Design of a Stroke Dependent Damper for the Front Axle Suspension of a Truck Using Multibody System Dynamics and Numerical Optimization

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SUMMARY

A stroke dependent damper is designed for the front axle suspension of a truck. The damper supplies extra damping for inward deflections rising above 4 cm. In this way the damper should reduce extreme suspension deflections without deteriorating the comfort of the truck. But the question is which stroke dependent damping curve yields the best compromise between suspension deflection working space and comfort. Therefore an optimization problem is defined to minimize the maximum inward suspension deflection subject to constraints on the chassis acceleration for three typical road undulations. The optimization problem is solved using sequential linear programming (SLP) and multibody dynamics simulation software. Several optimization runs have been carried out for a small two degree of freedom vehicle model and a large full-scale model of the truck semi-trailer combination. The results show that the stroke dependent damping can reduce large deflections at incidental road disturbances, but that the optimum stroke dependent damping curve is related to the acceleration bound. By means of vehicle model simulation and numerical optimization we have been able to quantify this trade-off between suspension deflection working space and truck comfort.

1. INTRODUCTION

Comfort is becoming increasingly important in truck design. Truck designers tend to lower the stiffness of the axle suspensions for the benefit of comfort of both driver and

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cargo. Stiffness reduction, however, comes at the expense of increased suspension deflections. This is not favorable since an increase of the suspension working space means a decrease of the available payload volume. Therefore, several advanced suspension systems have been introduced during the last decade to improve the compromise between conflicting measures such as comfort and suspension deflection working space (see [1] for a classified bibliography). Well-known examples are semi-active and active suspensions. A semi-active suspension can rapidly adjust its settings (usually a damping characteristic), whereas an active suspension can both dissipate and supply energy by means of an actuator.

Reduction of extreme suspension deflections may be also obtained by inclusion of a stroke dependent damper in the axle suspensions. This is a very simple suspension system that may prove to be a cost-effective alternative for semi-active or active suspensions. For normal operating conditions only small suspension deflections will occur. However, when the truck comes across an incidental big road disturbance, like a pothole or a traffic hump, extra damping may be applied to compensate for the negative effects of a low stiff suspension.

Generally, the basic axle suspension consists of a spring, a damper, and an anti-roll bar (see Fig. 1). Various types of dampers are commercially available. Within certain constraints, the design engineer can freely select the damper characteristic. A stroke dependent variant can, for example, be obtained by means of extra bypass channels and valves. To optimize the characteristics of such a new damper, vehicle model simulation and numerical optimization is used. The stroke dependent damping is modeled by adding an extra damper parallel to the original axle suspension, which will only contribute during large deflections. The optimization problem is then to find the damping curve of the added nonlinear damper that will give the best performance.

Some researchers already applied design optimization in the context of nonlinear damper characteristics. Demić [2] used a modified Nelder-Mead method to optimize the parameters of elasto-damping elements in a 4-DOF (degree of freedom) vehicle model subjected to micro-roughness of asphalt road. Spentzas [3] applied Box’s method on a 7-DOF model with deterministic road irregularities alike a traffic hump. The paper of Eberhard et al. [4] comes near the design problem described in this

![Fig. 1. Schematic representation of the front axle suspension of a truck.](image-url)
paper. However, they did not consider a stroke dependent damper, but optimized the damping curve of the original shock-absorber itself. The damping curve was parameterized by Hermite-splines, and an SQP algorithm was used to solve the optimization problem when driving over a bump. Finally, Tamis [5] did indeed study and optimize a stroke dependent damper for the rear axle suspension of a truck. Response-surface models were built of the maximum suspension deflection and the maximum vertical chassis acceleration as a function of four parameters defining the nonlinear damping curve.

In the current study a design optimization tool [6] is used in combination with multibody dynamics software [7] to obtain the optimum damping curve of the extra damping for the front axle suspension of a truck. The objective is to reduce the large inward suspension deflections. Sequential linear programming (SLP) with a move limit (trust region) strategy is applied using design sensitivities calculated by finite differences. SLP is easy to implement and robust for the possibly inaccurate finite difference gradients that are due to the limited integration accuracy of the numerical multibody analysis. Furthermore, the large number of constraint equations that result from the time discretization of the time dependent constraints can be effectively handled.

First, the truck is modeled by means of a two degree of freedom system, which is often called a quarter car model. The multibody analysis package DADS [7] is used to predict the dynamic displacements and accelerations. Three typical road undulations are studied, leading to an optimization problem with multiple loading cases. Then, a full-scale three-dimensional DADS model is used in the optimization. This model has 34 degrees of freedom and is a quite realistic representation of a typical tractor-semitrailer combination [8].

2. DESIGN PROBLEM

Dampers with a characteristic alike Figure 2 are commonly used in vehicle axle suspensions. For the design problem considered here, the original nonlinear damper of the axle suspension has also such a typical relation between damper force $F_D$ and relative velocity $v_D$ as shown in the example of Figure 2. Although the rebound stage (outward relative velocity $v_D > 0$) differs from the compression stage (inward relative velocity $v_D < 0$), the fundamental damper force characteristics resemble. For low velocities the damping force shows a quadratic functional behavior caused by the turbulent oil flow through small orifices or between piston and cylinder. This part of the curve is called bleed. Then a spring controlled valve opens, and a transition towards a flatter part of the curve follows (blow-off). Finally, the curve may rise again quadratically at higher velocities (port) due to the turbulent flow resistance of the total valving.
The generic shape of the damping curve can, to some extend, be adapted to meet the specifications of the designer. The contribution of the quadratic parts, the transition shape and the slope/off-set of the linear part can be realized for a range of values. The damper manufacturer changes the diameters of the orifices, or adapts the spring controlled valve with respect to preload and stiffness. This can be separately done for rebound and compression valves.

The rebound and compression stage of the damping curve are parametrized using the following empirical relation:

$$F_D(v_D) = \frac{\beta_0 v_D^2 (\beta_1 + \beta_2 v_D)}{\beta_0 v_D^2 + \beta_1 + \beta_2 v_D} + \beta_3 v_D^2.$$  

For small velocities this curve approaches $\beta_0 v_D^2$, assumed that $\beta_3 v_D^2$ does not yet contribute ($\beta_0 \gg \beta_3$). Increasing $v_D$ yields the much flatter blow-off phase of the curve, provided that $\beta_0 v_D^2$ is substantially larger than $\beta_1 + \beta_2 v_D$. Finally, the magnitude of $\beta_3$ determines when $\beta_3 v_D^2$ comes into play.

Under normal driving conditions the suspension deflections remain small and the extra damping is not needed. To this end, the stroke dependent damping does not contribute until the inward suspension deflection (compression) comes above 0.04 m, and linearly increases towards full damping force at 0.05 m. For compressions larger than 0.05 m the additional stroke dependent damping force directly follows from the nonlinear damping curve. The objective is to find the most effective stroke dependent damping curve.
The optimization problem is to minimize the maximum inward suspension deflection subject to constraints on the vertical acceleration of the chassis for three road undulations. The design variables correspond with the parameters of the damping curve: bleed $b_1 = -\beta_0^c$, blow-off preload $b_2 = -\beta_1^c$, and blow-off stiffness $b_3 = \beta_2^c$ for compression, and bleed $b_4 = \beta_0^r$, blow-off preload $b_5 = \beta_1^r$, and blow-off stiffness $b_6 = \beta_2^r$ for rebound. They are summarized in Table 1, together with the unscaled lower and upper bounds. Parameters $\beta_3^c$ and $\beta_3^r$ are not included as design variables, but kept fixed. Preliminary calculations showed that their contribution is small in the operating range.

The following incidental road undulations have been selected: a traffic hump, a wave and a railway crossing (see Fig. 3). The traffic hump is crossed at a speed of

![Fig. 3. Road undulations. (a) Traffic hump (w1), wave (w2) and railway crossing (w3); (b) Data points of the railway crossing.](image-url)
15 km·h⁻¹. It lasts 4.25 m and has a maximum height of 0.25 m for 2.25 m of the total width. The wave corresponds with a squared sinus of 25 m long and 0.5 m high. The corresponding driving speed is 80 km·h⁻¹. Finally, the third road profile represents a typical railway crossing, which is traversed at a speed of 25 km·h⁻¹. Both traffic hump and railway crossing are built from piece-wise linear interpolations, with a slight rounding of the corners using filtering techniques.

3. QUARTER CAR MODEL

3.1. Problem Description

A very simple model of the front side of the truck is the two-DOF system as depicted in Figure 4(a). The sprung mass (chassis) and unsprung mass (wheel axle) are denoted by \( m_c \) and \( m_a \), respectively. The corresponding vertical displacements are \( y_c \) and \( y_a \). The tire is modeled by a linear spring with stiffness \( k_t \). In vertical direction it is ‘attached’ to the road which means that tire lift off cannot occur. The axle suspension consists of a linear spring \( k_s \) and a nonlinear damper \( c_s \) with a typical damping characteristic such as Figure 2. The mass and stiffness parameter values are given in the table next to the quarter car model.

Numerical analysis of this two-DOF system using DADS [7] yields the displacement and acceleration responses presented in Figure 5. The three road undulations give rise to quite different responses. Compared with the traffic hump and the wave, the railway crossing is rather innocent: the amplitudes of both deflection and acceleration are much smaller. The maximum negative suspension deflection is

<table>
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<th>Value</th>
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<td>( m_c ) [kg]</td>
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</tr>
<tr>
<td>( m_a ) [kg]</td>
<td>748</td>
</tr>
<tr>
<td>( k_s ) [kNm⁻¹]</td>
<td>600</td>
</tr>
<tr>
<td>( k_t ) [kNm⁻¹]</td>
<td>2000</td>
</tr>
<tr>
<td>( c_s ) [Nsm⁻¹]</td>
<td>Figure 2</td>
</tr>
</tbody>
</table>

Fig. 4. Quarter car model. (a) Model; (b) Parameter values.
0.11 m, and occurs for the traffic hump after 1.25 s. The maximum acceleration is near 15 ms\(^{-2}\), reached for both the traffic hump and the wave at different moments in time.

The stroke dependent damper is placed parallel to spring \(k_s\) and damper \(c_s\) of the original axle suspension. Now, the optimization problem is to determine the parameters \(b\) describing the curve of the extra damper that will minimize the maximum inward (i.e., negative) suspension deflection:

\[
F(b) = \max_{t \in [0,3]} \{y_c(b, t) - y_a(b, t)\} \tag{1}
\]

subject to a maximum chassis acceleration \(a_c^m\) of 20 ms\(^{-2}\):

\[
g_1(b, t) = \frac{|\dot{y_c}(b, t)|}{a_c^m} \leq 1 \quad \forall t \in [0,3], \tag{2}
\]

a maximum outward (i.e. positive) suspension deflection \(d_{so}^m\) of 0.14 m:

\[
g_2(b, t) = \frac{y_c(b, t) - y_a(b, t)}{d_{so}^m} \leq 1 \quad \forall t \in [0,3], \tag{3}
\]

and a maximum tire deflection \(d_t^m\) of 0.09 m:

\[
g_3(b, t) = \frac{|y_a(b, t) - y_r(t)|}{d_t^m} \leq 1 \quad \forall t \in [0,3]. \tag{4}
\]

The corresponding side-constraints have been given in Table 1.
3.2. Optimization Results

The optimum design problem is solved using the sequential approximate optimization tool described in [6]. For computational convenience the max-value objective function is rewritten as:

$$ F(b) = b_{n+1} $$

with an additional constraint:

$$ g_4(b, t) = -\{y_c(b, t) - y_a(b, t)\} \leq b_{n+1} \quad \forall t \in [0, 3]. $$

Furthermore, every time dependent constraint is replaced by 601 time point constraints, equally distributed on the time interval of 0 to 3 s. Time point accelerations and displacements are linearly approximated with respect to the design variables. The required gradients of the displacements and accelerations are obtained by finite differencing, with a relative step size of $10^{-3}$. The integration accuracy of the DADS analysis is set to $10^{-7}$. The optimization converges for an accuracy of 0.1% (See [9] for more details).

Within 11 cycles the approximate optimization process converges, using initial move limit factors of 30% for all design variables. The optimization history is visualized in Figure 6. Clearly, the traffic hump is decisive. The initial and optimum design variable values are given in Table 2. For compression, the bleed $b_1$ moves towards a small value, while the preload $b_2$ increases to its upper bound value. The stiffness of the compression blow-off $b_3$ remains relatively small. The design variables of the rebound stage behave different, except for the rebound blow-off $b_6$.

![Fig. 6. Optimization history of the quarter car model with stroke dependent damping for a maximum acceleration of 20 ms$^{-2}$. (a) Maximum inward suspension deflection (solid) and maximum chassis acceleration (dotted) for traffic hump ($w_1$) and wave ($w_2$); (b) Initial (dash dotted) and optimum (solid) damping curve.](image-url)
which stays at the lower bound value. The rebound bleed $b_4$ is set at the upper bound level, whereas the preload $b_5$ hardly changes. In Figure 7, the suspension deflection and chassis acceleration corresponding to the optimum design are plotted. They can be compared with the responses of the initial design (see Fig. 5). The maximum negative suspension deflection has been minimized from 0.102 m to 0.0795 m with the maximum acceleration bounded to 20 ms$^{-2}$. The constraint on the tire deflection has never become active.

It is quite difficult to preselect the maximum acceleration bound beforehand. To judge the suspension capability, the design engineer would probably prefer a compromise plot of the maximum attainable suspension deflection versus the maximum chassis acceleration. Optimizations have been carried out for a range of acceleration bounds varying from 15.5 ms$^{-2}$ to 25 ms$^{-2}$. A design with a maximum acceleration lower than 15 ms$^{-2}$ was not found. Accelerations did not exceed

![Fig. 7](https://via.placeholder.com/150)

**Fig. 7.** Displacement and acceleration responses of the quarter car model with optimized stroke dependent damping for the traffic hump (solid), wave (dashed), and railway crossing (dotted). (a) Suspension deflection; (b) Chassis acceleration.

<table>
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<tr>
<th>Design variable</th>
<th>Initial design</th>
<th>Optimum design</th>
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<tbody>
<tr>
<td>$b_1$ [10$^6$ Ns$^2$m$^{-2}$]</td>
<td>10.0</td>
<td>0.563</td>
</tr>
<tr>
<td>$b_2$ [10$^3$ N]</td>
<td>20.0</td>
<td>200</td>
</tr>
<tr>
<td>$b_3$ [10$^3$ Ns$^{-1}$]</td>
<td>0.700</td>
<td>1.35</td>
</tr>
<tr>
<td>$b_4$ [10$^6$ Ns$^2$m$^{-2}$]</td>
<td>10.0</td>
<td>22.0</td>
</tr>
<tr>
<td>$b_5$ [10$^3$ N]</td>
<td>20.0</td>
<td>28.6</td>
</tr>
<tr>
<td>$b_6$ [10$^3$ Ns$^{-1}$]</td>
<td>0.700</td>
<td>0.700</td>
</tr>
</tbody>
</table>
25 ms$^{-2}$, whatever the curve of the stroke dependent damping. Furthermore, optimizations have been restarted from some different initial designs. The calculated optima are plotted in Figure 8(a). The solid line connects the designs with the smallest negative suspension deflection. Designs below this line can probably not be realized. Though, one hundred percent guarantee cannot be given since the global optima may not yet have been found.

Figure 8(a) shows that the additional stroke dependent damping can hardly reduce the maximum suspension deflection if the maximum acceleration is limited to the level of the original axle suspension. The original design without extra damper is marked with a plus sign and nearly lies on the compromise line. The stroke dependent damper can reduce the maximum negative suspension deflection, but only at the expense of increased maximum accelerations for the incidental road undulations.

Note that the comfort for (stochastic) road conditions with suspension compressions smaller than 0.04 m is not affected.

Some trends in the optimum design variable values are present (see Fig. 8). All optimum damping curves tend towards a rather flat blow-off for the rebound stage, and, to a less extent, for the compression stage as well (small $b_6$ and $b_3$). Furthermore, the bleed on the rebound side ($b_4$) is usually near its upper bound value. The other optimum design variable values show a much larger variation. The bleed ($b_1$) and preload ($b_2$) of the compression stage tend to increase for higher maximum allowed acceleration. This corresponds with what one would expect to happen. They also seem slightly correlated: lowering $b_1$ can partly compensate the effect of enlarging $b_2$, and vice versa. The same sort of correlation can be observed for $b_4$ and $b_5$. This confirms the existence of multiple local optima. Finally, the design variable values of the compression stage ($b_1$, $b_2$, and $b_3$) are much more determined by the maximum allowed acceleration than the design variables of the rebound stage.
4. FULL-SCALE MODEL

4.1. Problem Description

The quarter car model presented in the previous section is the most simple model incorporating the compromise between suspension working space and comfort. It requires only little computer time, and numerical optimization can be carried out without great difficulty. However, due to its simplicity, the quarter car model is not very accurate. A great deal of the dynamic behavior of the truck is not included. More complex models show a better correspondence with the actual behavior. Bekkers [8], for example, developed a full-scale (3-D) DADS model of a typical tractor-semitrailer combination (see Fig. 9).

The 3-D DADS multibody model is a quite close reproduction of the truck tractor and semitrailer. The main model components of the tractor are chassis, front and rear axles, axle suspensions, tires, engine, steering system, cabin, and cabin suspension. Further major components such as the fuel tank have been included. The semitrailer consists of a frame, three axles, and axle suspensions. Each component is modeled by one or more rigid bodies and couplings. Parameter values of nonlinear damper and spring characteristics are based on experimentally determined behavior. The complete model has 34 degrees of freedom, and is used in the optimization like the quarter car model in the previous section.

The full-scale model has been slightly adapted for optimization purpose. The main adjustment is the removal of bump stops. The bump stops represent rubber stops that restrict the suspension deflection working space. Contact with a bump is more or less an impact situation, leading to a series of peak accelerations added to the global response. These peak accelerations are of no interest for the design of the stroke dependent damper. The damper should reduce the suspension deflections to avoid the bump contacts. Besides, peak accelerations raise serious difficulties for the numerical optimization, and give rise to high computational cost of the numerical analysis.
Two road surfaces are considered: the traffic hump and the wave. Calculations for the quarter car model showed that the railway crossing does not significantly contribute. The truck follows a straight line such that both left and right wheels meet the road undulation at exactly the same time. So, left and right side of the truck behave identically, apart from small differences due to asymmetrical mass distribution (left and right are defined from the driver’s point of view).

The suspension deflection and vertical chassis acceleration at the front left side of the truck are shown in Figure 10 for traffic hump and wave. During the first 5 s the truck drives on a flat road. At \( t = 5 \) s the front wheels meet the road undulation. The wave causes a maximum negative suspension deflection of about 0.12 m and a maximum acceleration of 20 m/s\(^2\). The global behavior of relative displacements and accelerations shows a good resemblance with the responses calculated for the quarter car model (cf. Fig. 5). The timing of maxima and minima corresponds reasonably well. However, the amplitudes show larger discrepancies. Differences of a factor two are present for some of the extrema. Such large differences are possible since the quarter car model does not include all vehicle dynamics.

The front axle suspensions on both sides of the truck model are extended with an extra nonlinear damping that has to supply the stroke dependent damping for negative suspension deflections below \(-0.04\) m. The optimization problem is exactly the same as mentioned in Section 3, except for the constraints on the maximum tire deflection which are removed. So, only the suspension deflection and vertical chassis acceleration at the front left side of the truck are included in the optimization problem.

Figure 10(b) shows higher frequencies in the calculated accelerations compared with the quarter car model. Therefore, the density of the time point distribution is increased to 801 points on a smaller interval of 5 to 7 s. The integration accuracy is reduced to \(10^{-4}\), while the optimization convergence parameters and the finite difference step for the design sensitivities are increased to 1%. This is much cheaper...
in computational cost compared with an accuracy of $10^{-7}$. Even then, a numerical analysis of the full-scale model is still two hundred times more expensive than a quarter car analysis. Therefore, the number of design variables is reduced as well. Design variables $b_3$, $b_4$ and $b_6$ are kept fixed at $0.7 \cdot 10^3$ Ns$^{-1}$, $22 \cdot 10^6$ Ns$^2$m$^{-2}$ and $0.7 \cdot 10^3$ Ns$^{-1}$, respectively. Optimizations in the previous section have shown that these design variables tend to go to their upper or lower bounds.

4.2. Optimization Results

For a maximum allowed chassis acceleration of $20$ ms$^{-2}$ the optimization history is shown in Figure 11. The initial design equals the optimum parameter values as found for the quarter car model in Table 2. Convergence occurred after fourteen cycles. The bleed during compression $b_1$ increases about a factor ten to $5.8 \cdot 10^6$ Ns$^2$m$^{-2}$. Preload $b_2$ is halved towards: $10 \cdot 10^4$ N. The rebound stiffness $b_5$ hardly changes, $33 \cdot 10^3$ N instead of $29 \cdot 10^3$ N. So, the first two design variables show a significantly different value compared with the optimum design of the quarter car model. Furthermore, the wave is decisive instead of the traffic hump.

The suspension deflection and vertical chassis acceleration of the calculated optimum design are plotted in Figure 12. It is clearly visible that the negative suspension deflections below $-0.04$ m have been suppressed. For the traffic hump, this comes at the expense of an increase of the maximum acceleration. It is striking that this seems to be mainly caused by a more pronounced presence of the high frequency vibrations on the acceleration response. The global response stays at about the same magnitude. The same effect can be observed for the wave, but to a far less

Fig. 11. Optimization history of the full-scale model with stroke dependent damping for a maximum acceleration of $20$ ms$^{-2}$. (a) Maximum inward suspension deflection (solid) and maximum chassis acceleration (dotted) for traffic hump ($w_1$) and wave ($w_2$); (b) Initial (dash dotted) and optimum (solid) damping curve.
extent. Here the maximum acceleration is hardly affected by the amplification of the vibration.

Some additional optimizations have been carried out for other acceleration bounds than 20 ms$^{-2}$. Optimum designs with accelerations lower than 17.5 ms$^{-2}$, and higher than 21 ms$^{-2}$ have not been found. Figure 13(a) shows the compromise plot, based on optimizations with acceleration bounds 17.5, 19, 20, and 22.5 ms$^{-2}$. The optimization runs for the former two bounds did not converge. The desired accuracy could not be reached, which is probably caused by the combination of inaccurate sensitivities and high frequencies in the acceleration responses. For each of these two bounds, the plotted optimum design corresponds with the design nearest to the constraint bound.

![Fig. 12. Displacement and acceleration responses of the full-scale model with optimized stroke dependent damping for the traffic hump (solid), and wave (dashed). (a) Suspension deflection; (b) Chassis acceleration.](image)

![Fig. 13. Optimum designs of the full-scale model with stroke dependent damping for several acceleration bounds. The plus sign marks the original axle suspension without extra damping. (a) Compromise plot of the full-scale model (solid) and the quarter car model (dotted); (b) Damping curves.](image)
The compromise plot now clearly lies below the original full-scale model design without stroke dependent damping (plus sign). So, contrary to the quarter car model, the maximum negative suspension deflection can be reduced without increasing the maximum acceleration. However, it should be noted that this is mainly caused by the fact that the maximum acceleration due to the wave is hardly affected by the damping curve. Similar to the quarter car problem, the optimum design variable values of $b_1$, $b_2$ and $b_5$ are influenced by the acceleration bound. Figure 13(b) shows that the compression bleed $b_1$ increases if a higher maximum acceleration is allowed. The preload $b_2$ tends to increase as well. Parameter $b_5$ shows a completely different behavior compared with the quarter car model (compare Fig. 13(b) and Fig. 8(b)). The rebound preload decreases instead of increases as a function of the maximum acceleration.

5. CONCLUSION AND DISCUSSION

Inclusion of stroke dependent damping in the front axle suspension of a truck can reduce large (inward) deflections appearing at incidental road disturbances. In this way the increase of suspension deflections due to stiffness reduction can be compensated. At the same time, it may lead to increased chassis accelerations the moment the extra damping comes into play. This possible loss of comfort is preferable above contact with bump stops that restrict the suspension working space. The most important impediment, however, is the required stroke dependent damping force. In the current study, the damper force may rise to 200 [kN], which is about two and a half times as high as the damper force of the original absorber. The question is whether this is a cost-effective solution because frame and brackets must be redesigned to carry the increased loads. Possibly, the optimization problem definition should be reconsidered, e.g. by decreasing the design space or by including the damper force as objective function or constraint.

No single optimum damping curve has been found that generally yields the best stroke dependent damping. The shape of the curve depends upon the maximum allowed acceleration. Still, the optimization results point towards the following design rules for the stroke dependent damping. First of all, the blow-off stiffness can remain small. Furthermore, the bleed and blow-off preload are the most important variables influencing the compromise between suspension deflection and chassis acceleration. To reduce inward suspension deflections, the compression side of the damping curve is of main interest. It is recommended to choose the compression bleed and preload as high as possible until the selected acceleration bound is reached. Most attention should be paid to the preload since this parameter defines the magnitude of the damping force that can be generated. One may be tempted to apply the high bleed and preload levels to the rebound side as well. However, high rebound damping can cause
the axle suspension to show a bad recovery and to stay at large compressions. This means that bump contacts may occur for a second road disturbance or steer maneuver shortly after the first one. Additionally, the roadholding may be affected.

The design optimization was first applied to the quarter car model before the full-scale model was considered. The basic idea is that a rather simple model gives the opportunity to do several optimization runs and to get a feeling for the design problem. If necessary, the optimization problem can be reformulated or the mechanical model can be adapted. Hopefully, trends become visible, and the designer starts to form an idea of what is actually searched for. The latter is often far from a foregone conclusion. Then the step can be made towards a more complex analysis model. Starting point is that the basic response behavior corresponds for both models. The design engineer should now be able to do the optimization of the complex model in a structured way without too much experimentation.

For a well phased optimization the step from quarter car model to full-scale model has been pretty large. A reasonable resemblance of the global response behavior is present. Though, the discrepancies cause the compromise lines to deviate significantly. Large differences are especially present for the chassis acceleration. A more detailed confrontation of quarter car and full-scale model is desired to find out the precise cause of the deviation, and to determine which parts of the tractor-semitrailer behavior can and which parts cannot be neglected for the design optimization. Possibly, the quarter car model should be replaced by, for example, a six-DOF vehicle model. If the model can be kept simple the number of road excitation inputs in the optimization problem can be increased without difficulty.

Anyway, the design optimization tool has proven its value. For the quarter car model reasonable to good convergence was found. The moment the extra damping comes in, a sharp acceleration peak may be present, especially for the traffic hump. This suggests to let the integration procedure define which time points are included in the optimization, instead of an equal distribution on the time interval. Difficulties observed for the full-scale model are mainly due to inaccurate finite difference sensitivities. This underlines the necessity of an integrated multibody analysis and design sensitivity analysis. If, from a practical point of view, sensitivities can only be obtained by finite differences with a poor accuracy, a multi-point approach such as used in [10] for optimum crashworthiness design may be considered instead.

REFERENCES


