# 13

# Semi-Active Suspension Systems II




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The concept of suspension systems is treated for a broad array of vehicles including trains, airplanes (during ground motion), and off-road vehicles. The vehicle suspension has many important functions:

- Control of the attitude of the vehicle body with respect to the road surface
- Control of the attitude of the wheels with respect to both the road surface and the vehicle body
- Isolation of the vehicle body from forces generated by the roadway unevenness
- Control of the contact forces between wheels and the road surface
- Control of lateral and longitudinal motions

All these functions can be significantly improved via electronic control added into the system. This makes the suspension "active." In this section the treatment of suspension is limited just to the vibration control. We primarily focus on the isolation function and the contact force variation control (in particular, the normal component of it).

The lexicon commonly used in vehicle dynamics "semi-active" control implies that the control actuator requires very little power. Such control is where the actuator possesses many attributes of conventional (active) control but which requires very little control power. A semi-active actuator typically dissipates energy, thus it does not raise stability concerns.

## 13.1 Concepts of Semi-Active Suspension Systems

### 13.1.1 Karnopp's Original Concept

The fundamental concepts of semi-active suspension and semi-active vibration control go back to Karnopp's work.[1–3] For the suspension elements, which are electronically, controlled, a critical issue

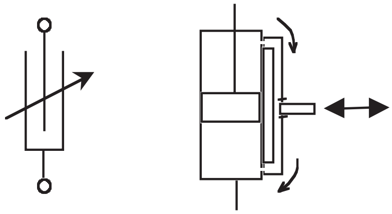

**FIGURE 13.1** Variable damper and the working principle.

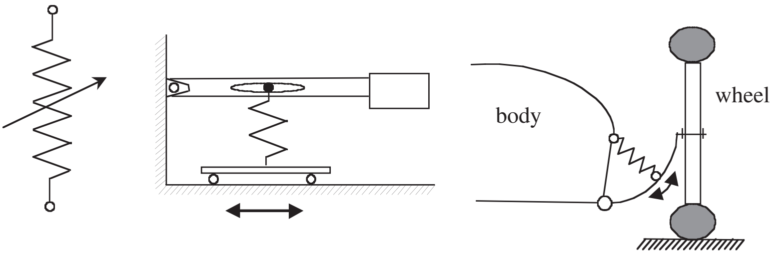


**FIGURE 13.2** Variable force transfer and the working principle.

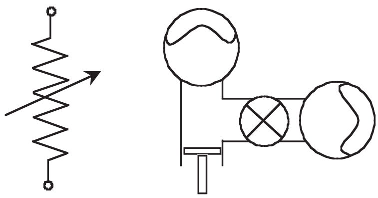

**FIGURE 13.3** Variable spring stiffness and the working principle.

is the power consumed. The semi-active element must be either dissipative or conservative when it comes to their energy needs. There are a number of general classes of such devices.

The first class is the variable resistors, which dissipate energy. In a typical vehicle suspension it is the variable damper. Constitutive laws between the system variables of force and velocity characterize these elements. These relations can be rapidly altered using a control input, which consumes very little power (Figure 13.1). Practically, it is conceived as a variable orifice viscous damper. By closing or opening the orifice the damping characteristics change from soft to hard and vice versa. Recently, this flow control has been achieved using electro- and magneto-rheological fluids and is available as industrial products.[39]

The second class is variable force transformers, which conserve energy between suspension and spring storage. Within the vehicle's suspension it is the variable lever arm. These elements are characterized by controlled force variation, which consumes minimal power (Figure 13.2). The physical materialization is conceived as a variable lever on which the force acts. By moving the point of force application, the force transfer ratio change. If these points move orthogonally to the acting force, theoretically no mechanical work is involved in control.

The third class of semi-active components exhibits a variable stiffness feature, which again dissipates energy. These elements are characterized by a variable free length of a spring, which is changed deploying minimal control power (Figure 13.3). For this hydropneumatic spring, if the valve is shut, only one volume is connected and the spring is stiff. When the valve is open, both volumes are connected and the spring is soft. During switching of the valve opening, the pressure in the chambers is equalized and the accumulated energy is dissipated.

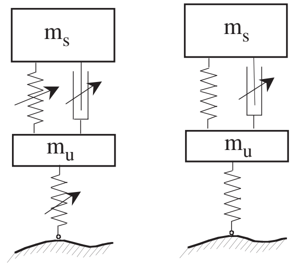


**FIGURE 13.4** Semi-active vehicle suspension, (a) theoretical possibility (b) current practice.

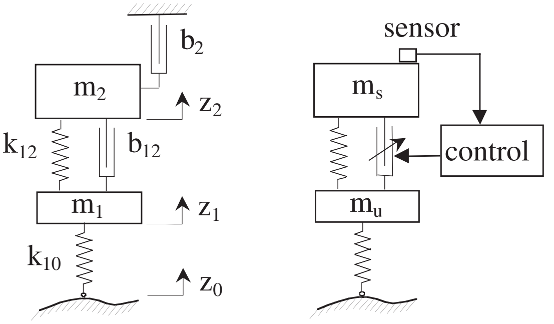


**FIGURE 13.5** Sky-hook — ideal concept (a) and realization (b).

Using these semi-active devices the properties of vehicle suspension can be controlled according to the scheme on Figure 13.4 where $m_s$ represents sprung mass and $m_u$ unsprung mass. Theoretically, all elements of vehicle suspension can be adjusted (Figure 13.4a), but generally the semi-active damper feature for the shock absorber is controlled (Figure 13.4b).

## 13.1.2 Sky-Hook for Comfort

The initial concept the control of both semi-active and active suspensions originates again in Karnopp's work[1] and was developed by many other authors (bibliographic references are in Sharp and Crolla[4] and Elbeheiry[5]). The initial aim of controlled vehicle suspension is driver (passenger) comfort. This performance index is equivalent to the minimization of sprung mass acceleration (or its filtered form) with respect to the inertial space.[38] This proposition yields Karnopp's idea of a sky-hook. Sky-hook is a fictitious damper between the sprung mass and the inertial frame (fixed in the sky) (Figure 13.5). The damping force of this fictitious damper reduces the sprung mass vibration.

Further design considerations are based on the simple quarter-car model (Figure 13.6). Despite its simplicity, it covers the basic properties of suspension dynamics of a real vehicle. To introduce the control concept a linear quarter-car model is used. The nonlinearities of the structure are also taken into account in the text. The equations of motion of the quarter car model in Figure 13.6 are

$$\begin{aligned} &m_1\ddot{z}_1 + k_{10}(z_1 - z_0) + b_{10}(\dot{z}_1 - \dot{z}_0) - k_{12}(z_2 - z_1) + F_d = 0 \\ &m_2\ddot{z}_2 + k_{12}(z_2 - z_1) - F_d = 0 \end{aligned} \tag{13.1}$$

where $m_1$ is the unsprung mass, $m_2$ is the sprung mass, $k_{12}$ is the stiffness of the main spring, $k_{10}$ is the stiffness of the tire, $b_{10}$ is the tire damping constant (usually negligible), and $F_d$ is the force of the passive or semi-active damper or active element.

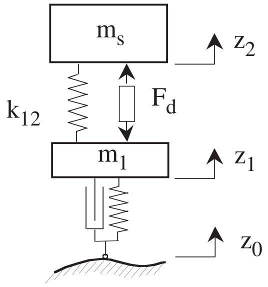


**FIGURE 13.6** Quarter car model.

Figure 13.5a represents a fictitious case. Added parallel damper $b_2$ tries to complement $b_{12}$ for the sprung mass. Figure 13.5b represents the realization of this concept. The fictitious force computed from the added sky-hook damper is applied by the actuator $F_d$. The actuator can be a fully active element (active force generator) or a semi-active element (variable shock absorber). The ideal active force $F_d$ of this element according to the sky-hook control law is

$$F_d = -b_2 \dot{z}_2 \tag{13.2}$$

If this force is directly applied by an active force generator, then the process becomes the active sky-hook suspension. For semi-active suspension, this force is limited to the range of forces applicable by the semi-active devices. For a semi-active damper, this is done by the transformation from the required (active) force $F_{act}$ to certain settings of the damping rate $b_{semi\text{-}active}$ such that the damping force is nearest to the desired value. For an ideal linear variable shock absorber, the damping rate $b_{semi\text{-}active}$ is set for the interval ($b_{min}$, $b_{max}$) as a linear saturation function:

$$b_{semi-active} = \begin{matrix} b_{max} & if & b_{max} < b_{act} \\ b_{act} & if & b_{min} < b_{act} < b_{max} \\ b_{min} & if & b_{act} < b_{min} \end{matrix} \tag{13.3}$$

$$b_{act} = \frac{F_{act}}{\left(\dot{z}_2 - \dot{z}_1\right)} \tag{13.4}$$

The semi-active sky-hook suspension is then realized by taking $F_{act} = F_d$ from (13.2) into (13.3) and (13.4) into (13.1). A real variable shock absorber is, however, nonlinear and time dependent with internal dynamics and the transformation (13.3)–(13.4) must reflect that.

The other problem is the usage of suitable sensors. The direct measurement of sprung mass velocity is usually not possible. Therefore, the acceleration sensor is used and the velocity is obtained by time integration after suitable filtering. Another new concept is the usage of acceleration feedback instead of velocity.[6]

The response of a nonlinear quarter car model with sky-hook control is given in Figure 13.7, for which a realistic model of the nonlinear damper is used and its response to chirp signal is observed. The responses of nonlinear semi-active damper and passive cases are compared.

### 13.1.3 Extended Ground-Hook for Road-Tire Forces

The road-tire forces are considered next as another performance index. Similar principle to the sky-hook called ground-hook is developed for this case.[7,8] The road-tire forces are proportional to

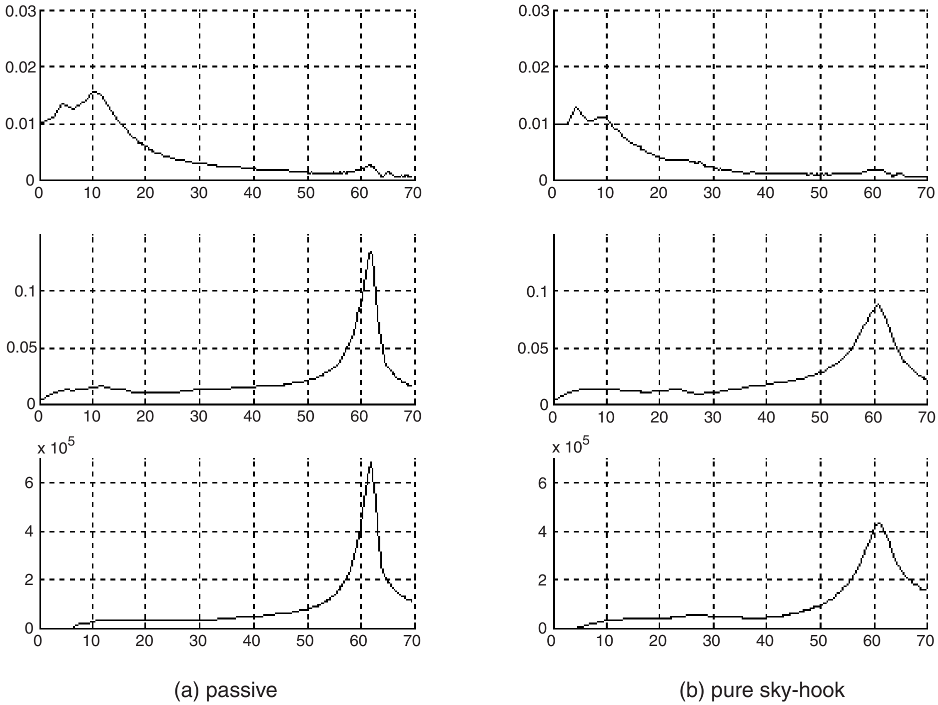


**FIGURE 13.7** Comparison of response of passive suspension and pure sky-hook to the chirp signal in rad/s (figures of sprung mass, unsprung mass responses, road-tire forces).

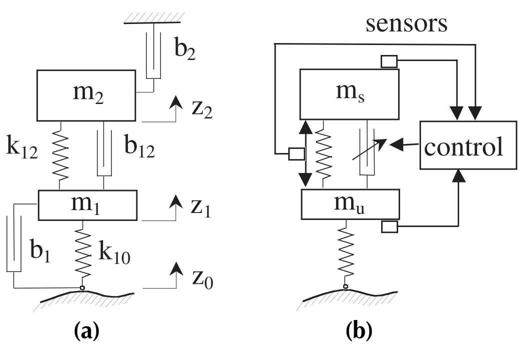


**FIGURE 13.8** Extended ground-hook — ideal concept (a) and realization (b)

the tire deflection. Any reduction of this deflection by increased damping also reduces the road-tire forces. This leads to the principle of ground-hook, which is depicted in Figure 13.8.

Figure 13.8a represents a fictitious case. The real damping of the tire $b_{10}$ has a very low value, and the parallel fictitious damper $b_1$ tries to add a higher damping value $b_1$ to it. Low accelerations of the sprung mass are obtained by the combination of a sky-hook and the ground-hook. Figure 13.8b represents the realization of this concept. The control force computed for this combination is applied by an actuator $F_d$. This actuator can be a fully active (active force generator) or a semi-active element (variable shock absorber). This concept was further developed into extended ground-hook. The ideal active force $F_d$ of this element is

$$F_d = b_1(\dot{z}_1 - \dot{z}_0) - b_2\dot{z}_2 - b_{12}(\dot{z}_2 - \dot{z}_1) + \Delta k_{10}(z_1 - z_0) - \Delta k_{12}(z_2 - z_1) \tag{13.5}$$

where $\Delta k_{10}$ and $\Delta k_{12}$ are additional terms for the fictitious stiffness cancellation, which is mainly important in the case of a fully active actuator. The direct application of the damping force (13.5) to the suspension in (13.1) gives the active extended ground-hook. The semi-active extended ground-hook suspension is then realized by combining $F_{act} = F_d$ from (13.5) into (13.3) and (13.4) and (13.1).

The advantage of the Equation (13.5) over standard [LQR] feedback is that all the terms are either directly measurable or reconstructable from other measurements. The sensors usually are the accelerometers on sprung and unsprung masses and the suspension displacement sensor. The value of $z_2 - z_1$ can be measured directly. The velocities $\dot{z}_1$ and $\dot{z}_2$ are observable from the measurements of accelerations, $\ddot{z}_1$ and $\ddot{z}_2$. By adding the Equations (13.1) the road-tire force is obtained as

$$F_{10} = m_1 \ddot{z}_1 + m_2 \ddot{z}_2 \approx k_{10}\left(z_1 - z_0\right) \tag{13.6}$$

If the tire damping $b_1$ is ignored, the tire deformation $z_1 - z_0$ can be solved from Equation (13.6).

By changing the parameters $b_1$, $b_2$, $b_{12}$, $\Delta k_{10}$, and $\Delta k_{12}$ a variety of modified control laws for the suspension system can be obtained. For the systematic determination of these parameters, the multi-objective parameter optimization (MOPO) or nonlinear quadratic regulations (NQR) approaches are presented below.

The parameters of the extended ground-hook were originally considered to be constants for the entire velocity interval of the shock absorber. Because the characteristics of the shock absorber are nonlinear, the corresponding extended ground-hook control with state-dependent gains (gain scheduling) is used. The strong nonlinearity of the variable shock absorber (see Figure 13.9), especially its asymmetry, can be taken into account to determine control-law parameters. Therefore, the nonlinear extended ground-hook version, which enables the state-dependent coefficients (gains) of the control law (13.5), is developed resulting in a very desirable performance.[9,10] Their dependence on the relative velocity is determined by the optimization MOPO or NQR approaches.

The response of a nonlinear quarter-car model with the pure ground-hook control is on Figure 13.10. This is obtained as a response to chirp signal. A comparison between passive (Figure 13.10a) and active cases (Figure 13.10b) can be made. Flat response of the road-tire forces especially in low frequencies is noticeable.

### 13.2.4 Semi-Active Actuators and Their Models

Some conceptual models of dampers and semi-active dampers exist in the literature (for example, Duym[11] and Spencer et al.[39]) as well as the physical models (e.g., Besinger et al.[12] and Botelle et al.[37]). For a realistic investigation of semi-active suspension we consider the conceptual model of controllable dampers.

These models should entail the nonlinear characteristics of the damping force as a function of the relative velocity and control current $F_{act} = fct(v_{rel}, i_{act})$, for the special semi-active damper given in Figure 13.9. Then it must also consider the damper control's dynamic response. The dynamic behavior caused by the response time of the valve adaptation, hydraulics, and compliance of damper mounting is modeled as a low-pass filter of the steering current with two different time constants $(\tau_{HL}, \tau_{LH})$ (see Figure 13.11.) There is also the concept of real damper control. The control law determines a commanded damper force $F_{des}$. It is transformed on the basis of actual damper velocity in the control unit into the specific commanded control current value $i_{des}$, which is applied to the variable damper.

## 13.2 Control Design Methodology

### 13.2.1 General Design Methodology

Under the support of the Copernicus SADTS (Semi-Active Damping of Truck Suspension and Its Influence on Driver and Road Loads) project, a new advanced methodology for the design

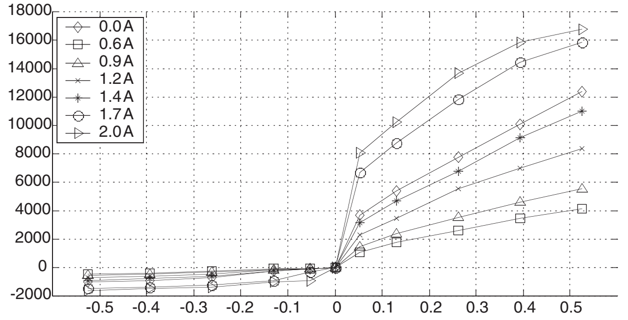


**FIGURE 13.9** Semi-active damper characteristics.

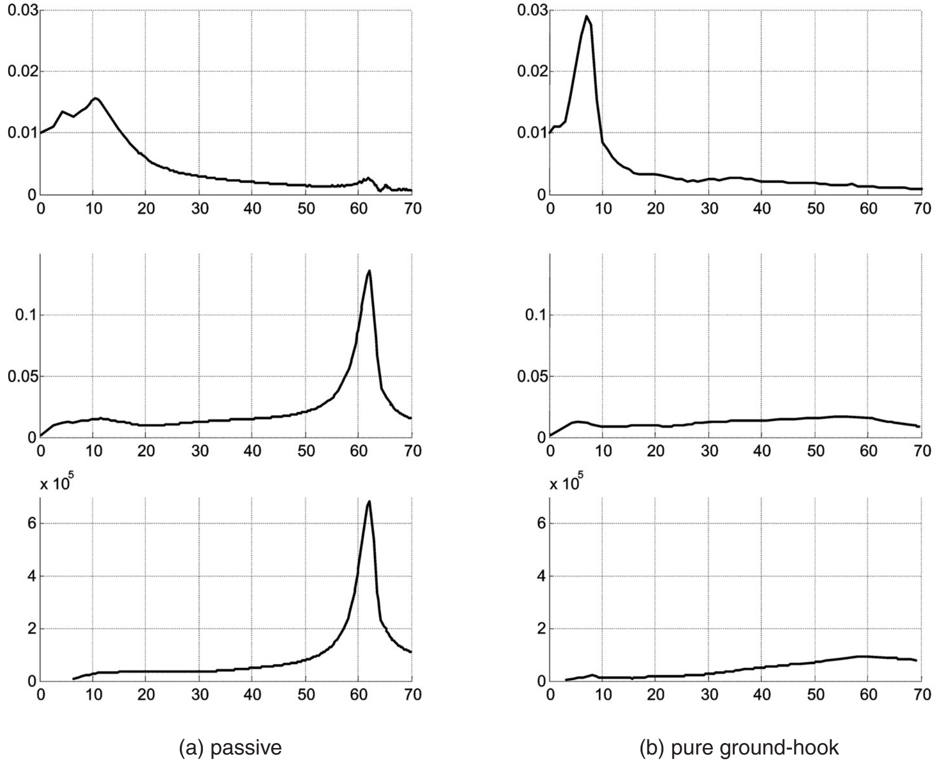


**FIGURE 13.10** Comparison of response of passive suspension and pure ground-hook to the chirp signal in rad/s (figures of sprung mass, unsprung mass responses, road-tire forces).

of semi-active truck suspensions was developed.[9,13,14] It consists of a modification over earlier approaches. It also solves the problem of whether the control design should be done on a simple linear quarter-car model or on the complex fully nonlinear 3D-vehicle simulation model.

The state-of-the-art of design methodology of controlled vehicle suspensions is based on restricted design models (quarter-car or half-car models with few degrees-of-freedom, linear kinematics, linear force laws, mostly using the same models for control design and for evaluation), linear control laws (LQG design procedure and "clipped" optimal strategy), and limited experimental verification backed only by experimental parameter tuning.

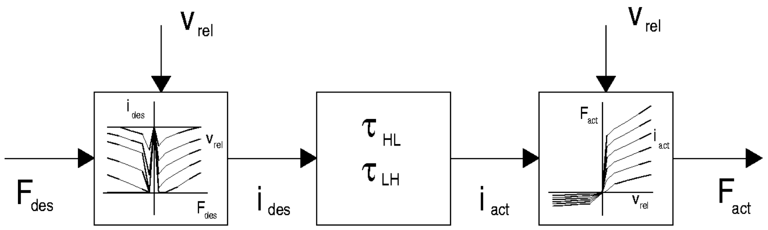


**FIGURE 13.11** Control model of semi-active damper.

The design methodology developed and applied within SADTS can be described as follows:[14]

- **Models.** Multibody system modeling enables full 3D models to simplistic quarter-car models, all available in the same environment and with easy operability (as complex and with nonlinearities as required).
- **Control law design.** Without neglecting some degrees-of-freedom or essential nonlinearities, the complete nonlinear model is available within the control design environment usually by means of co-simulation interface between vehicle modeling and control design packages. Multi-objective parameter optimization (MOPO) or nonlinear quadratic regulator (NQR) approaches (described below) do not restrict the control design to oversimplified models. However, simplification (reduced-order models) are applied because:

  The design method may be restricted to linear or low-order plant models.

  The computational effort, physical insight etc. may suggest simpler models especially in the early design steps.

At any stage the use of more complex models is possible for evaluating the performance or more advanced design strategies. If the design methods allow, design-by-simulation can be performed, i.e., the use of the simulation model (in any desirable degree of complexity) within the design loop. As the simulation model is usually nonlinear, performance evaluation is possible only in the time domain.

- **Verification.** The SADTS program entails verification of the plant and also a final experimental demonstration on a controlled truck.

The main steps of the design methodology can be summarized:

1. Develop an appropriate multibody system model including all degrees-of-freedom and nonlinearities.
   a. Verify the model.
   b. Reduce the model for further design steps with respect to system order or system complexity including linearization.
2. Transfer model data into control design environment.
3. Develop the control design starting with simple models up to the advanced models. Complete the performance evaluation all along.
4. Perform multi-objective parameter optimization (MOPO): A way to achieve a fine tuning of the control system using the best (complex) model just as the engineers do with the hardware prototype.
5. Validate the dynamic structure via driving tests. Conduct trouble shooting for unexpected differences.

#### 13.2.1.1 Design Tools

To apply the described design methodology a suitable design environment with particular software tools is necessary. There is usually a tool for modeling the vehicle as a multibody system including other components, a tool for modeling the control, optimization tool and suitable interface based

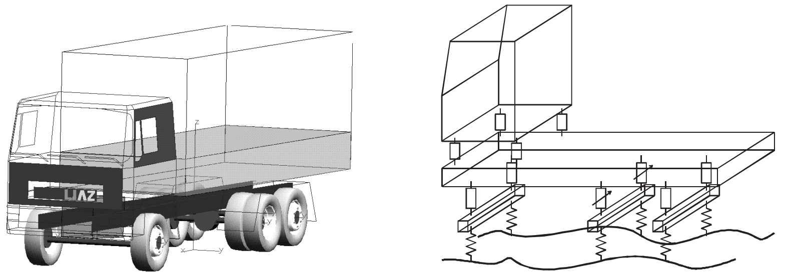


**FIGURE 13.12** Truck simulation and multibody model.

on co-simulation (see Veitl et al.[15]). A brief list includes ADAMS, SIMPACK, MATLAB-SIMULINK MATRIXx-SystemBuild.

#### 13.2.1.2 Design Models

For the control design, suitable models of the vehicle are necessary. An important result of new design methodology is that the models could be simplified (such as the quarter-car model) but they must display the main existing system's nonlinearities. Nevertheless, the final investigation and verification must be done on full 3D nonlinear simulation models.[10] Such a case is given in Figure 13.12 using a 3D model which is as close as we can recreate the real system.

### 13.2.2 Clipped Active Control

A systematic approach to semi-active control design is described here. It aims for the appropriate setting of $b_{semi\text{-}active}$ damping rate such that the damping force is nearest to the desired value (13.3) and (13.4). The ideal active force is computed according to the applied control design procedure (e.g., just sky-hook concept or LQR design). This force is then transformed (clipped) to the nearest realizable semi-active force. This is what we name "clipped active control."

This approach can accommodate any traditional (active) control design procedures. Probably, the most frequently used design methodology is the optimal LQR[3,16,17] for a linearized model (13.1), with a suitable cost function

$$J = \lim_{T\to\infty} \int_0^T (\mathbf{z}^T\mathbf{Q}\mathbf{z} + \mathbf{u}^T\mathbf{R}\mathbf{u})dt \qquad (13.7)$$

where $\mathbf{z} = [z_2 - z_1, \dot{z}_2, z_1 - z_0, \dot{z}_1]^T$ and $\mathbf{u} = F_d$. The semi-active control is then computed from the active force $F_{act} = F_d = \mathbf{u}$ as in Equations (13.3) and (13.4) and taking into account the limitation of the damper. Please note the selection of relative displacements as the state variables for practical reasons.

The clipped LQR control is investigated in Tseng and Hedrick.[18] It is really optimal control only for unconstrained semi-active control cases where $b_{min} = 0$ and $b_{max} = \infty$.

### 13.2.3 MOPO Approach

Due to the inherent nonlinearities of vehicle suspension structure, control synthesis has to be nonlinear. The traditional control methods based on linear design models cannot be used. The applicable approach for such a case is the MOPO.[19] The method is based on design-by-simulation. Control law is described in parametric form and its parameters are determined by the numerical optimization of the performance index evaluated by the simulation response of the plant to the excitations considered. Thus, by means of the MOPO approach, nonlinear models and models that

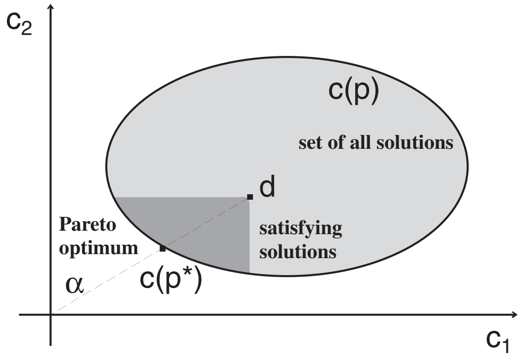


**FIGURE 13.13** Pareto-optimum for two objective functions.

cannot be analytically expressed can also be treated. This approach enables not only finding parameters of nonlinear control of nonlinear plants, but also allows finding a satisfactory compromise among the performance criteria despite the possibility that they may conflict with each other. The MOPO approach is based on a search in the parameter space (Pareto optimality) by model simulation. Free system parameters and tuning parameters (e.g., control coefficients, mass properties, or installation positions) are varied within their limits until an optimal compromise is found. The parameter optimization is finished when the maximum of all weighted criteria cannot be decreased further. The result is a point on the Pareto-optimal boundary (see Figure 13.13).

For example, in the case of nonlinear extended ground-hook, the nonlinear suspension model of different complexity, the performance index of the time integral of square of the dynamic tire forces

$$I = \int_0^{Tl\arg e} F_{10}^2 dt \tag{13.8}$$

and the excitation in the form of a cosine bump are used. By optimization the coefficients $b_1$, $b_2$, $b_{12}$, $\Delta k_{10}$, and $\Delta k_{12}$ of extended ground-hook or even their dependence on relative damper velocity are determined.[9,10]

### 13.2.4 NQR Approach

The MOPO approach suffers from common problems of global numerical parametric optimization methods. As a remedy, a new direct control synthesis was developed.[20] It was based on recent results in nonlinear optimal control called NQR (nonlinear quadratic regulator)[21] or the SDRE (state-dependent Riccati equation).[22] The dynamics of the nonlinear system is generally described by the equation

$$\frac{d\mathbf{x}}{dt} = \mathbf{f(x)} + \mathbf{g(x)}\,\mathbf{u} \tag{13.9}$$

where $\mathbf{x}(n \times 1)$ is the state and $\mathbf{u}(m \times 1)$ is the control and $\mathbf{f(0) = 0}$. If decomposition of the system dynamics exists

$$\mathbf{f(x)} = \mathbf{A(x)x} \tag{13.10}$$

which leads to the decomposed system

$$\frac{d\mathbf{x}}{dt} = \mathbf{A(x)x} + \mathbf{g(x)}\,\mathbf{u} \tag{13.11}$$

with some properties like controllability of couple (**A(x), g(x)**) in each state position **x**, then for the quadratic performance index of the infinite horizon control problem

$$J = \int_0^\infty (\mathbf{x}^T \mathbf{Q}\, \mathbf{x} + \mathbf{u}^T \mathbf{R}\, \mathbf{u})\, dt \tag{13.12}$$

there exists the suboptimal nonlinear control

$$\mathbf{u} = -\mathbf{K(x)x} \tag{13.13}$$

The state-dependent gain matrix **K(x)** is obtained as

$$\mathbf{K(x)} = \mathbf{R}^{-1}\mathbf{g}^T\mathbf{(x)P(x)} \tag{13.14}$$

where **P(x)** is the solution of the Riccati equations

$$\mathbf{A}^T\mathbf{(x)P(x)} + \mathbf{P(x)A(x)} + \mathbf{Q} - \mathbf{P(x)g(x)R}^{-1}\mathbf{g}^T\mathbf{(x)P(x)} = \mathbf{0} \tag{13.15}$$

solved in each state position **x**. Recent investigations have shown that this is the only suboptimal solution for the minimization of (13.12). Nevertheless, all simulation investigations have demonstrated excellent results.

The key problem is an efficient computation of the decomposition (13.9) which is not unique. An efficient solution that needs only the evaluation of the right-hand sides of (13.9) without the necessity to manipulate symbolically the system dynamics (13.10) (e.g., differentiation or integration) is described in Valášěk and Steinbauer.[21] This solution also takes into account the nonuniqueness of the decomposition. This procedure is capable of computing the decomposition of any complexity of dynamics **f**.

This theory can be extended for semi-active systems. In the case of semi-active actuation, the system Equations (13.12) are more complicated because the control is limited.

$$\mathbf{u}_{\min} \leq \mathbf{u} \leq \mathbf{u}_{\max} \tag{13.16}$$

The semi-active constraint (13.16) of control actuation is a highly nonlinear operation. However, the contol of such systems can be transformed into the standard NQR approach by adding artificial dynamics on the control variables d**u**/dt = (**h**(**x**,**u**) + **w**/T with new artificial input variables **w**, small time constant T, and special dynamics **h** which produces the output **u** in the limited interval (13.16) for any unlimited input **w**. The reader is referred to Valášěk and Kejval,[20] and Valášěk and Steinbauer[21] for a detailed treatment of the topic.

A comparison of optimization results of MOPO and NQR approaches with passive suspension for the performance index of comfort and road-tire forces is in Figures 13.14 and 13.15.

### 13.2.5 Preview Control

Control of vehicle suspension by previewing is an attractive natural idea of usage of driver experience with proximity sensors. The principle is to use knowledge of the road profile before passing the profile by the vehicle itself. The preview principle has been investigated since the idea of active or semi-active suspension emerged.

However, the theoretical derivation of preview control or of its optimal control is substantially more difficult than control without preview.[23,24] The only investigated objective of preview control is comfort.

The problem is the introduction and reconstruction of the preview signal. In a majority of studies it is supposed that the preview signal is really the road profile in advance and the reconstruction

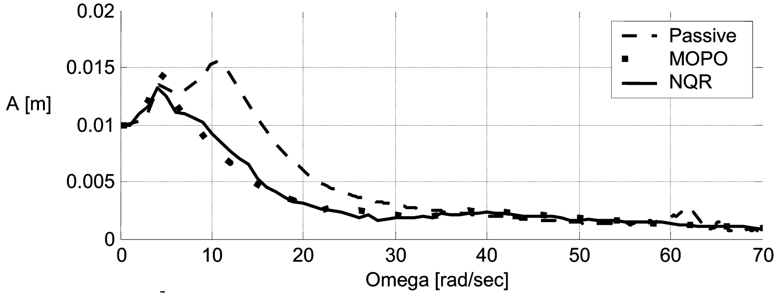


**FIGURE 13.14** Comparison of optimization for comfort.

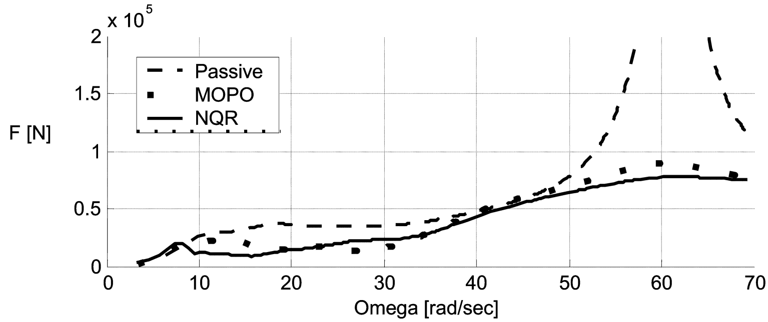


**FIGURE 13.15** Comparison of optimization approaches for road-tire forces.

is not investigated at all. However, reconstruction of the road profile is rather difficult, and the treatment of feedback signal as an absolute coordinate of road profile can cause control problems, for example, because of road altitude change.

It is also supposed that if the front axle is treated as quarter-car model it could be used for measurement and reconstruction of road profile. This process could be very sensitive. Another possibility is to apply directly the signal of unsprung mass acceleration on the front axle or tire deflection on the front axle. Thus, control of all axles is connected.

## 13.3 Properties of Semi-Active Suspensions: Performance Indexes

There can be different performance indexes for the optimization of semi-active suspension than those already considered. Performance indexes originate from the different interactions between the vehicle and the environment. Two kinds of interaction are with the human driver and with the payload. Other interactions are with roads, with bridges, and soil for off-road vehicles. These performance indexes can be further combined with the ultimate objectives to compromise conflicting requirements. Because these optimized performance indexes improve interaction of the vehicle with the environment it speaks to friendliness toward the considered property. There are the driver-load-friendly, road-friendly, bridge-friendly, and soil-friendly suspensions.

### 13.3.1 Influence on Comfort

Comfort or the interaction of the driver (passenger) or payload has been investigated since the first suspension research.[1,17,25,27,36] The performance index is the root mean square (RMS) value of the weighted acceleration value according to ISO 2631.[38] The result of semi-active suspension on comfort is evident.

### 13.3.2 Influence on Road Friendliness

Road–tire forces influenced by the semi-active suspension were investigated later.[26,27,28] Its importance recently increased with the growing demands on road transportation and the costs of road maintenance.[14,29,30] Therefore, road-friendly suspension is often spoken about.

Performance indexes besides the time integral of the square of the dynamic tire forces (13.8) can be divided into stochastic and simulated. The stochastic ones consider road damage on average, and the simulated ones try to consider the fact that road damage and maintenance are decided by one crack or hole on the road rather than average wear. The most widely used stochastic performance index is the dynamic load coefficient (DLC)[26]

$$DLC = \frac{RMS\ dynamic\ tire\ force}{static\ tire\ force} \tag{13.17}$$

and the dynamic load stress factor (DLSF)

$$DLSF = 1 + 6DLC^2 + 3DLC^4 \tag{13.18}$$

The simulated force is the aggregate force criterion. The dynamic tire forces of each axle raised to a power $n$ and applied to each location along the road are added. The $n$-th power aggregate force measured at location $k$ is

$$A_k^n = \sum_{j=1}^{N} P_{jk}^n, \qquad j = 1,2,\ldots,N_r \tag{13.19}$$

where $P_{jk}$ is the force from tire $j$ to road location $k$, $A$ is the number of vehicle axles and $N_r$ is the number of locations of irregularities on the road. The power $n$ is usually $n = 4$ representing the proportion of increased road damage due to the dynamic forces with respect to static ones. It is the well-known fourth-power law.

## 13.4 Examples of Practical Applications

### 13.4.1 Passenger Cars

Semi-active suspension for passenger car has been investigated since the first concepts. It is mostly based on the sky-hook concept and represents traditional semi-active vehicle suspension. An implementation on the passenger vehicle Nissan Cefiro in 1994 (Figure 13.16) is reported in Higashiyama.[31] The suspension is equipped with continuous shock absorbers (semi-active damper) that control all four wheels independently. The sensors are three vertical accelerometers together with the steering angle, vehicle speed, and brake sensors. Riding comfort improved by about 10% in almost the whole frequency range. Another implementation to a test passenger vehicle (Volvo) was reported in Venhovens.[25] It is based on adaptive sky-hook algorithm which takes into account the dynamic road–tire forces.

### 13.4.2 Road-Friendly Trucks

The first investigation of road-friendly truck suspension was performed by Yi and Hedrick[26] and followed by Valášěk et al.,[9] Kortüm and Valášěk,[14] and Besinger et al.[28] The recent investigation verified by prototype implementation and experimentation was done within the EU project Copernicus SADTS[9,14] and the IKA-DLR-CTU Workshop in Aachen.[32] In the first case, the prototype

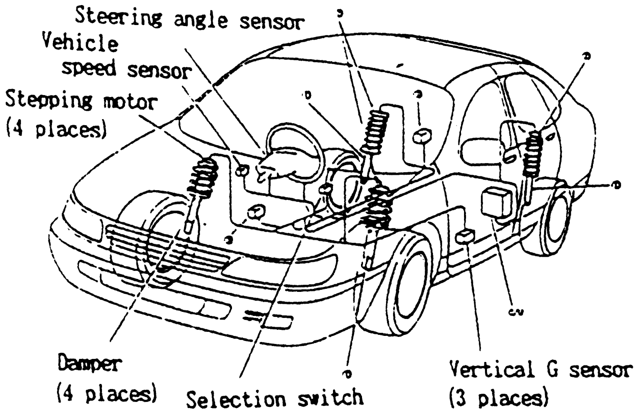


**FIGURE 13.16** Semi-active suspension of passenger car. (From K. Higashiyama, T. Hirai, S. Kakizaki, and M. Hiramoto, *Proc. of AVEC,* 331–336, 1994. With permission.)


(a) (b)

**FIGURE 13.17** Truck prototypes of SADTS Project (a) and IKA-DLR-CTU Workshop (b).

truck was a platform truck SKODA-LIAZ (Figure 13.17a), and in the second the MAN tractor with a Kögel semi-trailer (Figure 13.17b). In both cases, semi-active dampers from Mannesmann-Sachs were used on the rear-driven axle of the truck (Figure 13.9).

Each side of the rear-driven axle was equipped for control with two accelerometers (axle and chassis) and inductive displacement sensors. In addition, the road–tire forces were measured by strain gauges. Field experiments included passing the prototype truck over a good quality road such as an airport runway and then over known obstacles. The comparison between conventional, passive soft, passive hard, and controlled (semi-active) suspensions was also provided. The experimental results from the (airport) runway are shown in Figure 13.18a (evaluated as the DLSF factor) and from sinus-shaped obstacle in Figure 13.18b.

The table in Figure 13.19 summarizes the comparison of road-friendliness evaluations (advantages +/disadvantages –) for different damper settings and road excitations. Nonlinear extended ground-hook control of the damper always has advantages.

The overall evaluation of experimental results from both implementations is summarized as follows: semi-active truck suspensions to improve road friendliness demonstrate the capability of such control systems to reduce road–tire forces by about 10 to 20%, without adverse affects on riding comfort. This corresponds to a reduction of road damage up to 70% or to a possible payload increase by 1 ton for 10 tons of the design specifications. This simultaneously means a 50% reduction of truck loads fatigue.

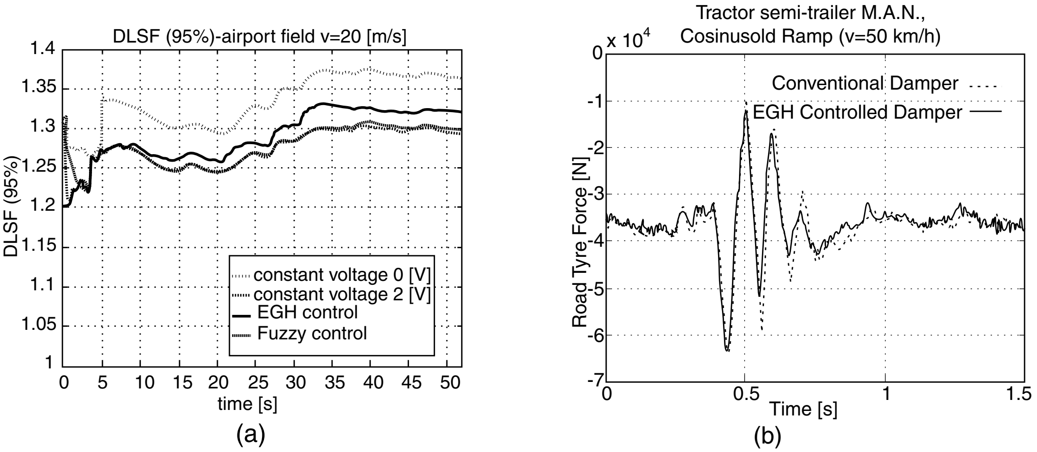


**FIGURE 13.18** Experimental results (a) DLSF on stochastic road, (b) road-tire forces on sinus bump.

| Damping | Big Obstacle | Small Obstacle | Stochastic Road |
|---|---|---|---|
| Conventional | + - | + | + - |
| Soft | - - | - | + + |
| Hard | + + | + | - - |
| Controlled | + + | + + | + + |

**FIGURE 13.19** Road-friendliness for different damper settings.

### 13.4.3 Trains

Semi-active suspensions for railway vehicles have been studied theoretically and experimentally for about 25 years. Increasing operational speeds and comfort demands require focusing on both the vertical and lateral riding quality of railway vehicles. Furthermore, it seems that lighter vehicles with one or two axles per vehicle will replace classical bogie vehicles. The vehicles without classical two-axle bogies (primary train suspension) will no longer profit from so-called mechanical pre-filtering of rail irregularities, and moreover, vehicles will be lighter. Both these factors have a negative influence on ride comfort, and emphasize the necessity of more advanced suspensions.

Railway vehicles are typically equipped with two levels of suspension. The primary suspension connects the wheel sets with the bogie. Its main function is to maintain running stability and to offer curving performance as well as reducing unsprung masses. The secondary suspension is placed between the bogie and the car body. This suspension isolates the car body from rail irregularities.

Semi-active suspension is usually applied in the secondary suspension of railway vehicles, as secondary vertical, horizontal, or yaw suspensions.[33] Implementation of the primary suspension is not common; however, it is experimentally applied to wheel set yaw control. Secondary suspension can be performed by coil springs or air springs. Air springs offer the possibility of using controllable orifice damping instead of classical hydraulic semi-active dampers as performed by the pipe between the main air spring bellow and the additional volumes (semi-active air spring).

Semi-active suspensions were implemented in several vehicles in Europe and Japan for preliminary tests.[33] One of the advanced semi-active studies was performed by Siemens SGP.[34] Among other mechatronic systems, their tilting bogie SF 600 prototype was equipped with vertical and lateral semi-active suspensions in parallel to the air springs (see Figure 13.20). Limited-state feedback control is applied for the hydraulic semi-active dampers. Semi-active suspension was also implemented on an experimental vehicle. The field experiments indicated that an improvement of up to 15% in ride quality measured by RMS acceleration can be reached, which is typical for semi-active railway vehicles.

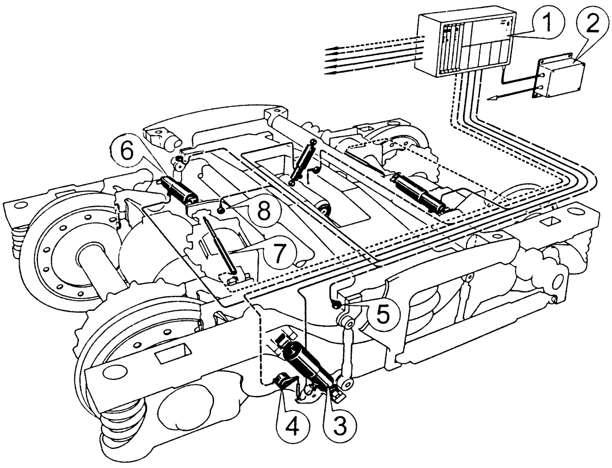


**FIGURE 13.20** Prototype bogie SF 600 with semi-active damping system. 1 = control unit, 2 = inertial sensor system, 3 = semi-active vertical damper, 4 = angle-of-rotation sensor, 5 = vertical stand-alone accelerometers, 6 = semi-active lateral damper, 7 = lateral stroke sensor, and 8 = lateral stand alone accelerometer. (From A. Stribersky, A. Kienberger, G. Wagner, and H. Müller, *Vehicle System Dynamics Suppl.,* 28, 669–681, 1998. With permission.)

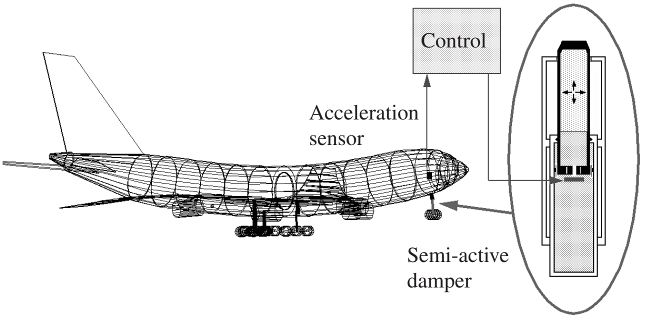


**FIGURE 13.21** Semi-active landing gear.

## 13.4.4 Airplanes

Airplanes during ground motion in airports face many problems similar to traditional road vehicles. The shock absorber of an airplane's landing gear is designed primarily for the landing impact. The resulting dynamic characteristics can be unsatisfactory for airplane ground motion. This is especially true for modern large transport airplanes. Often, the fuselage of such an aircraft is so flexible that during ground motion the first natural modes of the structure are excited to such an extent that the resulting vibrations degrade the passengers' ride comfort. Accelerations can become so strong that the pilots can no longer read the cockpit instruments. To resolve this conflict between the landing gear design for touchdown and the design for taxiing a solution based on the semi-active suspension was developed. After the landing, the airplane's shock absorber is switched from a purely passive to a semi-active mode which is controlled by the sky-hook law from an acceleration sensor in the cockpit (Figure 13.21). The simulated results suggest significant improvements in comfort.[35]

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