Pages 64 to 81

DEVELOPMENT OF SEMI-ACTIVE ROAD-FRIENDLY TRUCK SUSPENSIONS

W. Kortüm, Deutsches Zentrum für Luft- und Raumfahrt e.V. (DLR) Institute for Robotics and System Dynamics, Wessling, FRG
M. Valasek, Dept. of Mechanics, Fac. of Mech. Eng., Czech Technical University in Prague, CZ

Abstract:
The paper describes the results of the EC COPERNICUS-Project SADTS (= Semi-Active Damping of Truck Suspensions and its Influence on Drivers and Road Loads). This research project investigated the potential of semi-active truck suspensions optimised for road-friendliness. The improvements are compared to modern passive air springs.

1 INTRODUCTION
The interaction between heavy vehicles and its infrastructure (roads or bridges) has recently received increased attention. The static values of road-tyre forces are being regulated by the present standards. However, the results of the OECD project DIVINE [1] have again proved that the dynamic part of road-tyre forces causes significantly increased damage of roads and increased loading of bridges. The recommended actions are to modify the existing passive vehicle suspension, e.g. to replace the steel leaf springs by air springs. Such suspensions are termed as to be road-friendly and perhaps bridge-friendly. This paper deals with the results of the EC project COPERNICUS SADTS (Semi-Active Damping of Truck Suspensions and its Influence on Drivers and Road Loads). There are four partners: Deutsches Zentrum für Luft- und Raumfahrt e.V. (DLR), FRG, Czech Technical University in Prague, Czech Republic, Slovak Technical University, Slovak Republic and SKODA-LIAZ Truck Company, Czech Republic. The objectives of the project were to develop the methodology and software tools for the design and evaluation of semi-active damping of truck suspensions, to verify them on the application to a SKODA-LIAZ truck prototype (**Fig. 1**) with semi-active damping and to investigate its influence on drivers and road loads. The goal is to decrease road damage as well as drivers' comfort for a broad range of road unevenness by a suitable choice of control law structure and parameters. The reference for judging upon the possible improvement of semi-active suspensions is a well designed passive air suspension system which is known to be advantageous with respect to comfort and road friendliness as compared to the traditional leaf springs.

2 REFERENCE TRUCK AND FEASIBLE ROAD-FRIENDLY SUSPENSION CONCEPTS
Active and semi-active vehicle suspensions have been studied for a long time, [2-7]. The applied performance criterion in these studies has been mostly the ride comfort. Only few studies have investigated the road-tyre forces specifically [6, 8].

The evaluation criteria are road friendliness and comfort for driver and load. The Dynamic Load Stress Factor (DLSF), [6], is taken as an evaluation criterion of the road damage

$$DLSF = 1 + 6DLC^2 + 3DLC^4, \tag{1}$$

where DLC (Dynamic Load Coefficient) is

$$DLC = \frac{RMS \text{ dynamic tyre force}}{\text{static tyre force}}. \tag{2}$$

It is required to reduce this criterion. The ISO weighted acceleration RMS is used for the vibration comfort criterion; it is required - at least - not to worsen it in comparison with the passive suspension.

There are many variants of such advanced suspensions differing by the sensors, actuators and control laws. The standard sensors which are available on the SKODA-LIAZ prototype are the accelerations of sprung and unsprung masses and the suspension deflections on all axles. These sensors enable in principle to reconstruct all states of the dynamic model, [9], although some approaches require only the suspension deflection, [10]. The actuators exist of either active force generators or semi-active controlled shock absorbers, [11]. The energy consumption and the availability of variable shock absorber on the market have led to the concept of semi-active suspension. The largest variety of different concepts is due to the possible choice of suitable control laws; this is treated in more details in Section 4.

The truck prototype has been chosen as a three axle platform truck with middle driven axle and air springs. From the point of view of passive suspension it is already a road friendly design. The truck has been modified for four controlled dampers on the driven axle and the necessary sensors. The sensors used for feedback are accelerations of unsprung and sprung masses at front and rear driven axle on both sides, the suspension deflections at these axles and the acceleration at the cabin.

The truck prototype, Fig.1, has been carefully modelled with respect to the important degrees-of-freedom (heave, pitch and roll), simulated in SIMPACK [12] and validated with experiments, [13], **Fig. 2**.

In particular, all suspension elements (air and leaf springs, tyres) and damping elements are treated as nonlinear. The truck is considered fully loaded passing the road with the velocity of 72 km/hour. The parameter variations include the half loaded and empty truck and velocity 54 and 84 km/hour.

The reference input is a good stochastic road, although many other input signals are being used for the control design itself.

3 DEVELOPMENT STRATEGY AND SOFTWARE TOOLS

The vehicle suspension of the prototype is substantially nonlinear especially because of the unsymmetric nonlinear characteristics of the shock absorber with saturation. This means that the well established control synthesis procedures for linear systems cannot be applied. There exist some control synthesis procedures for nonlinear systems (e.g. exact input/output linearization, sliding mode control) but in this case the knowledge of the plant nonlinearities cannot be used directly. Therefore several control concepts were investigated [14] and specifically adapted to the nonlinear system under investigation. Their development and comparison have been performed on the verified 3D simulation model in contrast to many approaches based only on a linearized quarter car model.

There are many possible control schemes, see section 4, which can be applied for this purpose. In order to make the selection procedure as much objective as possible a reference simulation model of the truck prototype, a reference input road disturbance and reference evaluation of the performance criteria were prepared, [14].

Since no closed theory on the analysis and design of semi-active control laws for complex nonlinear plants exists (with the exception of bilinear theory [15] and the proposal to use "clipped" optimal for linear plants, [16]) at the beginning of the project a design methodology and appropriate software tools had to be selected and implemented.

Because of its capabilities, open structure and various existing interfaces to common CAE-Software (as CAD and FEM), the general purpose multibody system package SIMPACK was selected as the base tool for simulation and evaluation, [17, 12].

In short SIMPACK is an advanced simulation package resting on efficient algorithms for the automatic generation of the equations of motion for flexible and controlled multibody systems. SIMPACK is equipped with advanced solvers for differential-algebraic equations in time - and frequency domain.

Figure 3 shows SIMPACK with its interfaces to some common CAE-packages. (Note that the bi-directional interface to CACE is partially a result of the COPERNICUS project.).

The second basic decision was to use MATLAB [18] for the synthesis of control laws. MATLAB is a widely used environment for general computation and in particular for control system analysis and synthesis. SIMULINK, a MATLAB extension, is an efficient tool for modelling, analysis and simulation of nonlinear systems especially when the systems are defined in some block-oriented form. Both systems, SIMPACK and SIMULINK, have been connected by the SIMAT interface [17] into an integrated environment which enables to advantageously make use of both systems, i.e. the simulation of complex nonlinear multibody systems and the design and simulation of control systems.

With this interface, the standard control design procedures of the MATLAB toolboxes can be applied to the linearized or nonlinear models provided by SIMPACK. Since mostly the design models are simplified for control law synthesis, the performance based on the full nonlinear 3D-model is then checked via SIMPACK or SIMULINK (strategy I).

Because of the intrinsic nonlinearities of the reference model and the semi-active damper concept a second procedure (strategy II) directly uses the full nonlinear model (perhaps with some degrees of freedom neglected for design), supposes a meaningful feedback law (structure) with free parameters and uses multiobjective parameter optimisation for determining best parameters in order to minimise the performance measures (Pareto-optimality), [17]. Since this strategy can use the full nonlinear simulation models in the design loop, it is frequently called "design-by-simulation".

As optimisation packages two options have been realised: MOPS at DLR, [17, 19] and UFO at Czech Technical University, [20].

4 CONTROL CONCEPTS

4.1 Investigated Control Concepts

Within the SADTS project a number of control concepts have been investigated:

- **LQR or RICCATI** control has been used as a standard approach for comparison. (It is interesting that the choice of state variables is in this case critical because only relative co-ordinates can be reconstructed and the form of velocity state (relative or absolute) is decisive for the way of entrance of unmeasured disturbances (road velocity or road acceleration)).
- **Neural Networks** have been trained to the realisation of the control obtained as LQR for linearized systems; linearizsation has been performed with respect to various states.
- **Fuzzy Control** has enabled to choose the damping level as the direct function of sensor measurements by a simple optimisation selection in several reference simulations.

  Objective fuzzy is another non-traditional approach towards fuzzy control. Fuzzy control is used as an efficient approximation tool of control law nonlinearly dependent on the state value. The fuzzy approximation is determined by direct parameter optimisation of a quadratic performance criterion by the choice of control law parameters.
- **Sliding Mode Control (SMC)** is another nonlinear control law strategy promising parameter insensitive performance. The sliding surface has been selected as LQR control for minimising road-tyre forces. SMC has proved better robustness in clamping semi-active control from active as compared to LQR. However, because SMC leads to minimised transition phase of control similar to suboptimal bang-bang control the resulting peaks of forces may be high.
- **Extended Ground Hook (EGH)** is a new control concept in analogy to sky-hook which is described in more details in section 4.2.

4.2 Extended Ground-Hook

From the above concepts we shall describe here in more details the EGH approach which achieved good results and which demonstrates clearly the development strategy used for the control synthesis of nonlinear system of semi-active truck suspension.

The important principle for the ride comfort is the so called "sky-hook" (feedback of absolute velocity) developed by Karnopp, [2]. Despite many different theories of the optimal suspension control the sky-hook control concept is generally used for two reasons: as the "ideal" control concept for comparison with other approaches and as the basis for practical implementation of semi-active or active vehicle suspensions.

The novel control concept, the so called ground-hook, as a fictitious damping element between the wheel and the ground parallel with the tyre has been introduced [21, 22, 23]. The motivation of this concept is to develop an equivalent of sky-hook for the reduction of dynamic tyre-road forces. The preservation of low accelerations of sprung mass has been achieved by the combination of sky-hook and ground-hook and ground-hook extensions.

The basic ground-hook concept has been then extended to the several variants and implemented to the semi-active suspension. The suitable choice of control law structure and parameters enables to decrease criteria of road damage as well as ride comfort for a broad range of road unevenness.

For gaining some physical insight the further considerations are based on the simple linear quarter car model. Please note that our design methodology is not restricted to such models. Moreover, the judgement of performance is always based on the nonlinear 3D-model as mentioned above.

The model of a linear quarter car model is on the **Fig. 4**.

The equations of motion for this quarter car model are

$$m_1\ddot{z}_1 + k_{10}(z_1 - z_0) + b_{10}(\dot{z}_1 - \dot{z}_0) - k_{12}(z_2 - z_1) + F_d = 0, \tag{3}$$

$$m_2\ddot{z}_2 + k_{12}(z_2 - z_1) - F_d = 0, \tag{4}$$

where m_1 is the unsprung mass, m_2 is the sprung mass, k_{12} is the stiffness of the suspension, k_{10} is the stiffness of tyre, b_{10} is tyre damping constant and F_d is the force of passive or semi-active damper or active element. The meaning of geometric quantities z_2, z_1 *and* z_0 is also clear from this figure. The control law of extended ground-hook in combination with sky-hook is assumed as

$$F_d = b_1(\dot{z}_1 - \dot{z}_0) - b_2\dot{z}_2 - b_{12}(\dot{z}_2 - \dot{z}_1) + \Delta k_{10}(z_1 - z_0) - \Delta k_{12}(z_2 - z_1). \tag{5}$$

Equation (5) contains all terms of the full state feedback with the exception of an absolute position term like z_0, z_1 or z_2. The advantage of this feedback law is that all terms are either directly measurable or reconstructable from other measurements.

Directly measurable is $z_2 - z_1$. The velocities $\dot{z}_1$ and $\dot{z}_2$ are observable from the measurements of accelerations $\ddot{z}_1$, $\ddot{z}_2$. By adding the equations (1) and (2) the tyre-road force is obtained:

$$F_{10} = m_1\ddot{z}_1 + m_2\ddot{z}_2, \tag{6}$$

which can also be expressed as

$$F_{10} = k_{10}(z_1 - z_0) + b_{10}(\dot{z}_1 - \dot{z}_0). \tag{7}$$

Because the damping of the tyre b_{10} has very low values in the case of typical relative velocities of trucks, it can be neglected; the tyre deformation $z_1 - z_0$ can be computed from equation (3) and by differentiation the last term $\dot{z}_1 - \dot{z}_0$ can be obtained.

Further we will be concerned with a real system incorporating an active element. This element is located at the place of a passive shock absorber (**Fig. 5** b). The control law for this active element is given by the equation (5). The desired force of this active element is the basis for the real semi-active device. The semi-active device is the variable shock absorber used instead of the active element. The damping rate of variable shock absorber can be set to the damping rate b_{sa} which gives the force value nearest to the desired one

$$b_{sa} = \begin{cases} b_{\min} & \text{if } F_{act} \le b_{\min}(\dot{z}_2 - \dot{z}_1) \\ \dfrac{F_{act}}{\dot{z}_2 - \dot{z}_1} & \text{if } b_{\min}(\dot{z}_2 - \dot{z}_1) \le F_{act} \le b_{\max}(\dot{z}_2 - \dot{z}_{1)} \\ b_{\max} & \text{if } b_{\max}(\dot{z}_2 - \dot{z}_1) \le F_{act} \end{cases} \tag{8}$$

By variation of the parameters b_1, b_2, b_{10}, Δk_{10}, Δk_{12} a large variety of control laws can be obtained. For the systematic determination of these parameters multi-objective parameter optimisation (MOPO) was applied using representative excitations of the truck.

The following results have been obtained using the program package UFO, [20] and as input disturbance a cosine bump.

The performance criterion used here was the time integral of dynamic tyre force

$$I = \int_{0}^{T_{l\arg e}} F_{10}^{2} dt \,. \tag{9}$$

The results of such an optimisation are shown in **Fig. 6**, where the non-optimised passive, optimised semi-active and active suspension responses of tyre-road force to the cosine bump are plotted. These results confirm the development of extended ground-hook principle. As can be seen, the behaviour of semi-active suspension is worse than the fully active one. This demonstrates that there is substantial difference between the response of an active and a semi-active suspension for a bump disturbance.

Semi-active suspension has been optimised for different values of damping rate limits $(b_{\min}, b_{\max})$ and time constants of variable shock absorber and their interaction has been investigated. Finally, the optimisation has been computed for the values $b_{\min} = 60000\,Ns/m$, $b_{\max} = 300000\,Ns/m$ and the time constant 10 *ms*.

Some results for the reference 3D model to very good stochastic road are shown at the **Fig. 7**, where the following symbols are used

- Passive shock absorber COM,
- Passive optimised shock absorber PAO,
- Semi-active ideal shock absorber (with zero time constant) SAI,
- Semi-active real shock absorber (with 10 ms time constant) SAR.

4.3 Nonlinear Extended Ground-Hook

The EGH in the previous section has linear terms with constant gains although these gains have been adapted by MOPO for nonlinear system and appropriate excitations. The severe nonlinearity of the plant can be taken into account by nonlinear state dependent gains. The state space has been divided into regions (small and large, positive and negative shock absorber velocity). In each region different values of gains in EGH law are being considered. They are determined by MOPO approach. The results are excellent. The comparison of results for the ramp are shown in **Fig. 8** and for the stochastic road in **Fig. 9**, [24].

5 FURTHER ACTIVITIES; CONCLUSIONS

5.1 Implementation on the Prototype

The semi-active control of truck suspension is being implemented presently on the SKODA-LIAZ truck prototype (Fig. 1). Four variable shock absorbers are used (Fichtel&Sachs) at the rear driven axle. The truck is equipped with nine accelerometers (axle and chassis front and rear, left and right, driver deck), four suspension distance sensors and two air spring pressure sensors. The data from these sensors will be processed by a controller based on INTEL 196 processors and PC computer with ADVANTECH PCL-818 measuring card. The control processors will use the data from rear driven axle (accelerations and suspension distance) and will control the control unit of variable shock absorbers. This prototype implementation will not use the data for possible control adaptation (loading, speed, manoeuvre, etc.) yet.

5.2 Conclusions

There are two major areas of conclusions from the research project at the present time (December 97):

1. Methodology-Software Tools

Despite of choosing well proven, advanced software tools for modelling and simulation (SIMPACK) and for control law synthesis (MATLAB) as well connecting them by the SIMAT interface, a rather general strategy for the analysis and design of semi-active control concepts has been developed and tested. As a summary of this procedure the following main steps have to be performed.

- Develop a control law (structure) based on physical insight into the plant properties and control potentials.
- Determine the parameters (gains) by multi-objective parameter optimisation (MOPO approach) using representative excitations (manoeuvres).
- In case of highly nonlinear plants the control parameters (gains) may be chosen as state-dependent again being determined by the MOPO approach.
- In case of simplifications in the design model, apply the full nonlinear simulation model for final performance evaluation.

2. Road Friendliness

The results so far can be described as follows: the semi-active damping of truck suspension enables to reduce the road damage factor (DLSF) by about 5-10 % in comparison with air spring suspension which is already treated as road-friendly [1]. Verification of these predictions are presently prepared through implementation of the selected semi-active shock absorbers and preparing the experiments.

Introduction Content

Introduction Content

REFERENCES

[1] Proceedings of DIVINE Concluding Conference, Rotterdam 1997

[2] Karnopp, D., Crosby, M.J., Harwood, R.A.: Vibration Control Using Semi-Active Force Generators, Transactions of ASME, J. of Engineering for Industry, 96(1974), pp. 619-626

[3] Sharp, R.S., Crolla, D.A.: Road Vehicle Suspension System Design - a Review, Vehicle System Dynamics, 16(1987), pp. 167-192

[4] Venhovens, P.J.: Optimal Control of Vehicle Suspensions, PhD Thesis, Delft University of Technology, 1993

[5] Elbeheiry, E.M., Karnopp, D.C., Elaraby, M.E., Abdelraaouf, A.M.: Advanced Ground Vehicle Suspension Systems - a Classified Bibliography, Vehicle System Dynamics, 24(1995), pp. 231-258

[6] Yi, K., Hedrick, J.K.: Active and Semi-Active Heavy Truck Suspensions to Reduce Pavement Damage, SAE Technical Paper 892486, 1989

[7] Edge, C.Y., Tsao, Y.J.: A Fuzzy Preview Control Scheme of Active Suspension for Rough Road, Int. J. of Vehicle Design, 15(1994), 1/2, pp. 166-180

[8] Cebon, D.: Vehicle-Generated Road Damage: A Review, VSD, 18(1989), pp. 107-150

[9] Hedrick, J.K., Rajamani, R., Yi, K.: Observer Design for Electronic Suspension Applications, VSD 23(1994), pp. 413-440

[10] Bode, O. et al.: Impact of Different Tyres and Adaptive Suspension Control on Vertical Dynamics of Commercial Vehicles, In: Proc. of AVEC 96, Aachen 1996, pp. 119-140

[11] Irmscher, S., Hees, E.: Experiences in Semi-active Damping with State Estimators, In: Proc. of AVEC 96, Aachen 1996, pp. 193-206

[12] Kortüm, W., Rulka, W., Spieck, M.: Simulation of Mechatronic Vehicles with SIMPACK , MOSIS 97, Ostrava, Czech Republic, 1997

[13] Valasek, M., Stejskal, V., Sika, Z., Vaculin, O., Kovanda, J.: Dynamic Model of Truck for Suspension Control, In: Paper Summaries of 15th IAVSD Symposium, Dynamics of Vehicles on Roads and Tracks, Budapest 1997, pp. 60 - 62.

[14] Valasek, M. et al.: Development of Semi-Active Truck Suspension, In: Proc. of IFAC Symposium on Transportation Systems, Chania 1997, pp. 470-475

[15] Hrovat, D., Margolis, D.L., Hubbard, M.: An Approach Toward the Optimal Semi-Active Suspension, Trans. ASME, Journal of Dynamic System, Measurement and Control, 110 (1988), pp. 288 - 296

[16] Tseng, H.E., Hedrick, J.K.: Semi-Active Control Laws - Optimal and Sub-Optimal, VSD, 23 (1994), pp. 545 - 569.

[17] Vaculin, O., Kortuem, W., Schwartz, W.: Analysis and Design of Semi-Active Damping in Truck Suspension – Design-by-Simulation, In: Proc. of AVEC 96, Aachen, pp. 1087-1103

[18] MATLAB, SIMULINK, User Guides, The MathWorks, Inc., 1995

[19] Gretzschel, M., Bals, J., Mauer, L., Schwartz, W.: SIMPACK-MOPS, ein neues strategisches Verbundwerkzeug zur Konzeption, Auslegung und Optimierung von Schienenfahrzeugen. VDI-Tagung "Systemoptimierung im spurgeführten Verkehr". September 1997, Munich, Germany

[20] Luksan, L. et al.: UFO - Interactive System for Universal Functional Optimization, Institute of Computer Science of Academy of Sciences of the Czech Republic, Technical Report No.599, Prague 1994

[21] Valasek, M., Novak, M.: Ground Hook for Semi-Active Damping of Truck's Suspension, Proc. of CTU Workshop 96, Engineering Mechanics, CTU Prague, Brno 1996, pp. 467-468

[22] Novak, M. and Valasek, M. (1996): A New Concept of Semi-Active Control of Truck's Suspension, In: Proc. of AVEC 96, Aachen, pp. 141-151

[23] Valasek, M. et al.: Extended Ground-Hook – New Concept of Semi-Active Control of Truck Suspension s, VSD, 27(1997), pp. 141-151

[24] Valasek, M., Stejskal, V., Sika, Z., Vaculin, O.: Control Concepts of
[25] Semi-Active Damping of Truck Suspension for Road Friendliness, Poster at 15th IAVSD Symposium, Dynamics of Vehicles on Roads and Tracks, Budapest 1997.

Introduction

Content

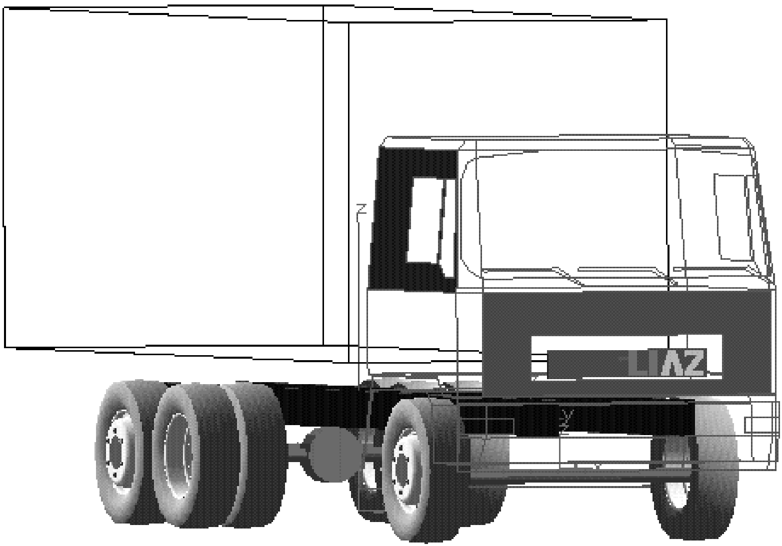

Fig. 1 SKODA-LIAZ Truck Prototype

Return to paper

Content

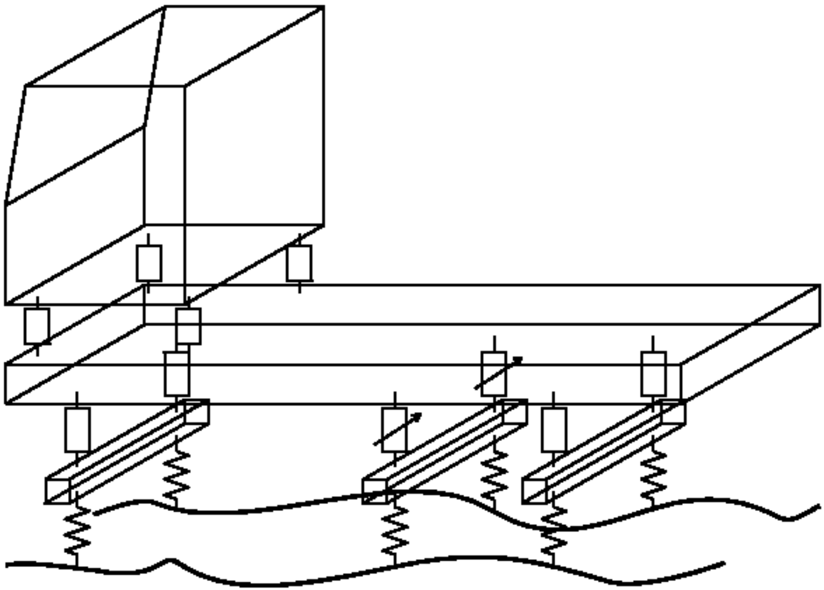

Fig. 2 Reference 3D Simulation Model

Return to paper

Content

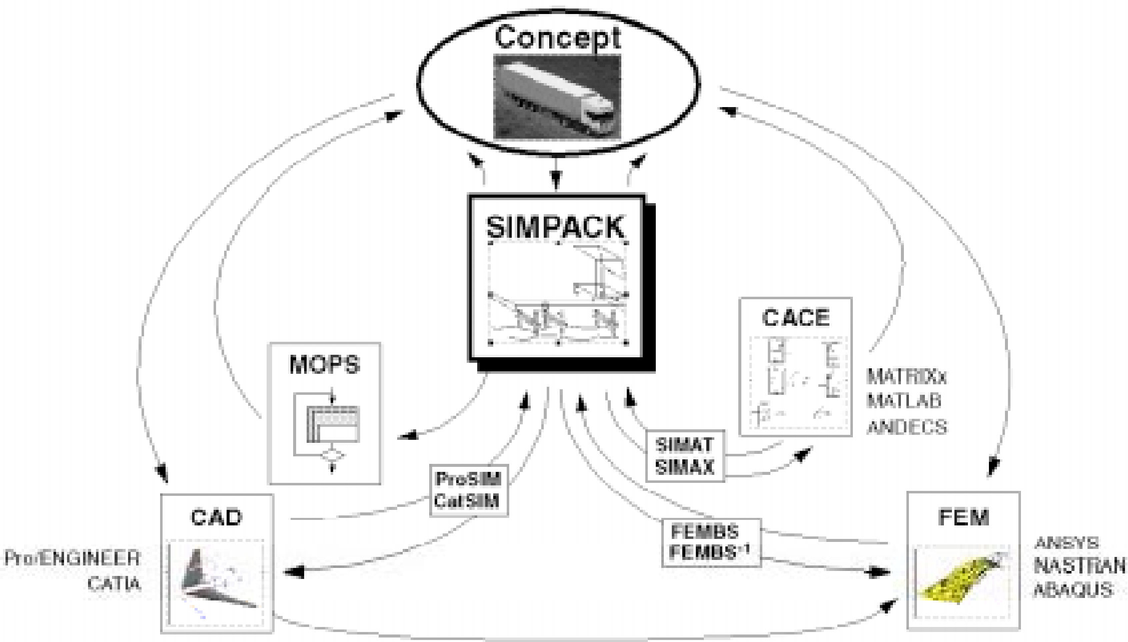

Figure 3: Interfaces of SIMPACK to Related CAE-Tools

Return to paper

Content

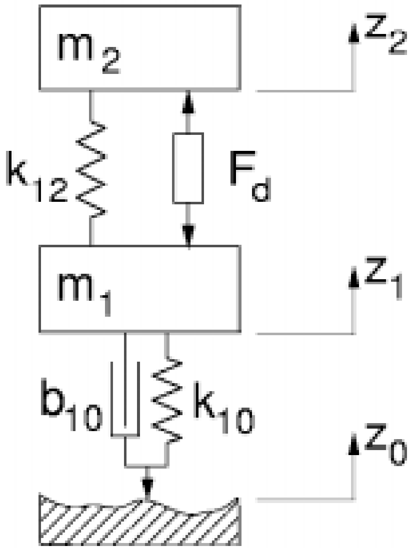

Fig. 4 Quarter Car Model

Return to paper

Content

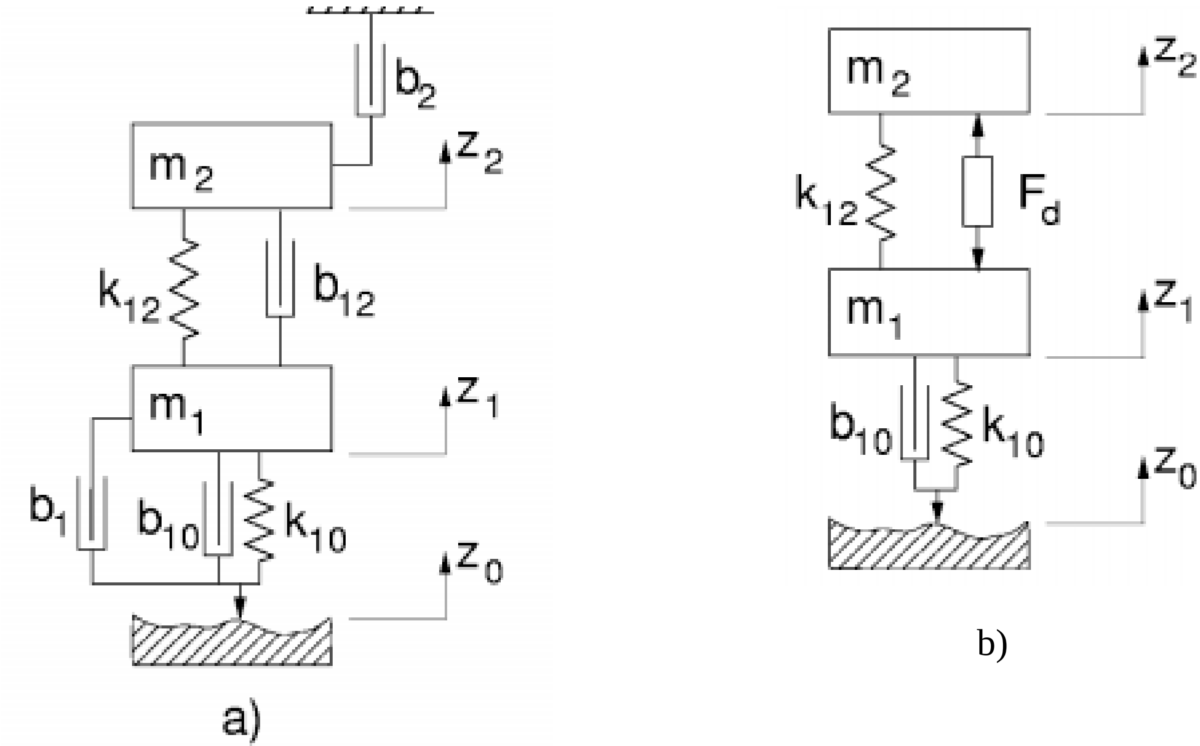

Fig. 5 Combination of Sky-Hook, Ground-Hook and Passive Suspension
a)Ideal Concept b)Realization of Concept

Return to paper

Content

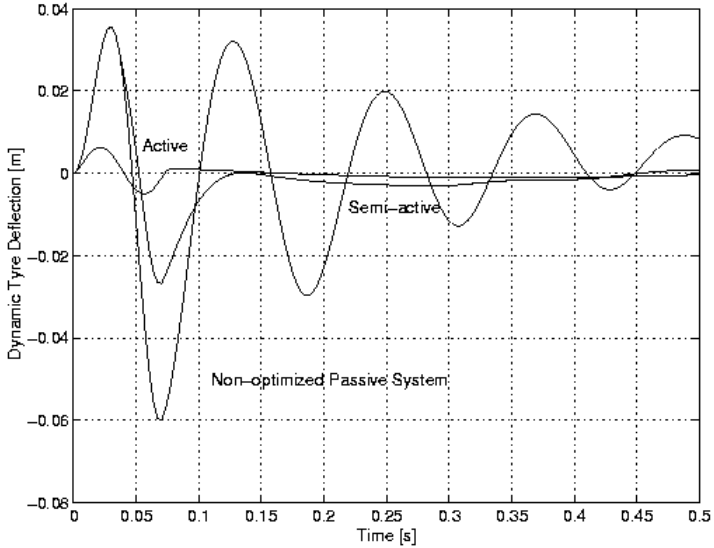

Fig. 6 Response to cosine bump of optimised active and semi-active and non-optimised passive system

Return to paper

Content

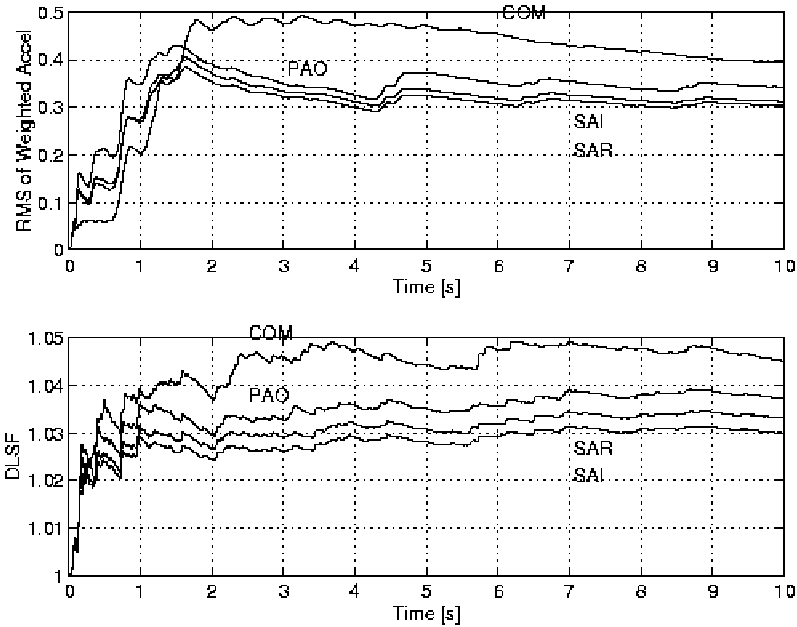

Fig. 7 Extended Ground-Hook Results

Return to paper

Content

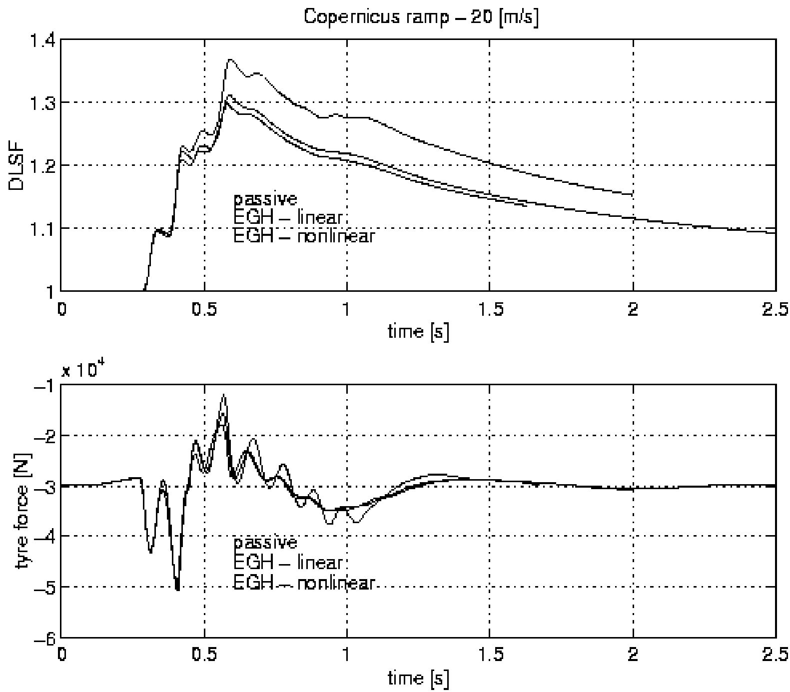

Fig. 8 Linear and Nonlinear Extended Ground-Hook Results for a Ramp

Return to paper

Content

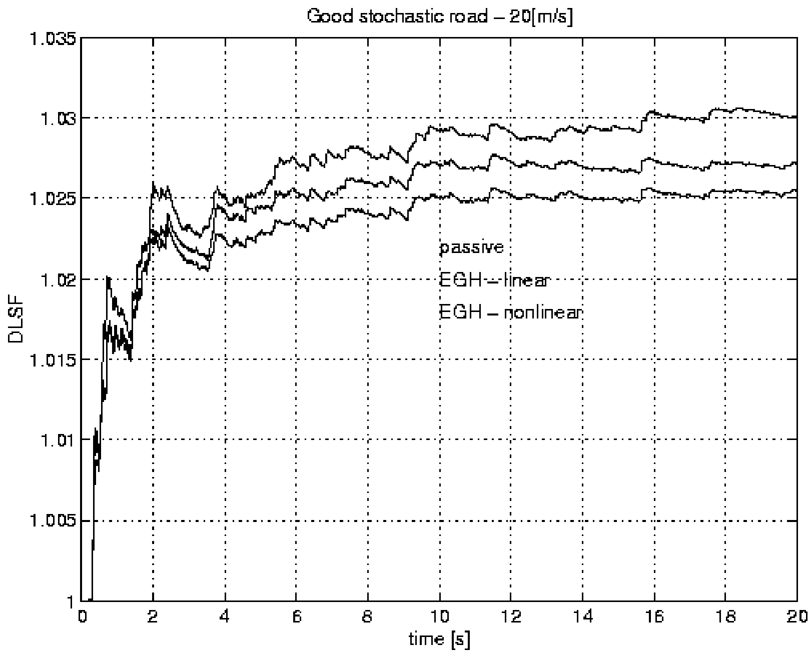

Fig. 9 Linear and Nonlinear Extended Ground-Hook Results for a Stochastic Road

Return to paper

Content