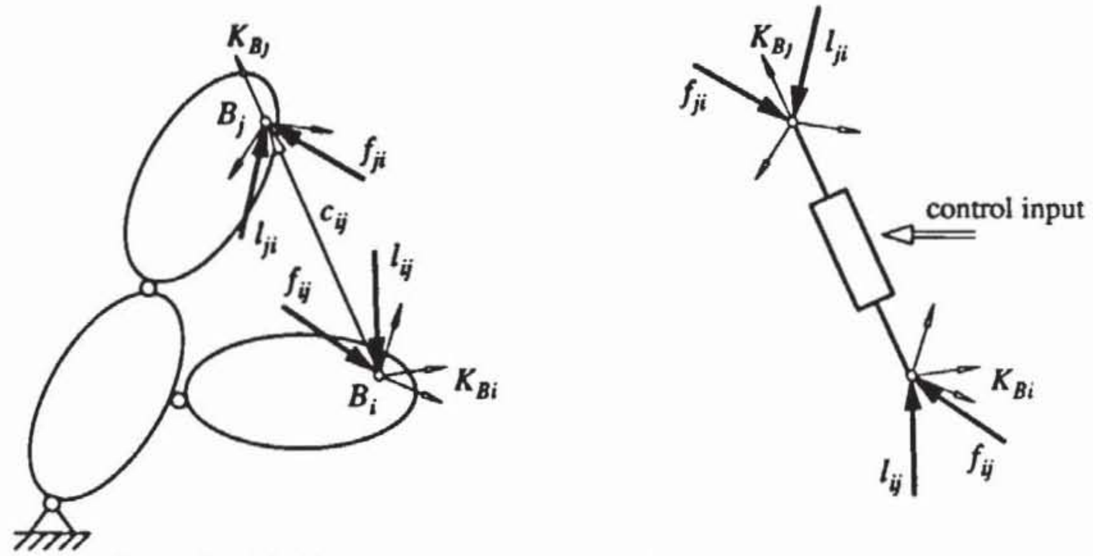

Figure 4: Multibody system and separated force–controlled actuator

ated in an experimental set–up and the coupling to the multibody system is achieved by hardware–software interaction. The structure of such a hardware–in–the–loop model was shown already in Figure 2 for an example of active suspension control. It can be seen that the loop formed by the multibody system model and the test hardware contains an interface assembly that is needed to provide the actual motion input to the hardware parts of the model. This funcional structure imposes strict requirements on the dynamic behavior of the test bed itself, as it has a major influence on the quality of the simulation results.

Hardware–in–the–loop modeling is a hybrid approach that requires the development of real–time simulation models of multibody systems on one hand, and the development of appropriate test rigs and hardware–software interfaces for the operation of actuator and control devices on the other hand. Once a test bed for a certain type of control device is established, a whole variety of systems can be investigated without considering the dynamic behavior of the hardware components in detail.

3. High–dynamic Hydraulic Test Bed

The work presented in this paper is focused on the development of a test bed for actively controlled vehicle suspension elements. Active suspension components can produce high forces and have an influence on vehicle motion within a considerable frequency range. The high bandwidth required for generating motion input to suspension elements can be achieved with hydraulic devices.

The main functional unit of the hydraulic test bed for hardware–in–the–loop testing of suspension elements is shown in Figure 5. A position–controlled servo system produces the motion input for a force–controlled hydraulic suspension actuator. The position servo system consists of a high–performance symmetrical double–ended hydraulic cylinder with integrated displacement sensor and a maximum total stroke of 200 mm. It is controlled by two two–stage high–response servovalves operating in parallel. Thus, the oil flow required for the maximum piston velocity can be provided with smaller

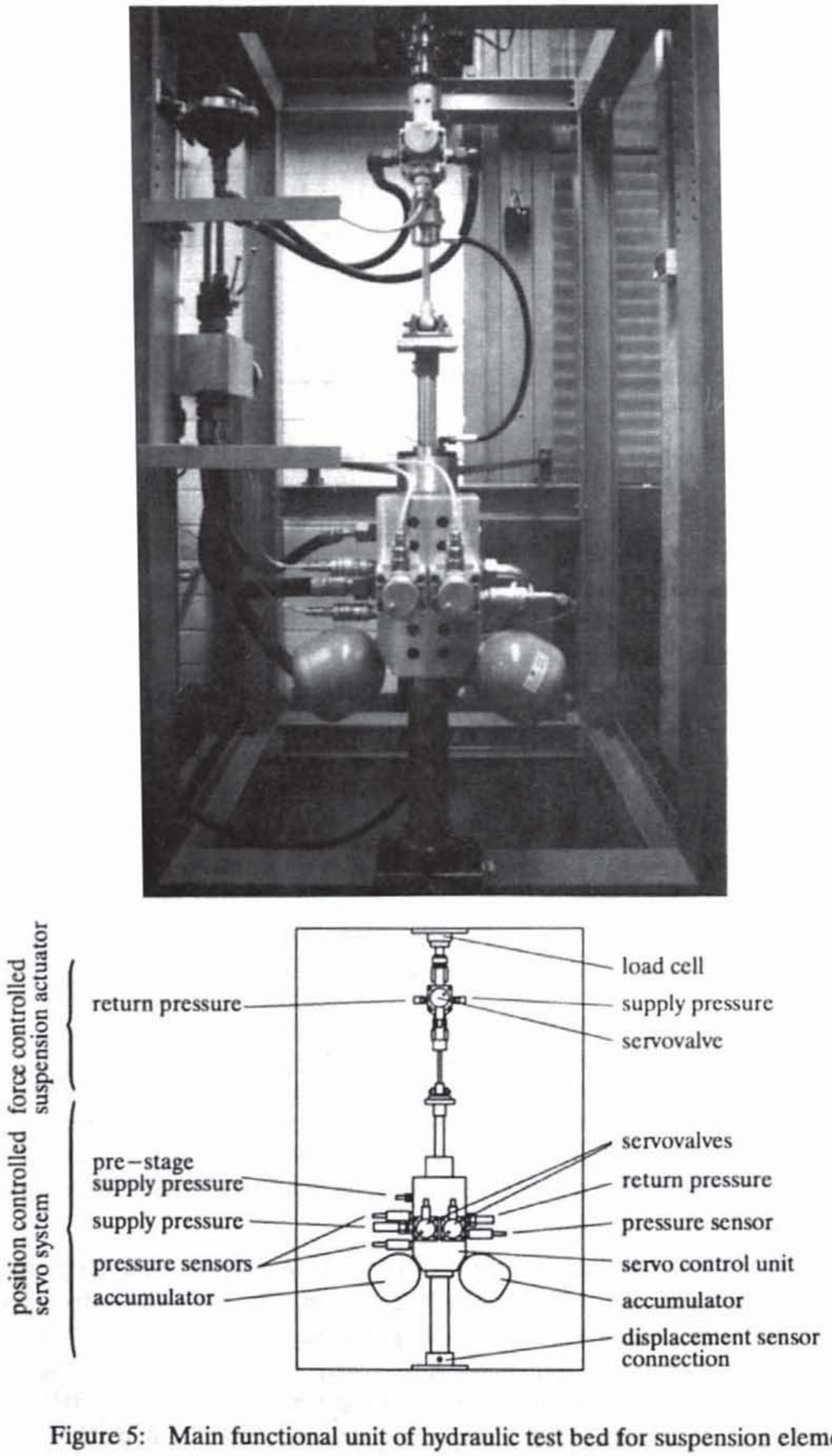

Figure 5: Main functional unit of hydraulic test bed for suspension elements

valves having considerably higher bandwidth than a single valve with double nominal flow. Another design measure for improving dynamic properties is the use of an external pressure supply for the pre–stage of the servovalves so that supply pressure fluctuations caused by fast motion patterns will not influence the servovalve behavior. Accumulators in supply and return pipes further reduce dynamic pressure changes. Pressure sensors provide measurements of supply and piston chamber pressures for control purposes. The force–controlled actuator consists of a single–ended cylinder and a two–stage servovalve. It is mounted in the test rig with compliant supports as used in realistic vehicle suspensions. A load cell provides the force measurement for the force feedback and the hardware–in–the–loop simulation.

The components described above are supported by a hydraulic pressure supply unit with a variable displacement pump for the main supply, a fixed displacement pump for the pre–stage supply and various components and valves for heating and cooling of the hydraulic fluid and for control and safety features. The power control unit is connected with peripheral interfaces of a transputer system for digital control and real–time simulation. A graphics workstation is used for data monitoring and operator interaction.

In order to achieve high bandwidth and precision advanced state space control concepts have to be applied for the position–controlled servo system. Therefore, a mathematical model of the servo system has to be derived for the purpose of controller design. The modeling of hydraulic servosystems is presented in detail in e.g. BACKÉ [12] or MERRITT [13].Figure 6 shows an idealized double–ended symmetrical piston controlled by a servovalve. The piston is characterized by the effective cross–section A_P and the mass m_P. Resisting forces due to leakage flow and friction are taken into account by a viscous damping coefficient d_P. The coordinates x and y denote the piston displacement and the servovalve spool displacement, respectively, and p_1 and p_2 denote the piston chamber pressures. The unilateral stroke of the piston is given by h_0. The volumetric flow through the servovalve control ports C_1 and C_2 is denoted by q_1 and q_2 and the leakage flow between piston chambers is q_{Li}. The load F_P summarizes external forces acting on the piston.

The dynamic model of the servosystem is comprised by the equations of motion of the piston and the servovalve spool and additional differential equations for the piston chamber pressures. A major nonlinearity results from the nonlinear characteristics of the servovalve orifice flow. Linearizing about mid–stroke position and applying additional assumptions about symmetric servovalve flow and about the leakage flow, e.g. BACKÉ [12] or MERRITT [13], finally leads to the linear state equation

$$\dot{\boldsymbol{x}} = \boldsymbol{A}\,\boldsymbol{x} + \boldsymbol{b}\,u_V + \boldsymbol{e}\,r \tag{3}$$

with the 5 × 1 state vector

$$\boldsymbol{x} = \left[x_1,\ x_2,\ x_3,\ x_4,\ x_5\right]^T = \left[x,\ \dot{x},\ p_L,\ y,\ \dot{y}\right]^T, \tag{4}$$

where $p_L = p_1 - p_2$ represents the pressure difference of the piston chambers. The scalar control input u_V is given by the servovalve input and the disturbance input $r = F_H$ contains the external load. The 5 × 5 system matrix $\boldsymbol{A}$ is given by

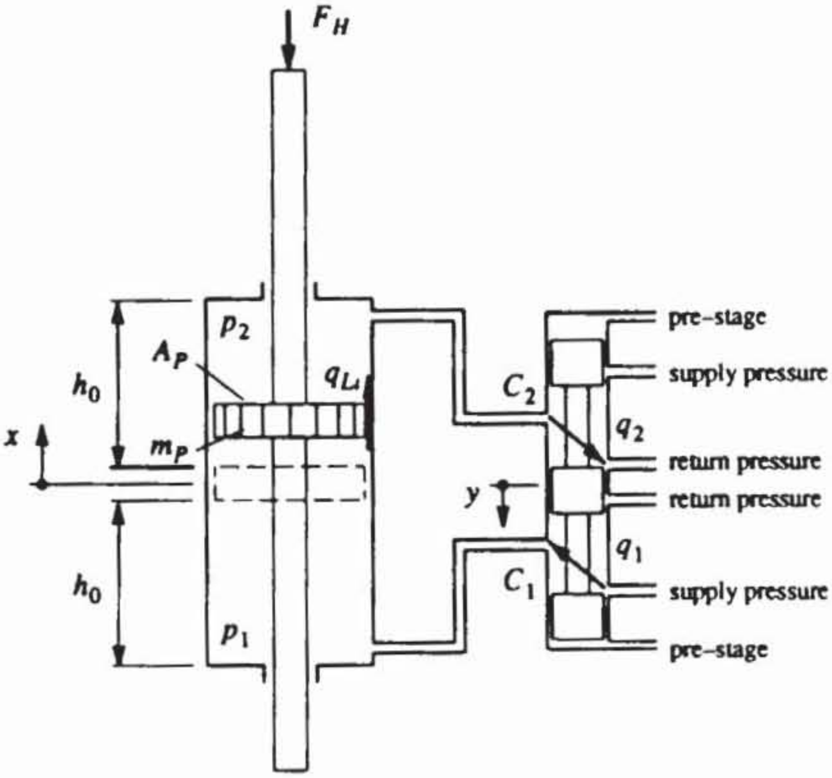

Figure 6: Double-ended piston controlled by servovalve

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ 0 & -\frac{d_P}{m_P} & \frac{A_P}{m_P} & 0 & 0 \\ 0 & -\frac{2}{c_H}(A_P + k_{Liv}) & -\frac{2}{c_H}k_{Lip} & -\frac{2K\,q_0}{c_H\,y_{max}} & 0 \\ 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & -\omega_V^2 & -2D_V\omega_V \end{bmatrix}, \tag{5}$$

and it contains several system parameters such as the maximum no-load servovalve flow q_0, the number K of servovalves, the maximum spool displacement y_{max}, the leakage flow coefficients k_{Liv} and k_{Lip} for velocity dependent and pressure dependent leakage, respectively, and the servovalve eigenfrequency ω_V and damping ratio D_V. The hydraulic capacitance c_H is determined by the efficient bulk modulus E_H of the hydraulic fluid and the efficient volume V_P of a single piston chamber according to $c_H = V_P/E_H$. Finally, the control input vector is given by

$$b = \left[\,0\,,\,0\,,\,0\,,\,0\,,\,\omega_V^2 k_V\right]^T, \tag{6}$$

where k_V denotes the servovalve gain and the disturbance input is characterized by

$$e = \left[\,0\,,\,-\frac{1}{m_P}\,,\,0\,,\,0\,,\,0\right]^T. \tag{7}$$

The state equations described above represent a 5th order model for the servosystem. Only a few of the parameters are a priori known, others have to be determined by means of extensive measurements.

The digital control of the system is implemented on a transputer system. The control law consists of a feedback part and a feedforward part. The servovalve input is computed from

$$u_V = -\boldsymbol{k}^T(\boldsymbol{x} - \boldsymbol{x}_S) + k_1\, k_S\, \dot{x}_S \tag{8}$$

where $\boldsymbol{x}_S$ contains the desired state, $\dot{x}_S$ is the desired piston velocity and k_S denotes a feedforward coefficient. The feedback gains are comprised in the 5×1 feedback vector $\boldsymbol{k} = \left[k_1,\ k_2,\ k_3,\ k_4,\ k_5\right]^T$. For the current implementation the control was developed using the linear–quadratic–regulator (LQR) approach. The piston displacement x as well as the servovalve spool displacement y are measured and the pressure state p_L can be computed from the measured chamber pressures p_1 and p_2. For the determination of the missing velocity states two digital subsystem observers are applied. The controller and observer design is outlined in more detail in SCHÄFER [9].

The feedback control part of eq. (8) handles deviations of the system state but a freqency dependent phase lag cannot be avoided. In a hardware–in–the–loop set–up this phase lag is itself fed back into the simulation model and may thus severely impair the quality of the simulation. Therefore, the feedforward part of the control was introduced for phase lag compensation, see also VOY, LENKE and GARAVY [2] and OCHNER and HENNECKE [5]. The additional control input can be considered a feedback control with a prediction of a future position as a command value which explains its phase lifting effect. With an appropriate choice of the feedforward coefficient k_S the additional control input causes a servovalve spool displacement corresponding to the flow required for a quasistatic linear motion of the piston with the desired velocity. The effect of this measure can be seen in the comparison of frequency response measurements taken at the position–controlled system with and without lag compensation, Figure 7. The frequency response was obtained using a stochastically prescribed position with 50 Hz bandwidth as system input and the measured position as output. The phase lag of the system without compensation is clearly noticeable. The lag compensated control system shows an excellent phase behavior up to approximately 35 Hz. As a consequence of the compensation, the amplitude ratio is slightly increased. However, in the frequency range up to 35 Hz, which is fully sufficient for covering body and wheel modes of suspension motion, the increase is of minor extent.

The frequency response characteristics show the high bandwidth of the position–controlled servo system. In the next step the behavior of the overall system has to be investigated in the hardware–in–the–loop set–up.

4. Real–time Simulations

For the testing of the overall system behavior a simple two–body vehicle model is sufficient and allows analytical investigations for comparison. Figure 8 shows the corresponding hardware–in–the–loop set–up. The masses of body and wheel are m_b and

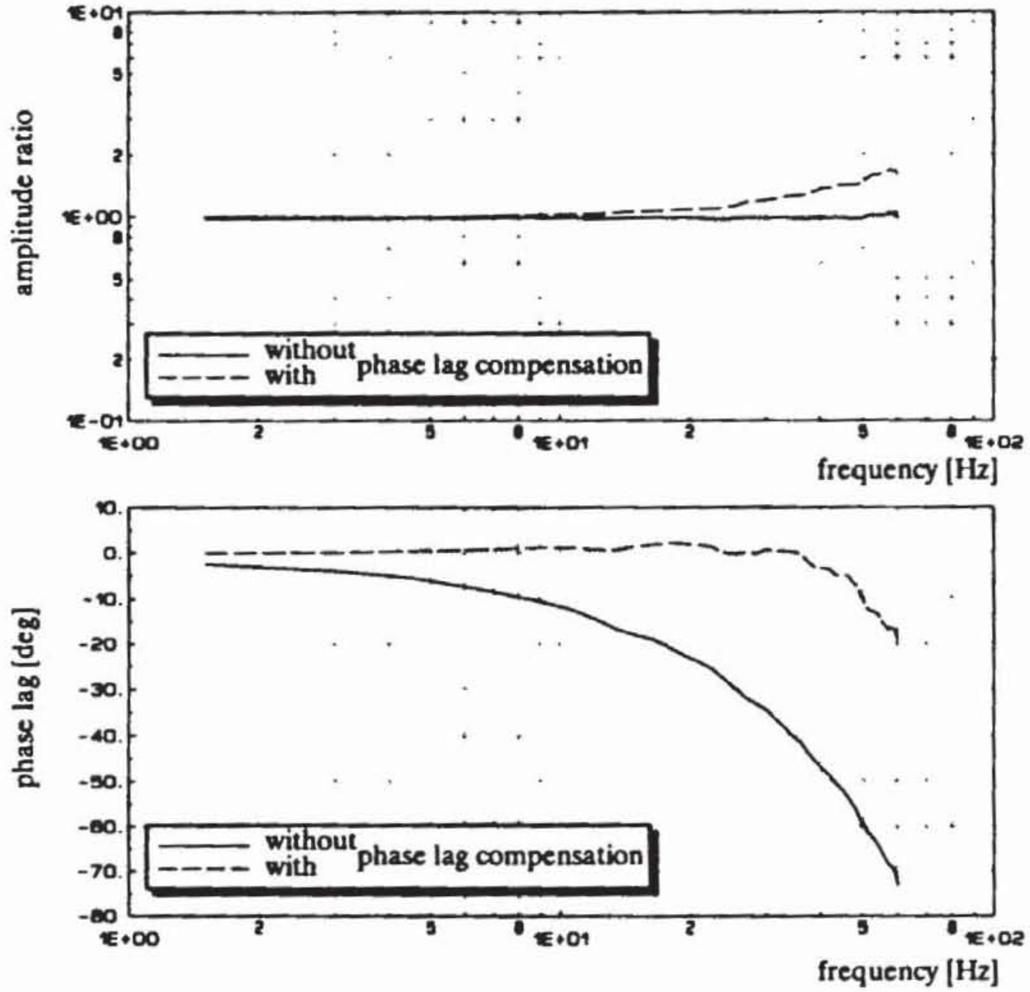

Figure 7: Frequency response of position–controlled servo system

m_w, respectively. The suspension is represented by a linear spring with stiffness c_b and a force input F_H measured at the hardware element. A linear spring with stiffness c_w is used to include tire elasticity. The road profile is described with the coordinate u_w, whereas x_b and x_w are used for the description of body and wheel motion, respectively.

Figure 9 gives an outline of the structure of the transputer network used for hardware–in–the–loop simulation and control. The real–time simulation of the simple model considered here can easily be done with a single transputer. However, for the evaluation of realistic models of complex suspensions additional transputers can be used as indicated by dashed lines. The communication between simulation and control is handled by a control interface transputer. The control itself is implemented on another transputer communicating with front–end transputers for the control of analog–to–digital (A/D) and digital–to–analog (D/A) conversion processes. Data management and communication with the host–computer for user interaction is also allocated on a separate transputer so that negative influences from handling large data records on the real–time behavior of the other processors are avoided. The network consisting of T800 transputers and the real–time software allow simulation and control of the system with separate time–steps and with sampling or integration rates up to 2 kHz.

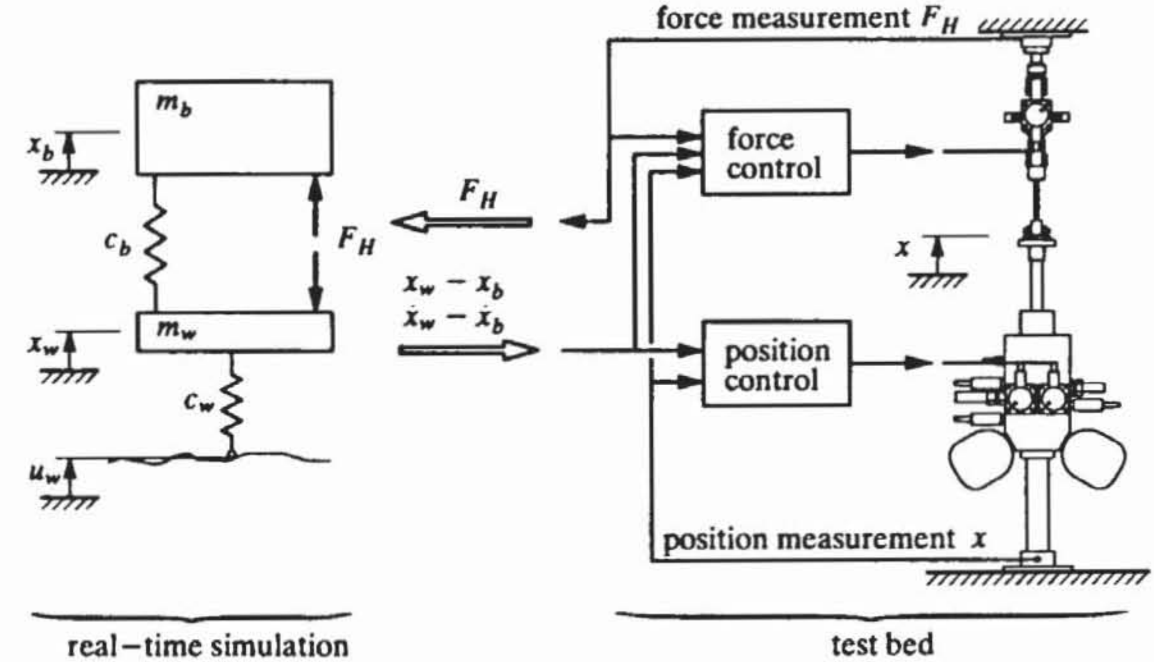

Figure 8: Hardware–in–the–loop simulation set–up

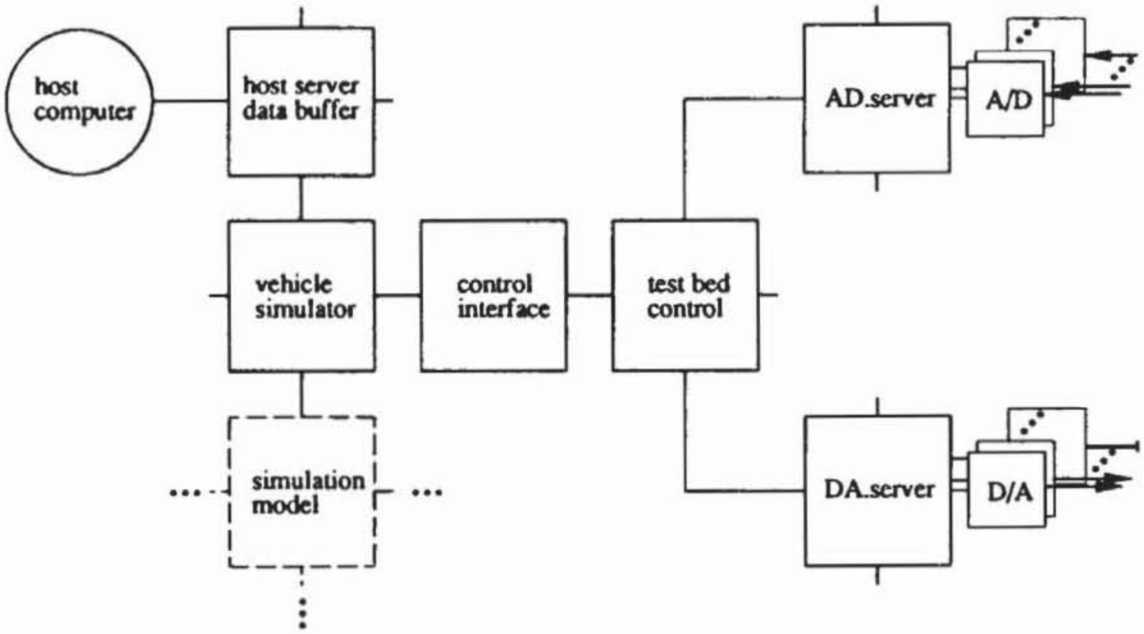

Figure 9: Transputer network for hardware–in–the–loop simulation

A crucial point in establishing a reliable test bed for the hardware–in–the–loop simulation is the validation of the system. Since a mathematical model for the overall system is not available, the results obtained by means of using a real hardware component in a simulation can normally not be checked with reference simulation results. However, at least the performance of the interface components of the test bed can be evaluated with a simple trick. Instead of using the measured force output of the hardware element in the simulation a force can be computed with the measured motion of the interface system and a given force law. The structure of this validation test procedure is illustrated in Figure 10. The results obtained in this fashion can directly be compared to a pure simulation which allows a very rigorous assessment of the quality of the test environment.

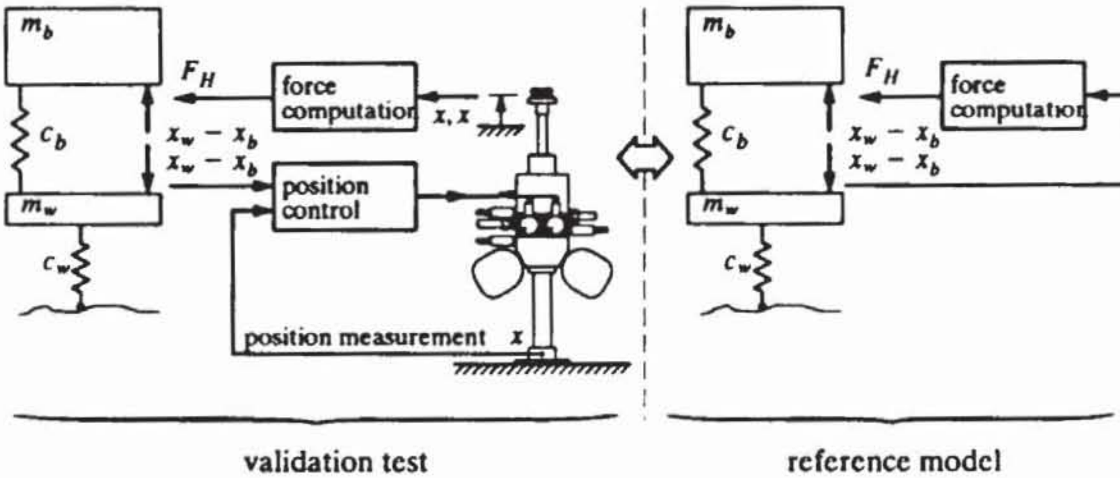

Figure 10: Validation test and reference model of hardware–in–the–loop set–up

The results of a validation test are shown in Figure 11 in the form of frequency response plots of body and wheel displacements of the linear vehicle model described before. For force computation, a simple linear damper model was applied. In the validation test, the observed piston velocity is used for the evaluation of the damper force in simulation and frequency response computation, whereas the reference model is completely linear and the frequency response can be provided analytically. In a frequency range up to 10 Hz there is good agreement of the simulated and the analytical results even without a phase lag compensation. However, above 10 Hz deviations show up especially in the response of the wheel motion. The amplitude amplification and the phase drop turn out to be too moderate. Using the phase lag compensation as described in the previous section yields major improvements. The results show a very good agreement with the analytical curves up to a frequency of 25–30 Hz. Thus, the influence of the wheel motion can be reproduced correctly and the test bed allows investigations of dynamic tire forces and ride safety under realistic conditions.

The final step toward full operation of the hardware–in–the–loop test bed consists in closing the loop and using the measured force as an input to the real–time simulation. For simplicity, again linear damping is used as a control law for the force controlled actuator. This is not a realistic choice for practical applications but it allows an assessment of the results which is intended for this report. The force controller for the hydraulic actuator was implemented with a simple proportional force feedback and a feedforward part similar to the one discussed above. The feedforward control improves the dynamic behavior with respect to the motion enforced by the position–controlled actuator. The differences between pure simulation of an ideal force–producing element and the hardware–in–the–loop simulation of the actual component, as shown in frequency response diagrams of body and wheel motion in Figure 12, stress the usefulness of the

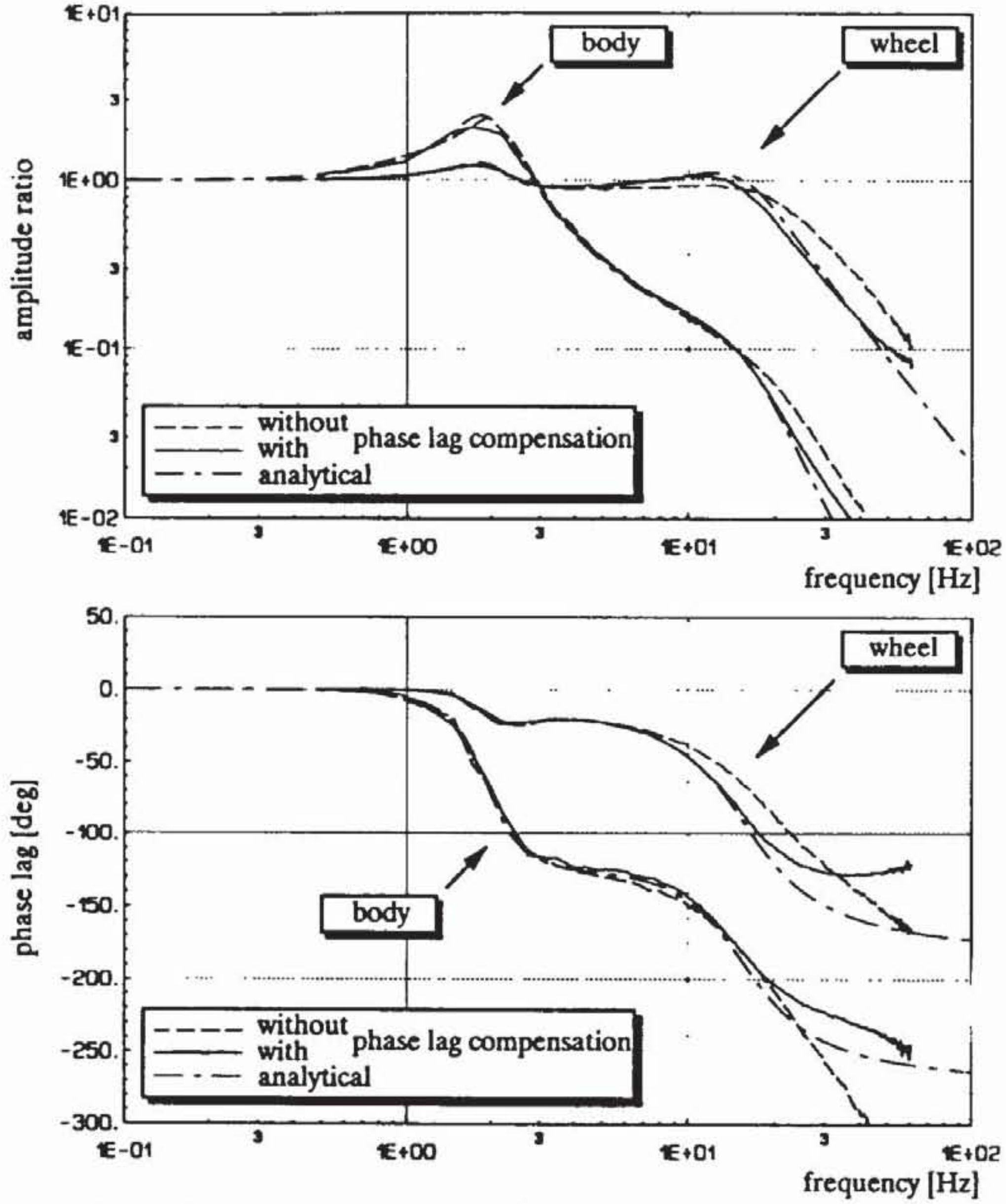

Figure 11: Frequency response of vehicle model from validation test

hardware–in–the–loop approach. In particular, the eigendynamics of the actuator lead to higher amplitude amplification and higher frequency of the wheel mode which shows that the system is stiffer and less damped than it would be expected by simply considering an ideal force element. The time history of body motion, wheel motion and damper force of the hardware–in–the–loop model and the linear reference model for alternating step input, presented in Figure 13, shows the same properties. Additionally, it shows the asymmetry of contraction and extension due to the single–ended piston.

The experiments for validation and the actual hardware–in–the–loop operation show that the hydraulic test bed described before represents a valuable tool for the investigation of mechatronic vehicle suspension systems, for the development of suspension elements and for the development and operational testing of control devices and algorithms.

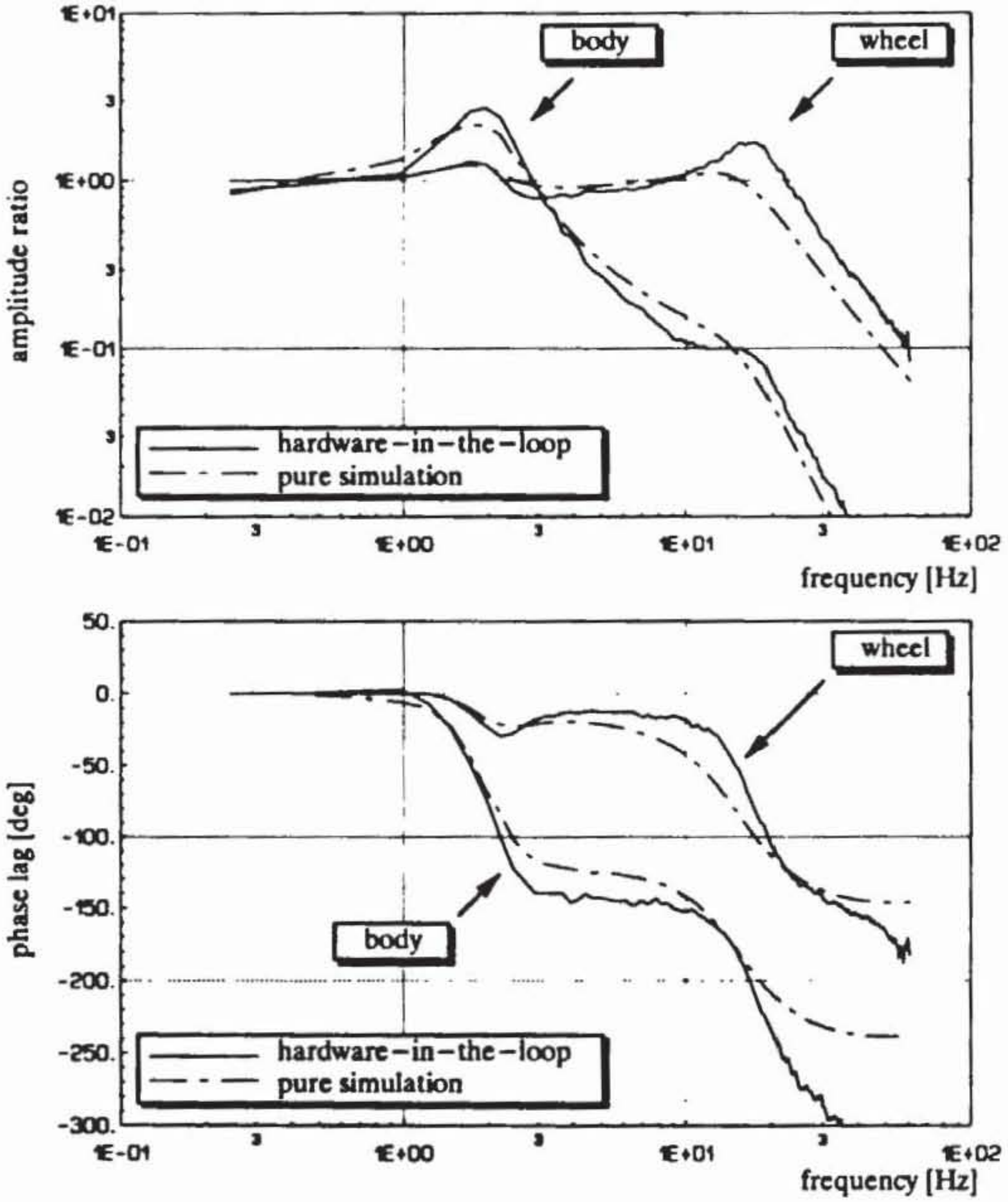

Figure 12: Frequency response of vehicle model from hardware–in–the–loop simulation

5. Conclusions

Mechatronic vehicle suspensions are complex parts of advanced vehicles requiring extensive testing. Hardware–in–the–loop simulations are proposed as an economic means for suspension tests. The mechanical parts of the vehicles are modeled by multibody system software, while the mechatronic parts are assembled as hardware in a hydraulic test bed. It is shown how real–time simulations of multibody systems may be combined with force measurements from the test bed subject to controlled motions. The design principles of the high–dynamic test bed are presented and first experiments have shown the feasiblity of the approach. More detailed studies of mechatronic vehicle suspensions are going on.

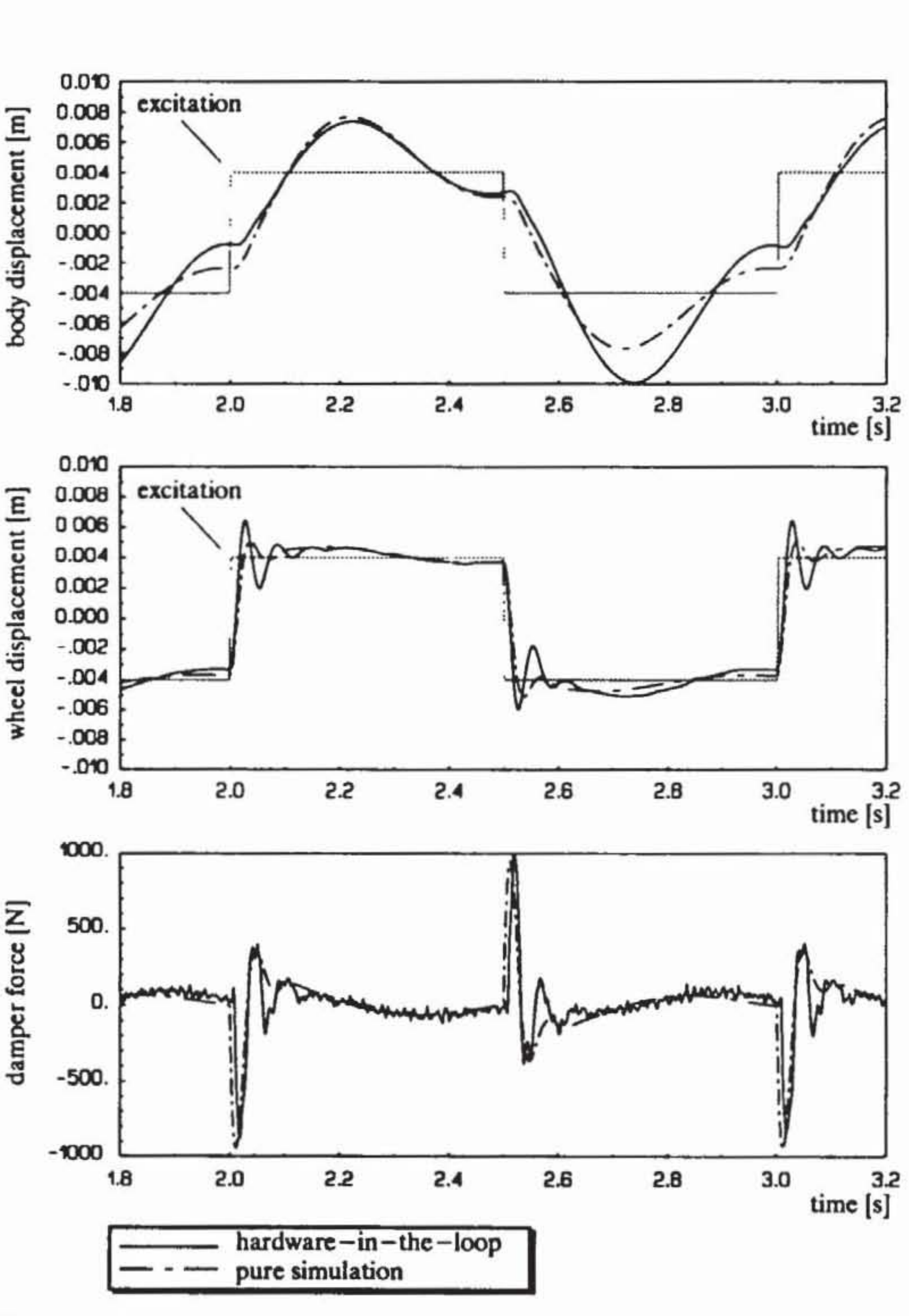

Figure 13: Motion and damper force of vehicle model, hardware-in-the-loop vs. pure simulation

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